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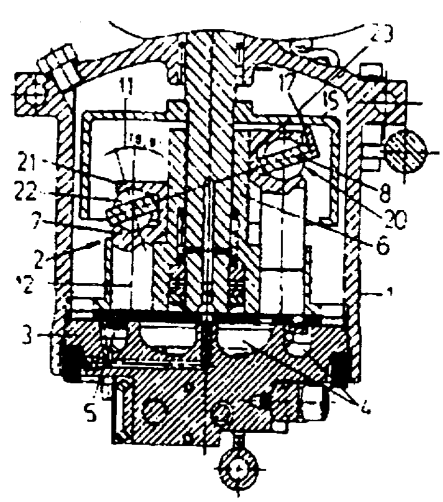
(54) Abstract Title
Swash plate compressor

(57) The invention relates to a compressor especially for an automobile air conditioning system, containing a compressor unit (2) for suction and compression of a cooling medium arranged inside of the housing (1). Said compressor unit (2) has a piston (7) which operates inside a cylinder block (6) and a driving plate (8), i.e. a swash or pivot plate, which actuates said piston (7). In order to increase efficiency, the invention is characterized in that the driving plate (8) is mounted in such a way that the center line (9) of mounting, i.e. swash or mounting axis, forms a tangential line to the reference circle (10) defined by the stroke so that the angle of inclination (11) of the driving plate (8) can be changed without displacing the dead center above the stroke position. The compressor providing increased efficiency is also characterized in that the driving plate is mounted in such a way that the center line of mounting, i.e. swash or mounting axis, forms a tangential line to the reference circle defined by the stroke so that the angle of inclination of the driving plate can be changed without displacing the dead center above the stroke. In order to increase efficiency, the compressor is constructed in such a way that the components which come in contact with the cooling medium, preferably the walls configured between the intake and outlet areas along the flow path, are slightly thermally shielded from the cooling medium in the area of the flow path. In addition, the compressor is constructed in such a way that the belt is protected by means of uncoupling of the coupling assembly when a defined thermal and/or mechanical capacity limit is exceeded during blocking of the drive shaft or compression unit.



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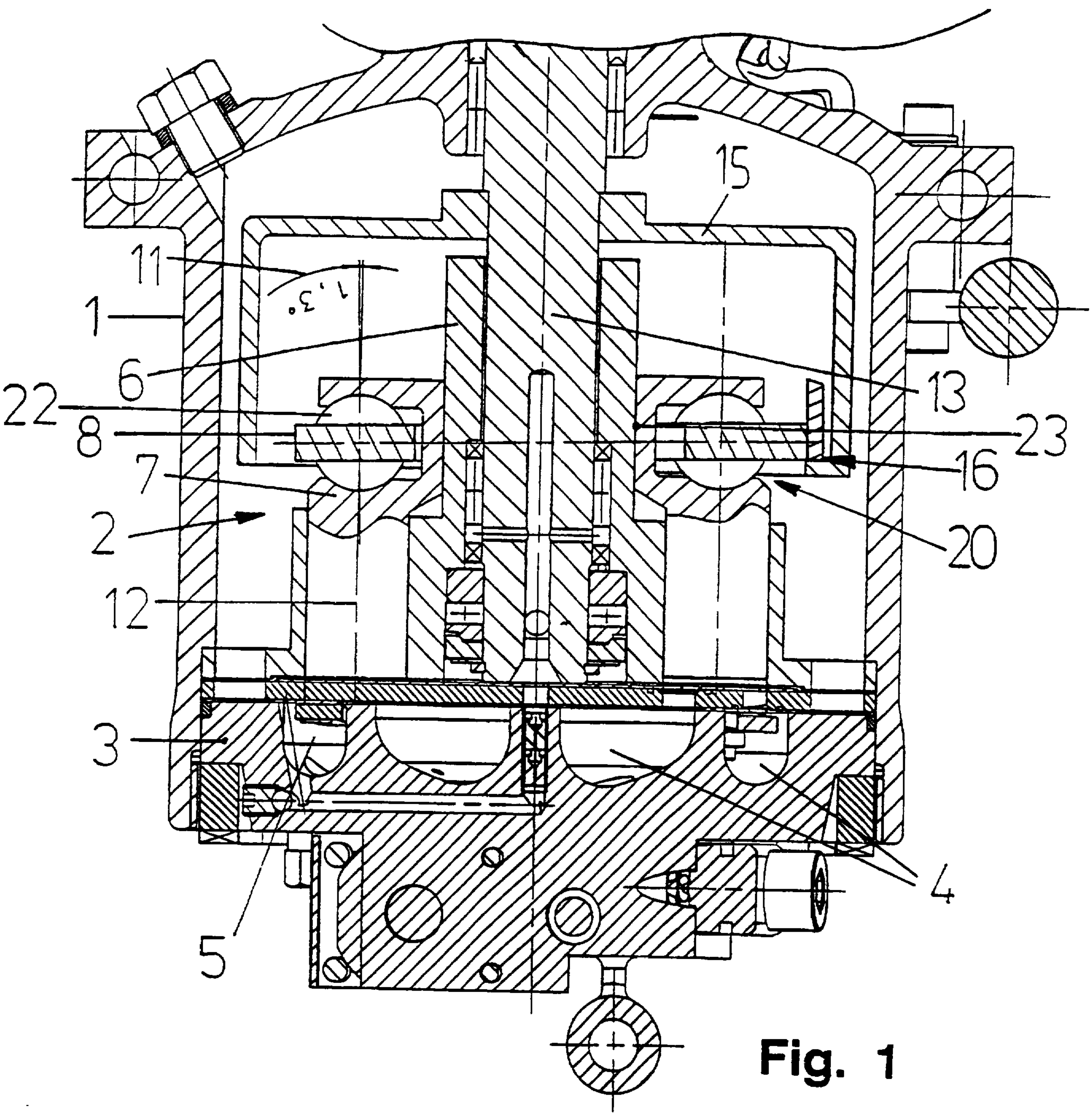

