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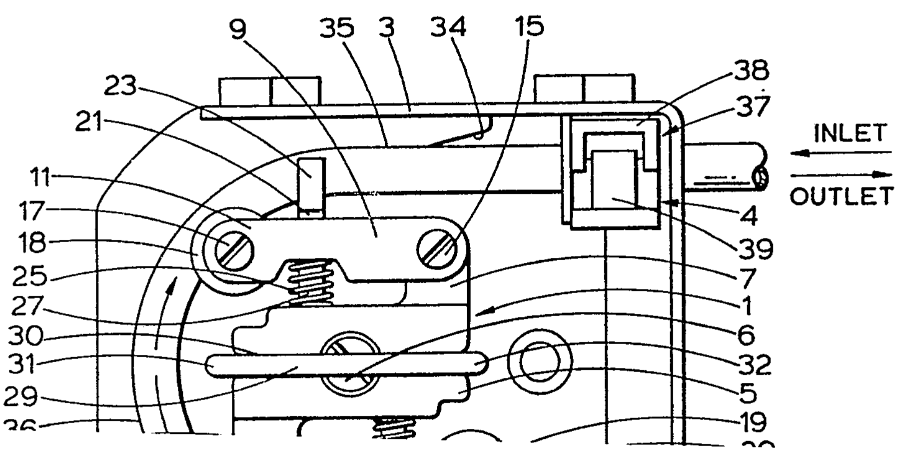
(54) **Peristaltic fluid-machines**

(57) A peristaltic pump module has a part-circular track 2, a rotor 1, arms 9 pivoted thereto, each arm carrying a roller 18, and springs 25 to bias the arms outwardiy and press the rollers 18 against a flexible tubing located on the track. The relative dimensions and geometry of the module are such that the pressure exerted on the tubing by the rollers is greater in one direction of rotation than in the other thus giving high-pressure and low-pressure modes of operation. Alternatively, the rollers may be mounted on spring-loaded radially-movable portions of the rotor, Fig. 6 (not shown).



ERRATUM

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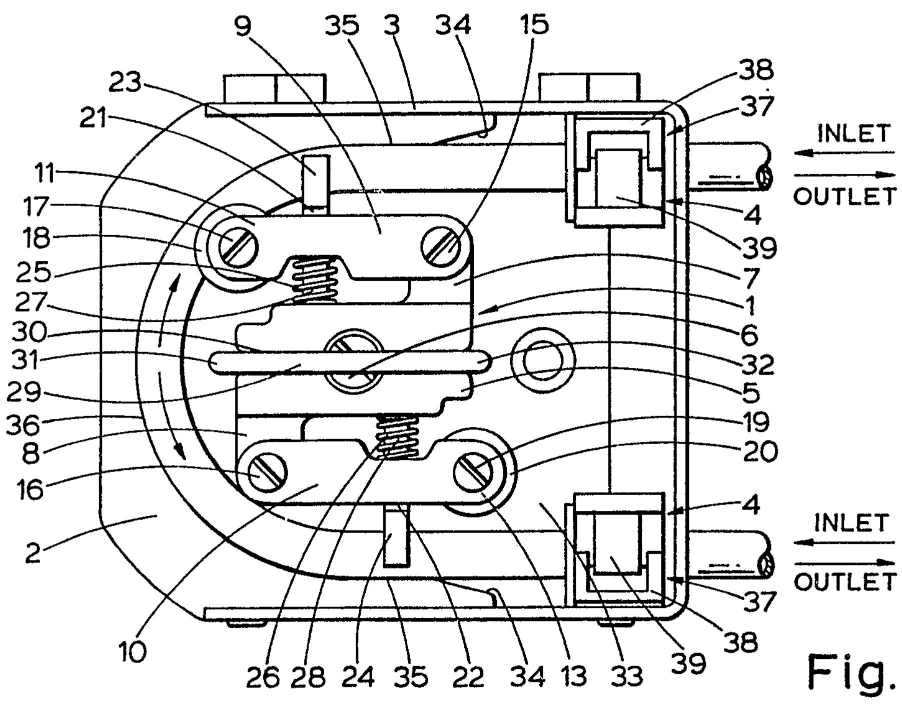