Fig. 1

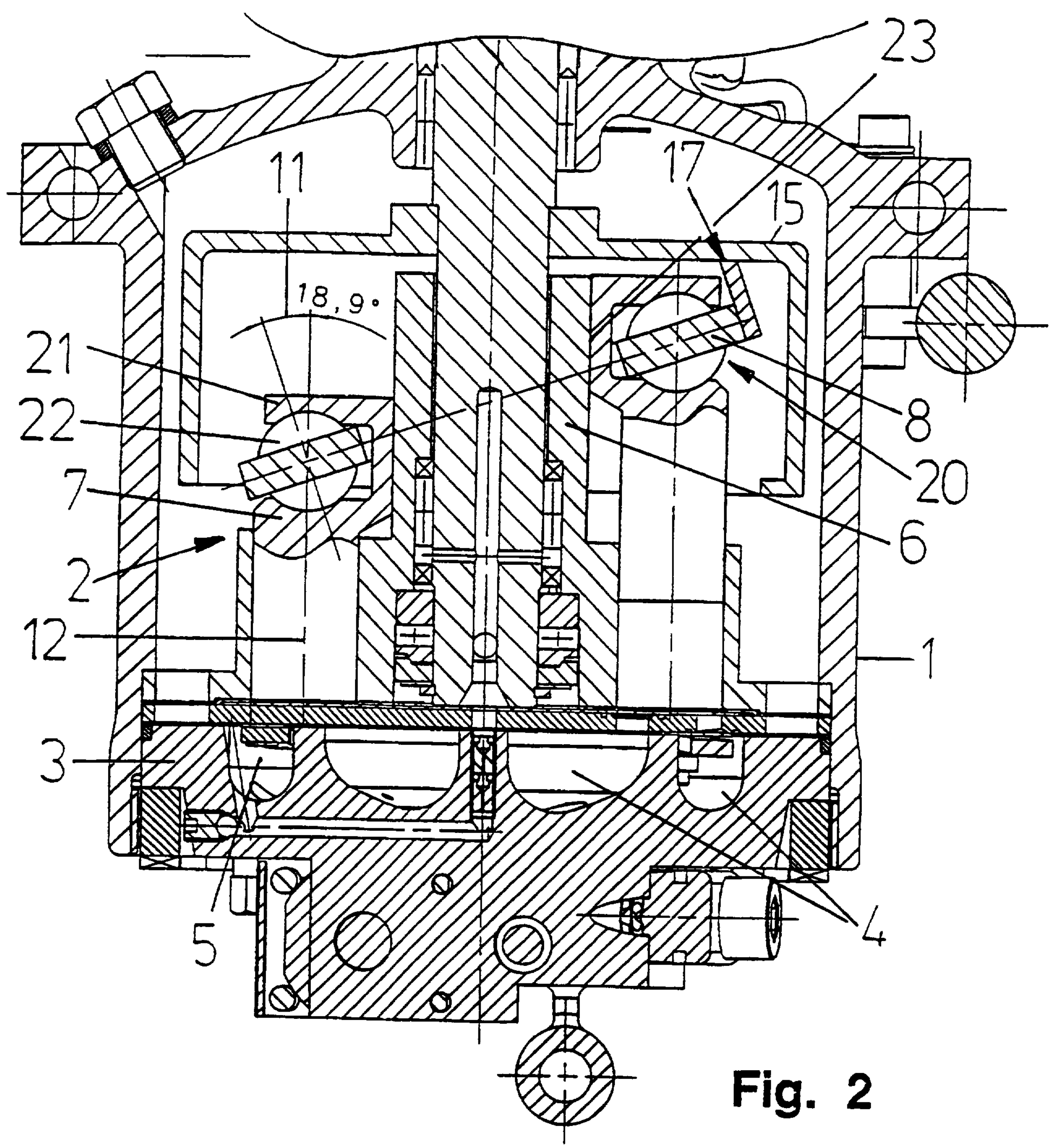

Fig. 2

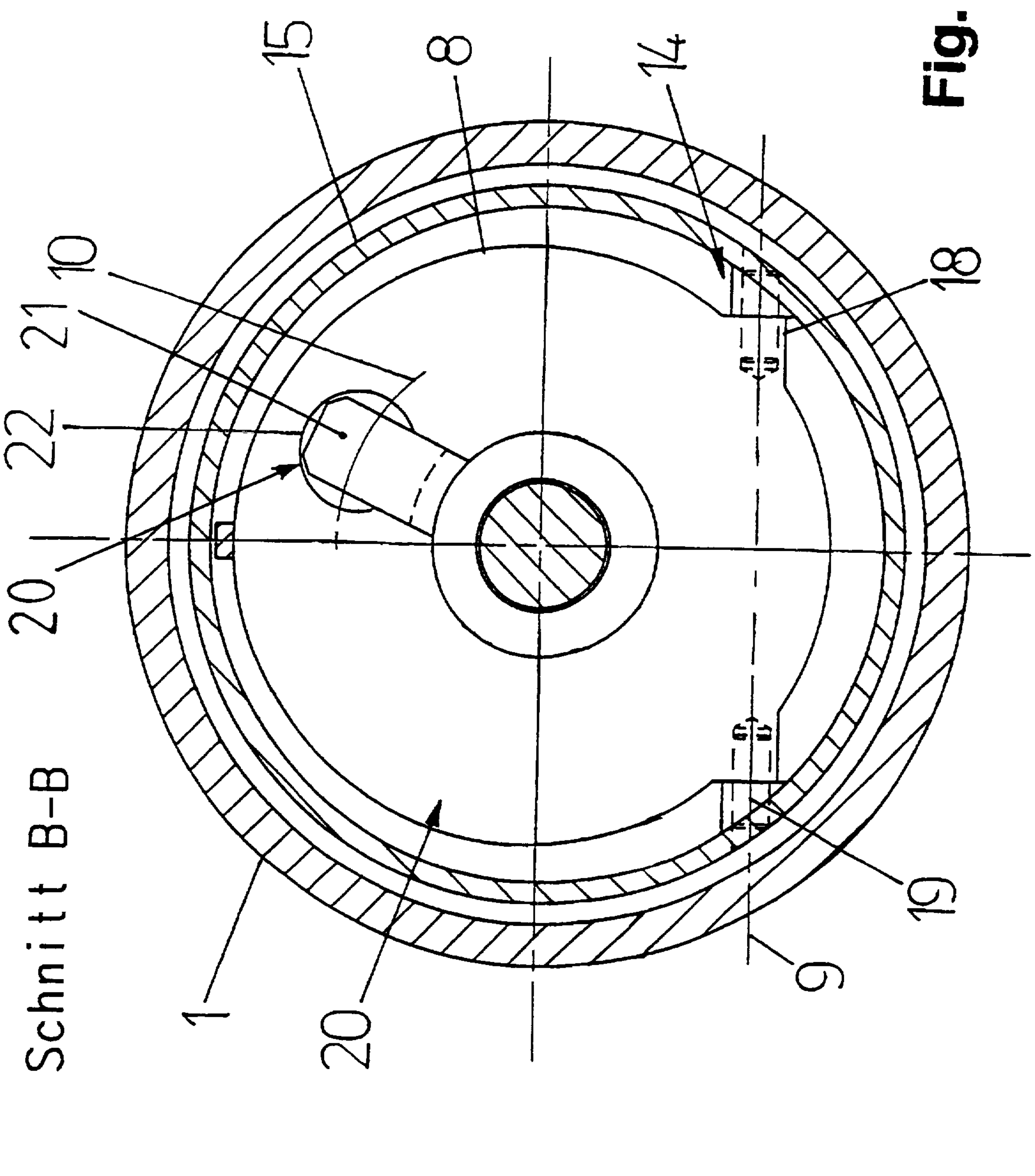