Fig.1

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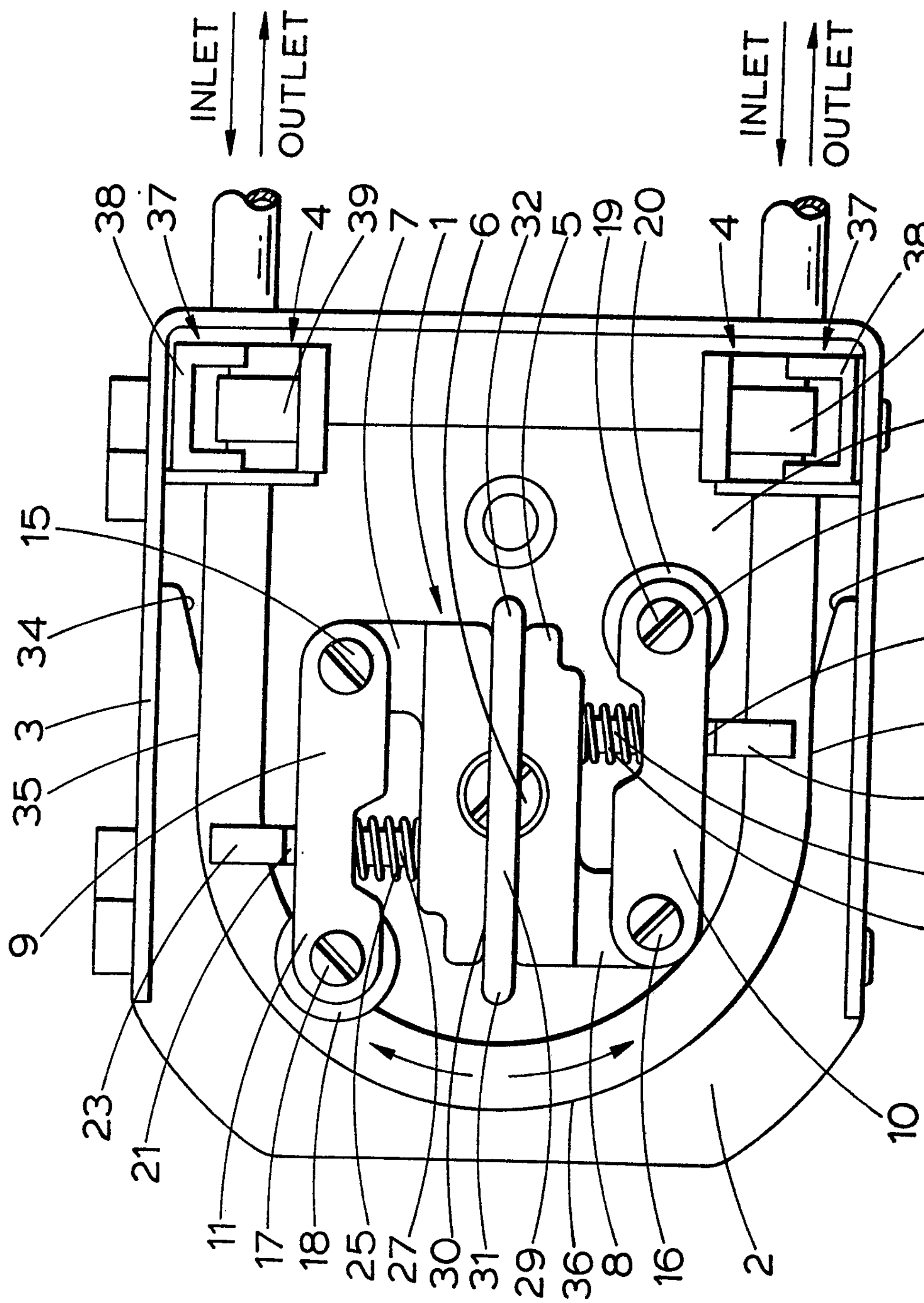