Fig. 3

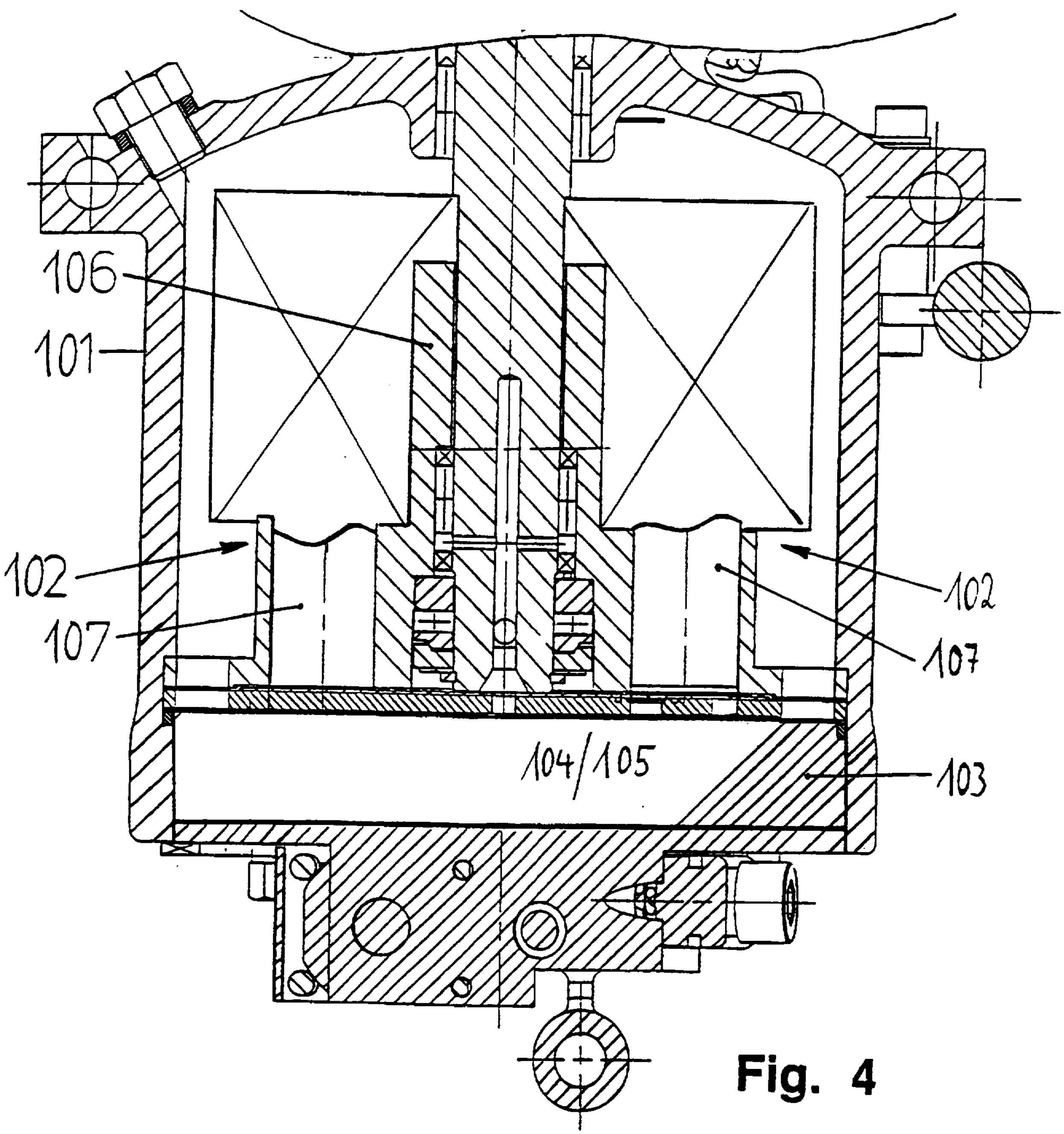

Fig. 4