Fig.1

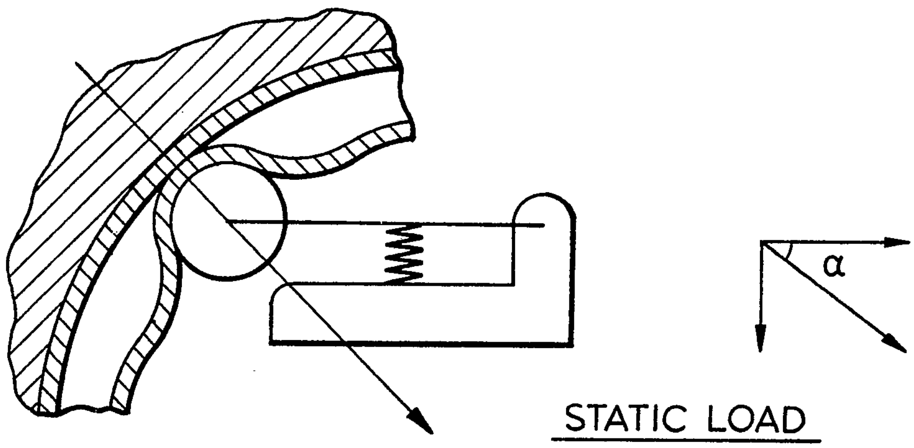

Fig.2a

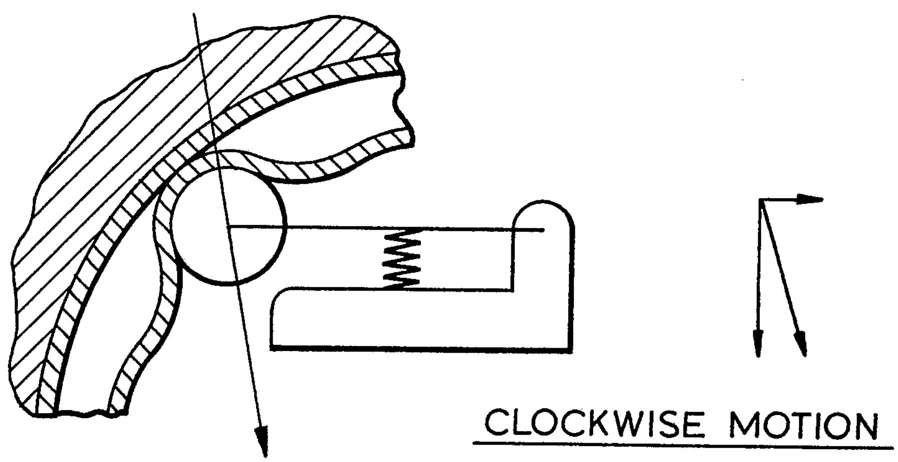

Fig.2b

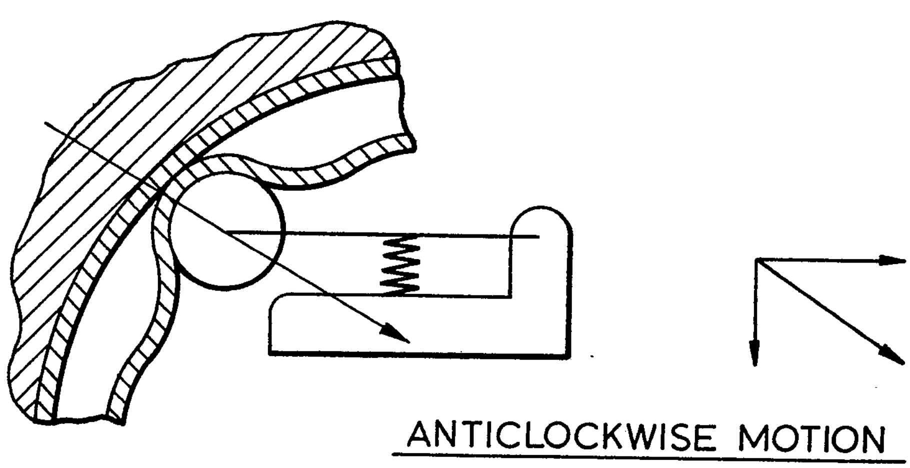

Fig.2c

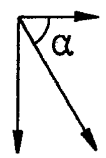

Fig. 3a

Fig. 3b

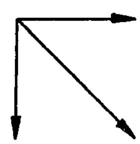

Fig. 3c

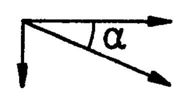

Fig. 4a

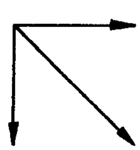

Fig. 4b

Fig. 4c

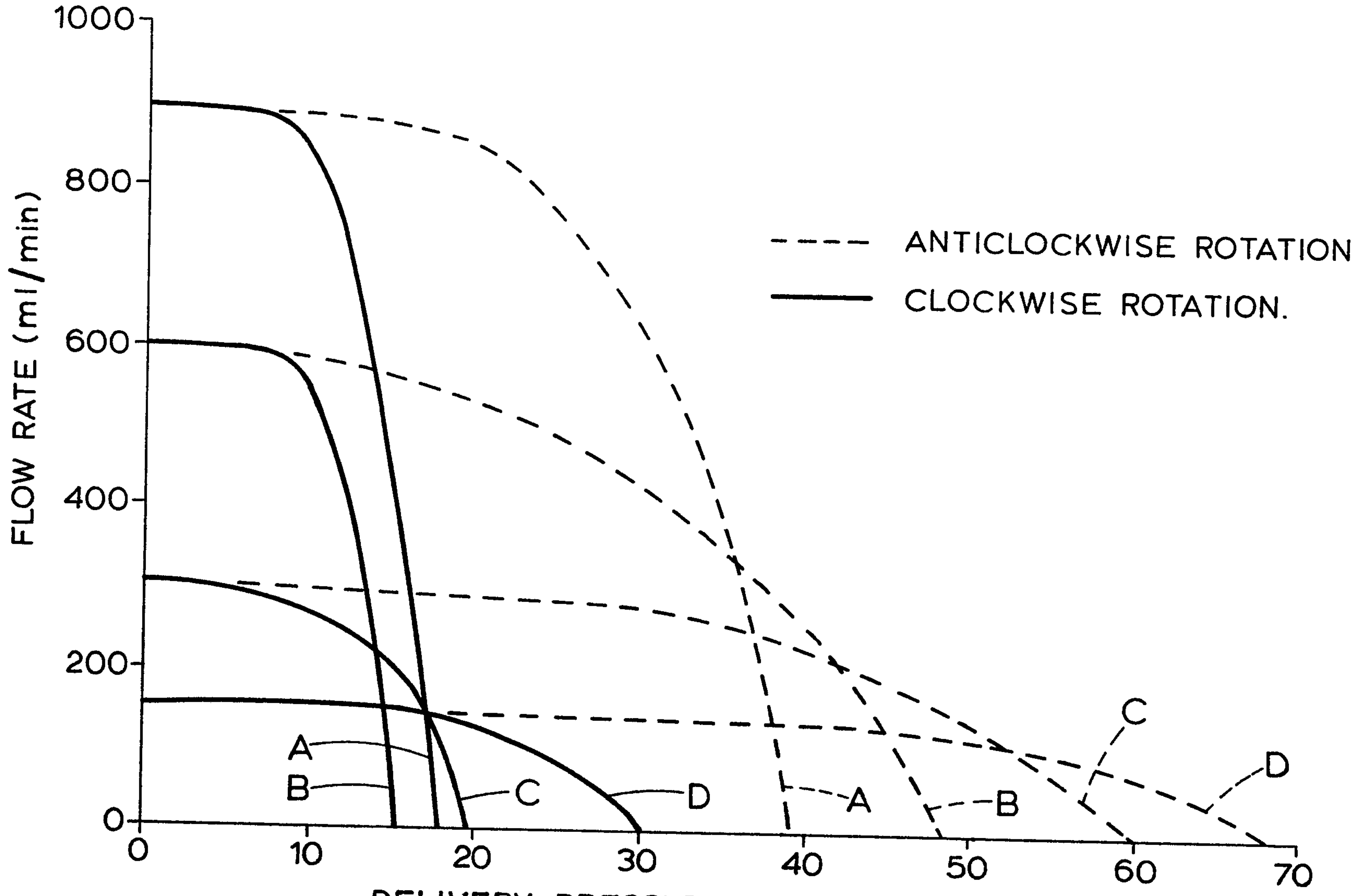
FLOW RATE (ml/min)
1000
800
600
400
200
0
0
10
20
30
40
50
60
70
DELIVERY PRESSURE (P.S.i.)
ANTICLOCKWISE ROTATION
CLOCKWISE ROTATION.
A
B
C
D
A
B
C
D