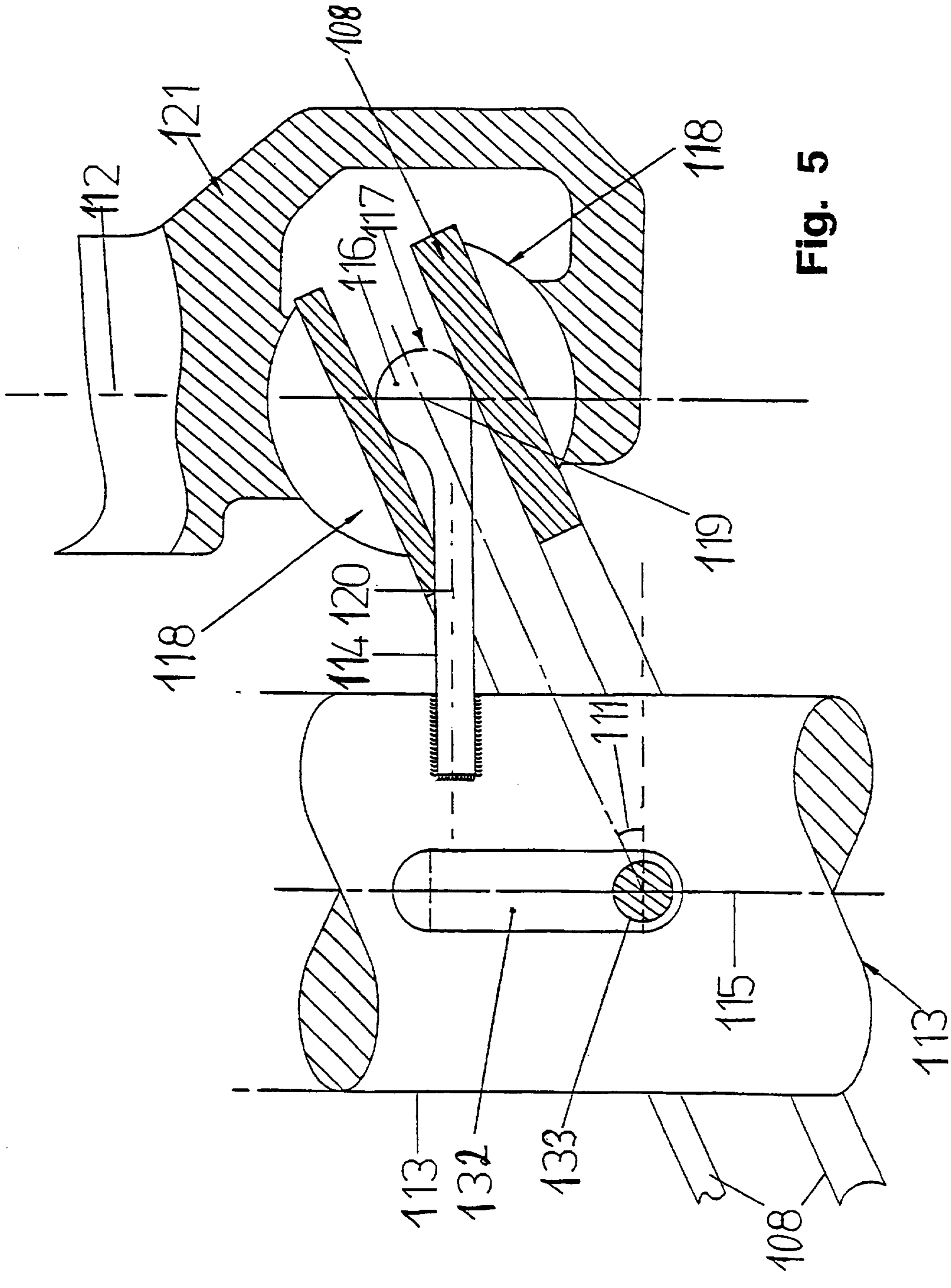

Fig. 5

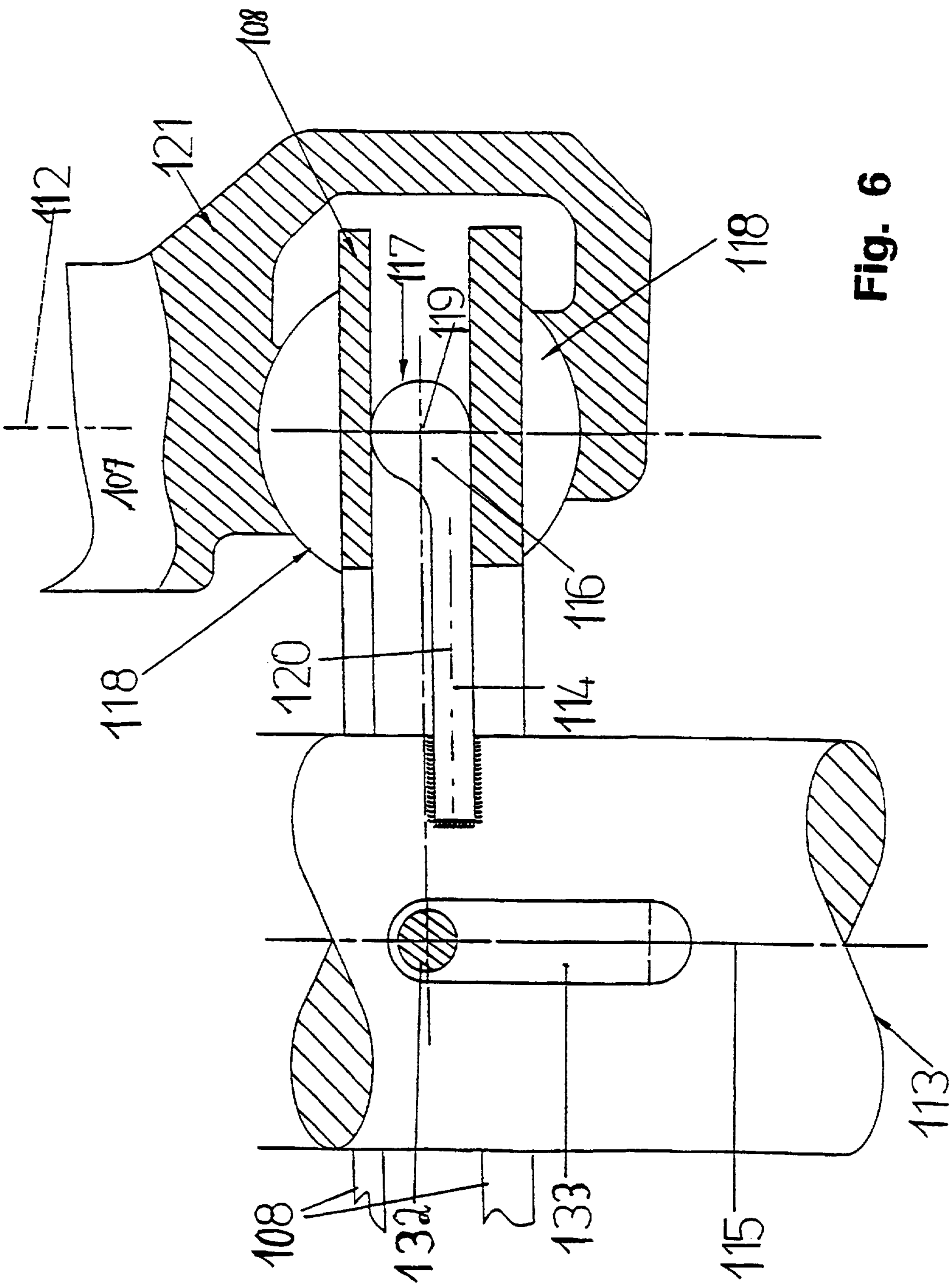