Fig.5

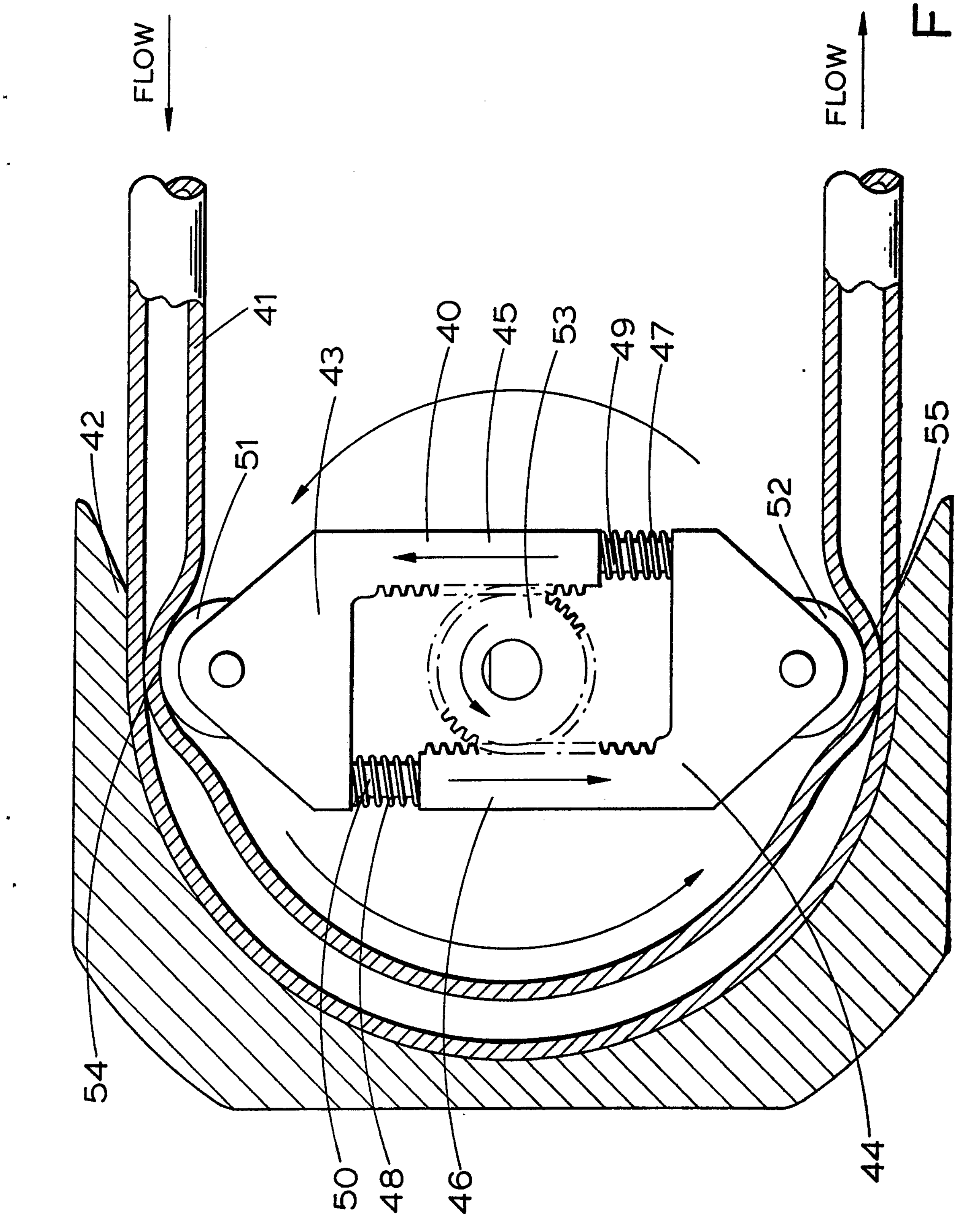

Fig.6

SPECIFICATION
Pump Module

This invention relates to a pump module, that is, a mechanism which when attached to a source of motive power exerts a pumping action on a fluid. More especially the invention relates to a pump module for a peristaltic pump, example as used in the pumping of live blood in surgical operations or renal dialysis or as used in a wide range of chemical industries and especially the foodstuffs, toiletries or beverage industries.

A peristaltic pump consists essentially of a guide surface for locating a length of flexible tubing and an operating member which moves along the tubing, compressing it against the guide surface, so that any fluid contained therein is progressed along the tubing in the direction of movement.

For convenience, there is usually provided a rotor mounting one or more of the operating members for travel in a circle closely adjacent to a part-circular, or closely similar arcuate, guide surface, so as to compress a length of tubing in the sucessive gaps. Conceivably one such member, often two, but most usually three such operating members, (although four or more have been proposed) embodied as rollers idly mounted on the rotor can be used.

By way of further elaboration, a number of such modules can be mounted around a common spindle, thus giving for example, a multichannel dosage pump for flavouring or colour materials in the food or drink industries, or for use with analytical equipment in the chemical or biochemical field.

As indicated, therefore, such modules are capable of a wide range of application. However, because the fluid tubing itself is an integral part of the pump module, a range of such modules is necessary for efficient marketing, since it is difficult to adjust one model of pump to accommodate a wide range of uses.

For example, when pumping live blood for a renal dialysis machine only a relatively low pressure is needed. However, the tubing must be made of blood-compatible material i.e. a material neither inducing clots nor promoting cell-damage, and the squeezing action must be gentle and not so fast that cell-damage is caused by rapid tubulence in the advancing nip. This gives constraints on tubing material; on inside diameter to give the necessary flow rate at an acceptable rotor speed; on outside diameter to give the necessary flexibility of tube wall; and on spacing to ensure that the tube walls close together but are not squeezed unduly to lead to rapid deterioration.

It is not always possible therefor to take the module as above and use it for some other purpose. Higher desired back-pressures may need greater clamping pressure against the guide; but this will lead to undue wear of soft blood-compatible polymers. Therefore different harder tubing may be necessary; but this will have less compressibility thus absorbing some of the higher clamping pressure. Again, desirable changes of flow rate may be greater than can be accommodated by simple rotor speed variation since unduly low speeds may give unduly pulsatile injections; but incorporation of different inside-diameter tubing of the same outside diameter, leads to flexibility problems or pressure problems.

There is also a maintenance problem connected with wear of the tubing either by abrasion or flexure. A three-arm rotor at 60 r.p.m. squeezes and rubs against its relatively small associated length of tubing about 250,000 times in every day's operation. This magnifies up the effects of small irregularities in the rotor or tubing, of or wear in the rotor or roller bearings, to necessitate frequent replacement of tubing, especially where this is very soft or very stiff.

This maintenance problem is exacerbated by adjustability of the module. As explained above, wide adjustability is not always possible. However, one common expedient is to move the semicircular guide towards the rotor, if it is desired to increase operating pressure to some small extent. This type of adjustment means that the gap between the rollers and the guide is not constant, to achieve which would need a change in guide radius, not displacement. Thus there are regions of extra wear towards the ends of the guide.

All in all, therefore, it can be said that any model of a peristaltic pump has hitherto been optimised in size and shape characteristic to one primary intended use, it being fortunate coincidence if it could be used with a different combination of liquids, flow rates, pressures and tubing.

According to the present invention there is now proposed a peristaltic pump module which has two modes of use, exerting substantially different pressures on the tubing and thus provides a much wider range of suitable utility.

In one aspect the present invention provides a pump module for a peristaltic pump of type wherein at least one operating member passes within a clearance of an arcuate track such as to squeeze together the walls of flexible tubing located along the track and thereby exert a pumping action; wherein the operating member is a roller resiliently mounted in relation to a driven rotor, the dimensions and method of mounting being such that a significantly different resultant pressure is exerted on the tubing during clockwise rotation from that exerted during anticlockwise rotation. Usually the clockwise progression is the lower. Usually, moreover, more than one operating member is so mounted on the rotor, two being preferred but three or more being possible.

In one preferred embodiment there is provided a part-circular track; a rotor body or block coaxial therewith; at least one rotor arm mounted on said block, said arm possessing at or near one end a pivot means for pivotal attachment to said block, and at or near the other end an idly mounted roller; and resilient biasing means located to bias the rotor arm away from the rotor block so that said roller bears against and compresses together the walls of a length of resilient tubing located against said track: wherein the effective direction of load to compress the tubing, under static load conditions being the line joining the roller and rotor axes, is deflected to one side or other as the rotor rotates clockwise or anticlockwise to exert a dynamic load; and the effective magnitude of such dynamic load, governing the operation pressure of the module is significantly different from one side to the other being such as to exert the same compressive force, when resolved in the necessary direction, to resist the resilient biasing means, to thereby maintain the same shape relationship.

Such a module preferably possesses a semi-circular track and two rotor arms spaced at 180° around the block. It is preferred for the rotor arms to be essentially straight and parallel to one face of the block, so that the resilient biasing means can comprise one or more compression springs between each arm and the block.

In usual practice the track is sufficiently wide and the rotor is thus sufficiently deep that it is preferred to have the or each rotor arm with two forwardly projecting lugs to mount between them the idle roller on its spindle, and two rearwardly projecting lugs to accomodate the pivot spindle and an optional rotary sleeve, e.g. in combination with two like lugs on the rotor block. To prevent the tubing from slipping off, there can be two compression springs for each arm, located on a common line at 90° across each rotor arm, each loosely guided on a pin protruding outwards from the rotor arm to provide two spaced guide pins for the tubing and each, if desired, with a lose rotary sheath.

In an alternative emobidment there is provided a part-circular track; a composite rotor body coacial therewith; and an idly mounted roller at each end of said rotor body to bear against, and compress together the walls of a length of tubing located against said track: wherein the rotor body is formed as two like portions each possessing a straight limb extending towards the other portion symmetrically with respect to the axis of rotation, the facing edges of each limb being toothed and the two portions being resiliently biased away from each other: whereby locating the teeth on said limbs around a tooth drive wheel provides in one direction of rotation thereof the additive effect of the biasing pressure plus the dynamic motor load, to compress the tubing, but in the other direction of rotation only the effect of the biasing pressure minus the dynamic motor load.

In this embodiment the biasing means are preferably compression springs around forwardly extending pins on the limbs which extend in each case into a corresponding socket on the other portion.

The invention will be further described with reference to the accompanying drawing, in which:

Fig. 1 is a front view of a pump module:

Figs. 2*a*, 2*b* and 2*c* illustrate features of operation of such a module;

Figs. 3*a*, 3*b* and 3*c* similarly illustrate features of operation of a module outside the scope of the invention;

Figs. 4*a*, 4*b* and 4*c* similarly illustrate features of operation of a module similarly outside of the scope of the invention but in the opposite direction to that of Figs. 3*a*, 3*b* and 3*c*;

Figure 5 is a graph showing performance of a module as shown in Fig. 1;

Figure 6 is a front view of the rotor of an alternative pump module.

In Figure 1 is shown a pump module with a rotor assembly 1, a track 2, a fold-over clear polymer safety shield 3 and tube clips 4.

Rotor assembly 1 is the main component feature. It consists of block 5 with central bore 6 for mounting on the spindle of a drive module and two pairs of outwardly directed mounting lugs 7 and 8 (only one member of each pair being shown) at opposite ends of of opposite faces.