Fig. 6

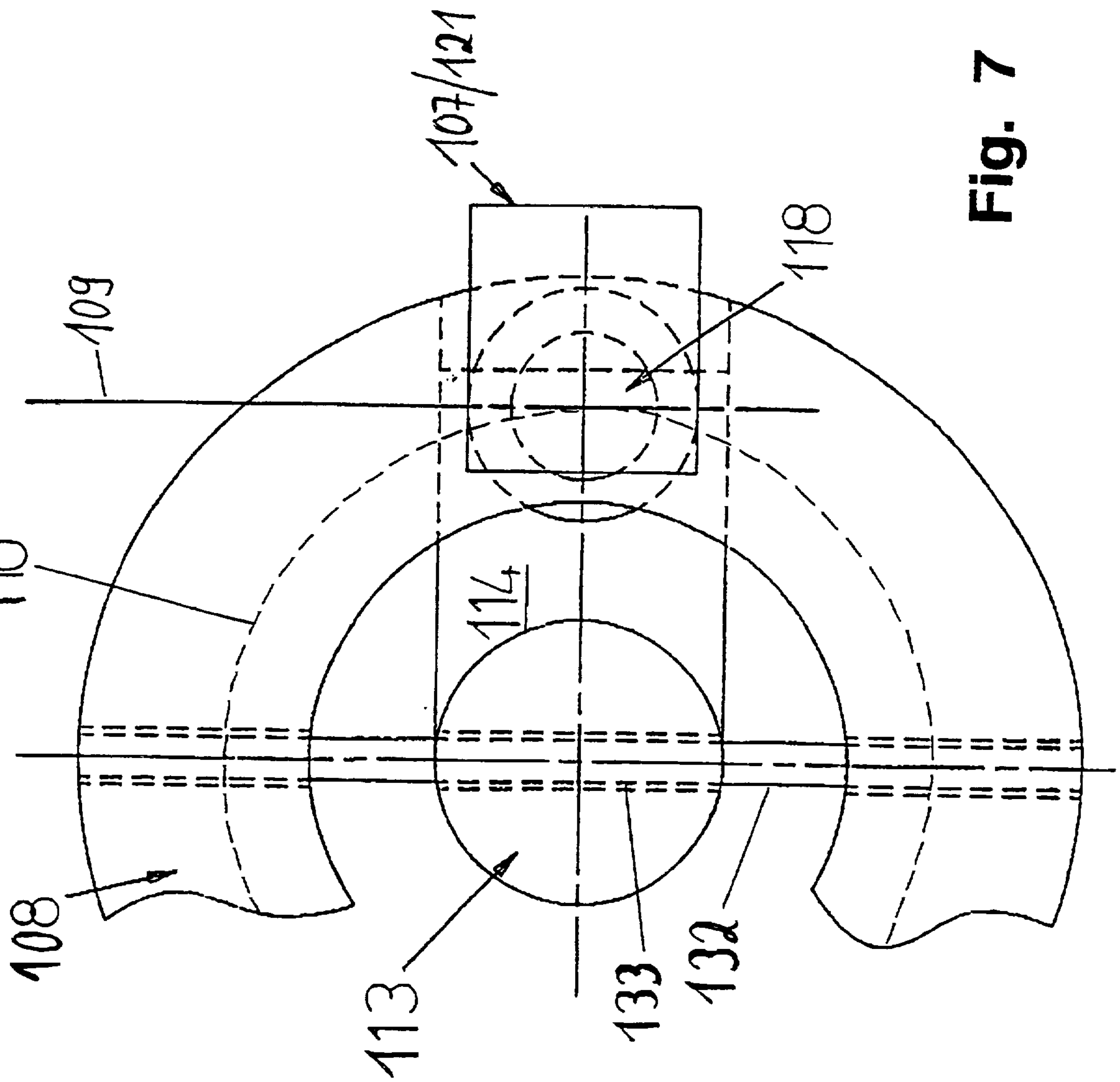

Fig. 7