Rotor arms 9 and 10 are similarly provided with pairs of mounting lugs 11, 12, 13 and 14, again one only member of each pair being visible. Rotor arm lugs 12 are mounted on a common spindle 15 with rotor block lugs 7, and arm lugs 13 on a common spindle 16 with block lugs 8, spindles 15 and 16 in each case possessing a freely rotary sleeve (not shown) between the lugs. Rotor arm lugs 11 carry a spindle 17 mounting an idle roller 18, and similar lugs 14 carry spindle 19 and roller 20.

Each rotor arm 9 and 10 has a pair of bores (not shown) near the base of pairs of lugs 11 and 14 respectively, to accommodate a freely moving pair of pins as at 21 and 22 having rotary guides 23 and 24 on their free ends (only one of each visible) and surrounded by a compression spring 25, 26 on their shanks as at 27, 28.

Across bore 6 is a foldaway handle 29 located in groove 30, with handle end portions 31, 32 extending back at 45°. this handle 29 is pivoted at the approximate junction of the end portion 31 and its main length, and is capable of opening through 135° so that the other end portion 32 protrudes at 90° from the face of the rotor.

Track 2 is of the same depth as the rotor, and is formed as a generally C-shaped boundary to a backing plate 33. On its internal surface it has lead-in portions 34, straight portions 35 and a semi-circular portion 36 with its centre at the axis of rotor movement.

Safety shield 3 hinges over the front of the assembly, protecting the moving rotor and having a transparent face parallel to backing plate 33. The shield 3 leaves an opening to one side of the assembly, the right-hand side as shown.

To either side of this opening are located tube clips 37 for holding but not of course closing the tube. As shown they are each shaped as flexible plastic jaws, of integral construction, with projecting outer and inner jaw members 38 and 39 respectively: they could be more positively constructed and fixed for a particularly large or heavy tubing.

To assemble the pump module for use the rotor bore 6 is locked over a suitable spindle of a reversible multi-speed drive module. A suitable tubing (one which, when compressed fits in the space between roller 18 and semi-circular portion 36 of the track, that is, one whose wall thickness is half that distance) is fed between a pair of rotary guides 24 and the handle 29 is folded out. By manual clockwise turning of the rotor 1 the forward end of the tube can be progressively introduced over the rotary sleeve of spindle 16, over roller 18, between the pair of rotary guides, and out past the rotary sleeve or spindle. The tube can thereafter be clipped in the clips 4.

In typical operation the tubing gripped between the roller 18 and track 36 separates liquid in the upper portion of tubing, referring to the drawing, from the remainder unless there is excessive back-pressure. Assuming clockwise rotation, this separated body of liquid is moved along the tube and away from the pump as the rotor rotates; just as roller 18 diverges from the track 36 roller 19 makes a fresh compression seal at the bottom to define a new body of separated liquid to be pumped forward.

The present invention permits a reverse, counter-clockwise, mode of movement to be used and unexpectedly finds that a higher pressure grip upon a given dimension tubing can be obtained, thus allowing the pump to be used against a higher back pressure. Thus, for example, pump modules according to the invention can be used with soft blood-compatible tubing in a low-pressure application e.g. renal dialysis, or with more resistant tubing in a different pumping context. The two different flow directions are easily accommodated by switching across the connections at the pumped vessel or apparatus.

While the Applicants do not wish to be limited to any hypothesis as to the mode of action of their invention, the following is believed to be a generally correct explanation of the different pressure characteristics for the embodiment as shown and an explanation of the natural limits of the invention in such a form that the skilled engineer can identify these limits in any given instance.

Fig. 2*a* shows what is believed to be the direction of static load on a system as shown in Fig. 1, including a tube under compression. The resultant force passes through the rotor axis, since else there would be rotor movement. This resultant force can in this embodiment readily be resolved into two components, one in the direction of the spring compression axis and the other along the rotor arm, as shown in the force diagram, since as shown these two components are at right angles.

Fig. 2*b* shows the dynamic load on clockwise rotation. Roller 18 is, as it were, trying to "climb out" of its depression and being resisted by the slope of the tubing wall. The resultant force is as shown, and when resolved as for Fig. 2*a* gives the force diagram at the right of Fig. 2*b*.

Fig. 2*c* similarly shows the dynamic load, direction of resultant force, and force diagram for anticlockwise rotation.

However, the geometry and dimensions of the rotor/track/tubing combination are the same in each case. In particular, the spring is the same length, i.e. subject to the same compressive force. To bring this about, the total force exerted on the tubing in Fig. 2*c* must be appreciably greater than that exerted on the tubing in Fig. 2*b*; in terms of the force diagrams of Figs. 2*b* and 2*c*, the vertical lines must be made the same length, so that the sloping lines (equivalent to actual forces exerted on the tubing) are greatly different.

The above useful relationship, however, only holds for certain combinations of dimensions and angles and basically, for a range of, for example, angles α between the compressive axis of the spring and the line joining the roller and rotor axes. In the following discussion, all angles will be referred to the orientation of the compression spring axis; but this is a mere convenience because the rotor arm and spring are at right angles, and the angle between rotor arm and such line could be used in other cases.

Figs. 3*a*, 3*b*, 3*c*, show force diagrams similar to those of Figs. 2*b* and 2*c* but where the geometry of the system is such that the basic angle α under static load is too low, i.e. where α in Fig. 3*a* is appreciably less than α in Fig. 2*a*. Comparison of Figs. 3*b* and 3*c* will make it clear that a desirable marked diference in length of their vertical lines does not exist, so that the total exerted pressures on the tubing are similar whether clockwise or anticlockwise rotation is used. Thus the pump will operate in either direction, but not to give markedly different pressures.

Fig. 3*b* may slope to the left rather than to the right. However, if this takes place the differential of resultant force has a component away from the pivot of the rotor arm; in other words, the sloping line of Fig. 3*b* may slope to the left rather than to the right. However, if this takes place the diferential of exerted pressures is effectively the same (it begins to fall again very slightly) so that the above conclusions still hold.

Figs. 4*a*, 4*b* and 4*c* show the converse situation to that of Figs. 3*a*, 3*b* and 3*c*, i.e.. where angle α is higher. In this instance the ratio of the vertical components of Fig. 4*b* and Fig. 4*c* is attractively large; but the small *magnitude* of the Fig. 4*c* vertical line means that any small disturbance e.g. vibration, non-uniform tubing, non-uniform wear, buildup of back pressure, altering tubing shape, change of pump speed, could push the force diagram into a negative mode, as discussed above in relation to Fig. 3*c*. There, it was of no account, but in Fig. 4*c* it would trap the roller into a situation where it was pushed into an incompressible double thickness of tubing, or into thicknesses only compressible with great force and associated damage, so that the motor would stall or jam. Of course, if the angular displacement of forces given by anticlockwise motion was too great even without such disturbances the same results would apply anyway.

To sum therefore, there appears to be a useful range of angle α, having on one side embodiments which work in either direction but without useful pressure differentials and on the other side embodiments which cannot be reversed, either inherently, or because they are so unstable that any disturbance will cause them to jam.

Although it is again stressed that Applicants do not wish to be limited to the specific figures given, two simple numerical examples may serve to illustrate the above discussion.

EXAMPLE 1.

Compare three embodiments where $\alpha = 25°$, $45°$ and $65°$ respectively and the displacement of the effective direction of transmitted force is $\pm 30°$. The ratio of the pressure exerted on the tubing in each case, in this simple explanation, is the ratio of the cosines of the respective angles. For the $45° \pm 30°$ embodiment, this ratio is .966 : 259 i.e. 3.73. For the $25° \pm 30°$ embodimnent it is .996 : 574 i.e. 1.74. For the $65° \pm 30°$ embodiment it is a negative figure, .819 :—.087 i.e. —9.41, indicating that the rotor is irreversible in practice since roller 18 will attempt to press into the tube wall material and will thus stall the motor.

EXAMPLE 2.

Compare (a) $65° \pm 20°$ (b) $45° \pm 20°$ and (c) $25° + 20$ as in Example 1

$$(a) = \frac{\cos 85}{\cos 45} = 8.13, \quad (b) = \frac{\cos 65}{\cos 25} = 2.14$$

$$(c) \frac{\cos 45}{\cos 85} = 1.41$$

As in Example 1, the embodiment (b) shows a clear advantage, especially when it is considered that a proportion of the compressive force is used to compress the tube and not to cope with back pressure, thus reducing the differential towards unity anyway. The apparent high value of (a) is only obtained with undesirable instability of operation, and changes as discussed above can force the roller into the tube and stall the motor.

The angle α, between the spring compression axis and the line joining the rotor axis and roller axis depends upon the geometry of the rotor and track, e.g. upon rotor arm length, rotor arm pivot position, roller radius and track radius. These factors, especially roller radius, also affect the displacement of the effective direction force on dynamic load, along with speed, back pressure and compressibility of tubing (dependent in turn on material and inside and outside tube diameter) rate of change of back pressure and compressibility and possibly changes in tube characteristics on wear. Of course, this displacement of effective force is not necessarily the same in the forward and reverse directions, especially when the pressure has built up to desired levels; in simple terms, the "hill" the roller has to climb in one direction is a different shape from the "hill" in the other direction.

Although a single numerical optimum cannot therefore be selected, we believe that preferably α is from 40° to 50°; that the displacement of the line along which pressure is exerted should preferably leave it at +20° to —20° for clockwise rotation and not more than 80° (preferably not more 75°) for anticlockwise travel; and that the ratio of effective maximum delivery pressure should lie between 1.5.1 and 4:1, preferably between 2:1 and 3:1.

The dimensional relationships are also difficult to define, and are generally subservient to those angular relationships above, Operative embodiments will be easiest to construct, however, if the following desiderata are followed:

(i) track radius:roller radius preferably equals 4:1 to 7:1

(ii) roller radius preferably not less than tubing outside diameter more preferably not less than such outside diamter plus twice wall thickness.

(iii) ratio of distance between rotor and arm pivots to distance between arm pivots to distance between arm and roller pivots preferably 1.3:1 to 1.7:1

(iv) ratio of track radius to distance between rotor and arm pivots, preferably 1:1 to 1.1:1.

It must be remembered that when the above explanation is considered the case of a rotor arm with a compression spring at right angles has been selected for simplicity; however, similar explanations can be given for any resilient mounting of the rotor arm in rotation to the rotor block e.g. resilient rubber blocks springs at other than 90° to the arm, or U-springs back towards the pivot.

Figure 5 shows, for the embodiment of Fig. 1 delivery pressure (lb/in^2) against flow rate (ml./min) at 100 r.p.m. Track radius was 43mm, roller radius 8.1mm) roller axis: pivot axis distance 40mm, rotor axis: pivot axis displacements 22.35 and 16.8 (arm direction). In each of the four cases the tubing was in polyvinylchloride material of Shore A hardness 65°. The dimensions of the tubing (bore and wall thickness, mm) flow rate at zero back pressure, pressures at zero flow rate (clockwise and anticlockwise rotation) and ratio of such pressure are as shown in the following Table, intermediate flow rates and pressures being shown in Fig. 5.