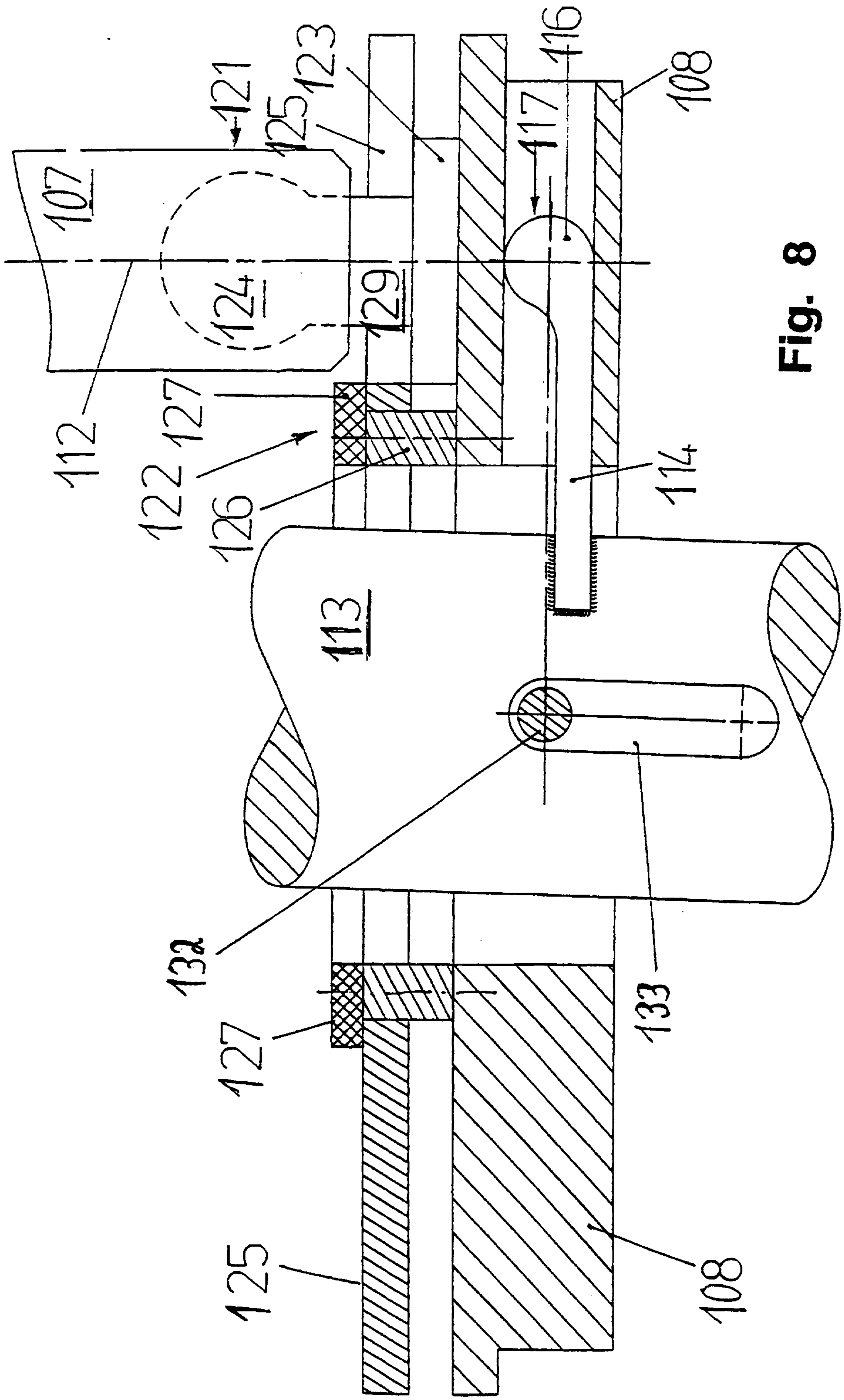
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Fig. 8