	Bore	Wall	ml/min	lb/in²	lb/in²	ratio
A	8.0	1.6	950	18	39	2.2
B	6.4	1.6	615	16.5	47	2.8
C	4.8	1.6	330	20	60	3.0
D	3.2	1.6	165	30	69	2.3

It can be seen that in general qualitative terms, for clockwise rotation the roller trails and the tubing is subject predominately to the action of the spring; in the other the rollers leads and the tubing is subject predominantly to the action of the motor acting along the rotor arm; the two pressures are loosely speaking "with" and "against" the motor.

This leads naturally to a consideration of the different embodiments shown in Fig. 6.

In Fig. 6a rotor 40 compresses tubing 41 against track 42.

Rotor 40 is overall generally lozenge-shaped, and of composite construction in two halves 43 and 44 each having a protruding toothed limb 45, 46 respectively, the two halves 43, 44 being biased apart by springs 47, 48 around guide shafts 49, 50. The extremities of each half rotor each carry an adle roller as at 51 and 52. The limbs 45, 46 mesh with toothed drive wheel 53.

Rollers 51, 52 produce, and rest in, depressions 54, 55 in the tubing 41, pressing the tubing against track 42 which is identical to track 2 in Fig. 1.

In anticlockwise rotation of the drive wheel and limbs 45, 46 jointly press the rollers 51, 52 into the tubing; in the other direction only the spring pressure, minus the drive force, exerts this effect. Thus, although a different mechanical construction is used, the effect is to give different operating pressures in different directions, e.g. as given by Fig. 1.

Figs. 1 and 6 both show the pump module only. In practice this is mounted on a horizontal drive spindle protruding from the front face of a drive module. This drive module is essentially a low—H.P. electric motor in a casing provided with on-off, speed-change, and direction change control. Optional extras, either incorporated therein or in a further control module are revolution counters, dosage period times and dosage interval times. Of course, if desired, more than one successive module can be mounted on the same drive shaft.

CLAIMS

1. A pump module for a peristaltic pump of the type wherein at least one operating member passes within a clearance of an arcuate track such as to squeeze together the walls of flexible tubing located along the track and thereby exert a pumpi[illegible] action; wherein the operating member comprises a roller resiliently mounted in relation to a driven rot[illegible] block, the dimensions and method of mounting being such that a significantly different resultant pressure is exerted on the tubing during clockwise rotation from that exerted during anticlockwise rotation.

2. A pump module according to claim 1 in which the roller is idly mounted at or near the end of a rotor arm, which is pivottally attached at or near its other end to the driven rotor block.

3. A pump module according to claim 2 in which there are two of said roller-rotor arms carrying operating members spaced at 180° round the driven rotor, the rotor arms being straight and parallel to one face of the driven rotor block.

4. A pump module according to any of claims 1 to 3 in which the resilient mounting comprises one or more compression springs placed between each arm and the driven rotor block, so as to biasing said rotor arms away from the driven rotor.

5. A pump module according to claim 4 in which the biasing means comprises two compression springs placed between each arm and the driven rotor block such that the spring compression axis is at right angles to the rotor arm.

6. A pump module according to any of the claims 1 to 5 in which the arcuate track is semi-circular.

7. A pump module as claimed in any of claims 3, 5 and 6 in which the angle between the spring compression axis and the line joining the rotor axis and roller axis is between 40 to 50°.

8. A pump module according to claim 7 in which the ratio of track radius to roller radius is between 4:1 and 7:1, the ration of the distance between rotor block axis and rotor arm pivot to the distance between rotor arm pivot and roller axis is between 1.3:1 and 1.7:1 and the ratio of track radius to the distance between driven rotor block axis and rotor arm pivot is between 1:1 and 1.1:1.

9. A pump module for a peristaltic pump comprising a part-circular track; a composite rotor body mounted coaxially therewith; and an idly mounted roller at each end of said rotor body to bear against,

and compress together the walls of a length of tubing located against said track; wherein the rotor body is formed as two like portions each possessing a straight limb extending towards the other portion symmetrically with respect to the axis of rotation, the facing edges of each limb being toothed and the two portions being resiliently biased away from each other; whereby locating the teeth on said limbs around a toothed drive wheel provides in one direction of rotation thereof the additive effect of the biasing pressure plus the dynamic motor load to compress the tubing, but in the other direction of rotation only the effect of the biasing pressure minus the dynamic motor load.

10. A pump module as claimed in any of claims 1 to 9 mounted on a horizontal drive spindle of a drive module; said drive module containing a drive means capable of operating at variable speed and capable of imparting both clockwise and anticlockwise rotation to the driven rotor.

11. A peristaltic pump comprising a module as claimed in any of claims 1 to 10.

12. A peristaltic pump comprising at least two pump modules as claimed in any of claims 1 to 10 mounted on a common horizontal drive spindle of a drive module so as to provide multi-channel pumping capacity.

13. A rotor assembly comprising a rotor block with a central bore for mounting on the drive spindle of a drive module; at least two rotor arms pivotally attached at or near their ends to the rotor block; an idle roller axially mounted at or near other end of each rotor arm; and resilient biasing means to bias each rotor arm away from the rotor block.

14. A rotor assembly comprising a rotor block with a central bore for mounting on the drive spindle of a drive module having at least one pair of outwardly directed mounting lugs and at least one rotor arm being similarly provided with mounting lugs at or near each end, one set of mounting lugs being mounted on a common spindle with the lugs of the rotor block, the other set of mounting lugs on the rotor arm carrying a spindle mounting an idle roller; said rotor arm also having a pair of bores to accommodate a freely moving pair of pins having rotary guides for the tubing on their free ends and surrounded by compression springs to provide a resilient biasing means to bias the rotor arm away from the rotor block.

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