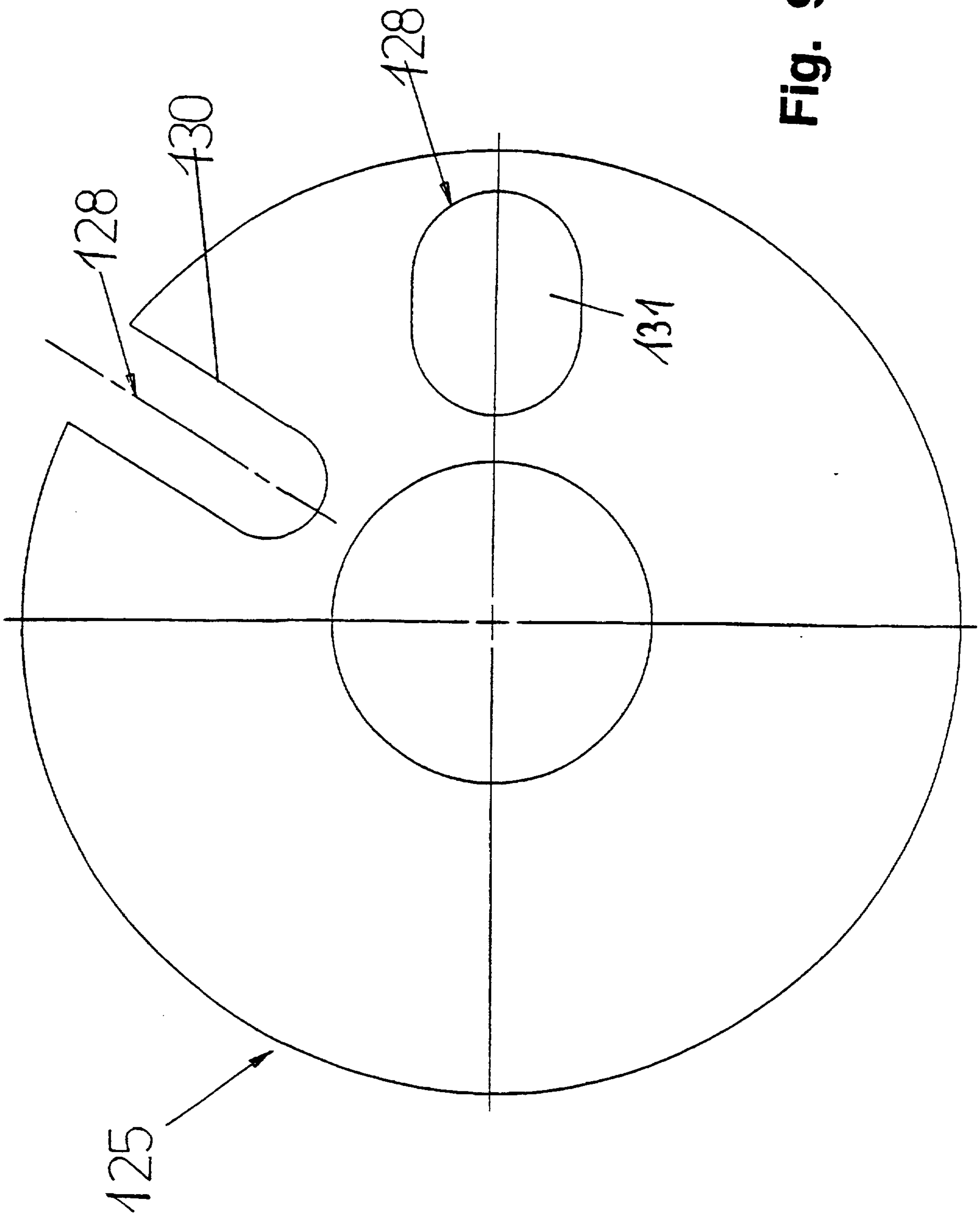

Fig. 9

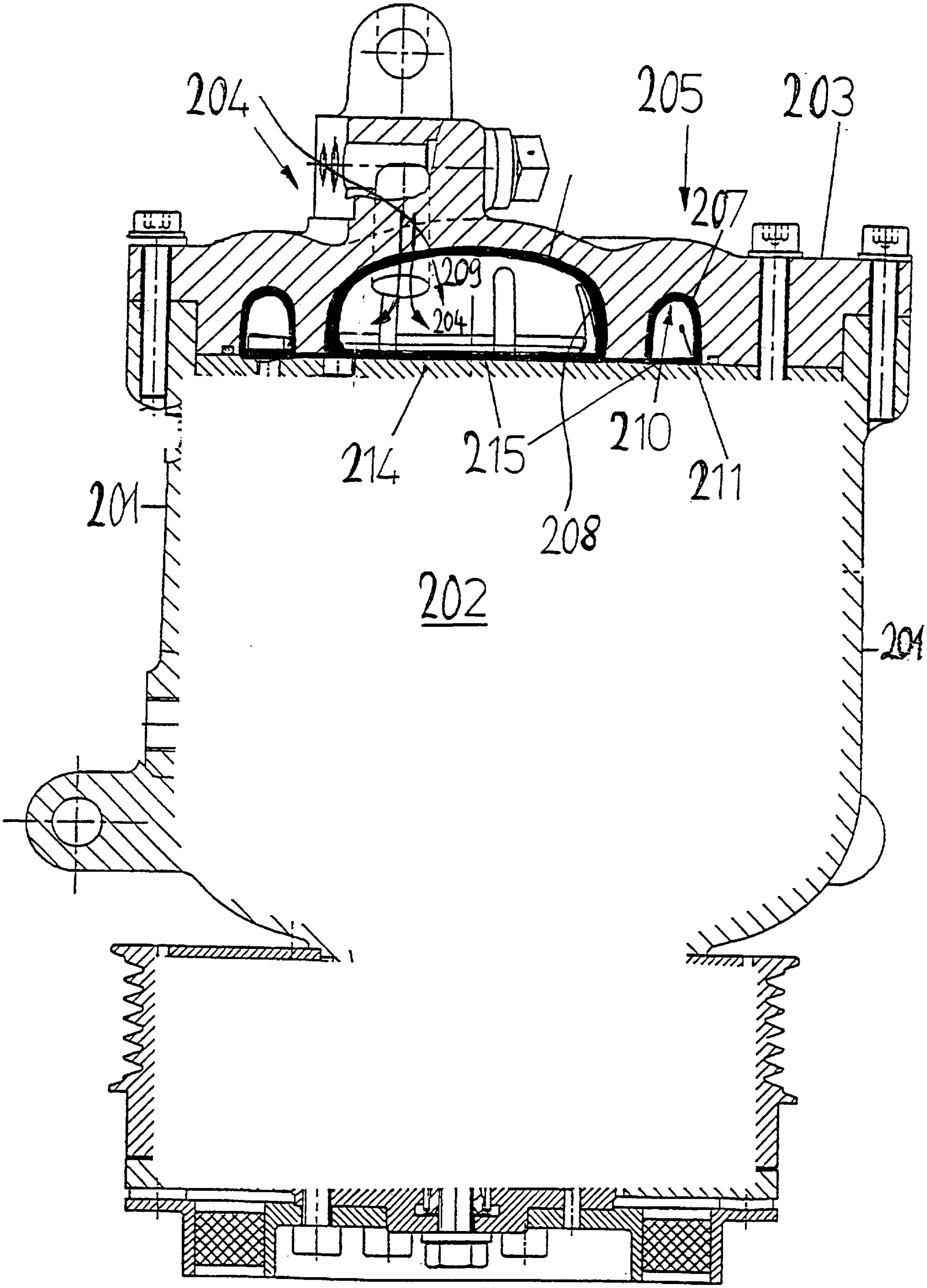

Fig. 10

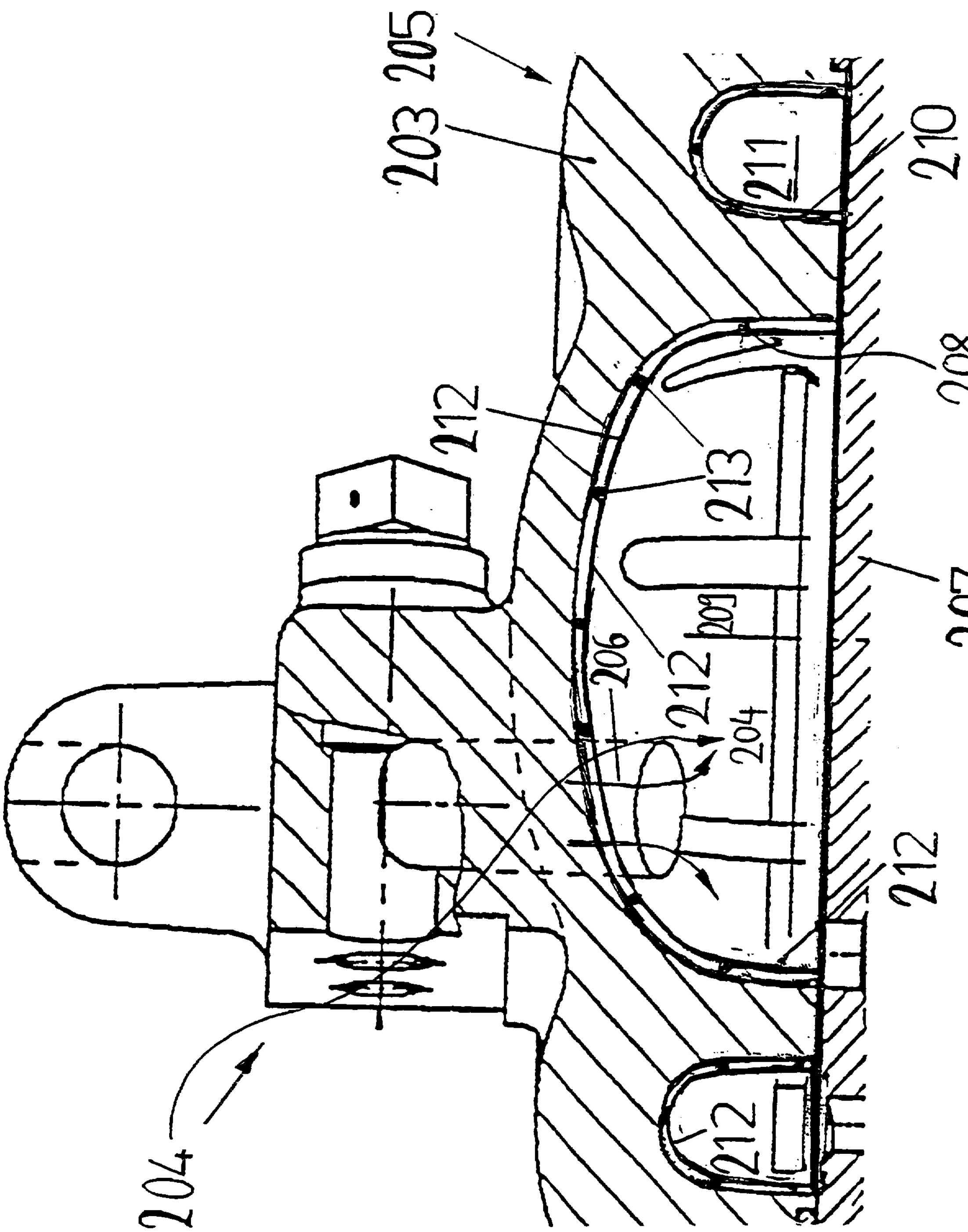

Fig. 11

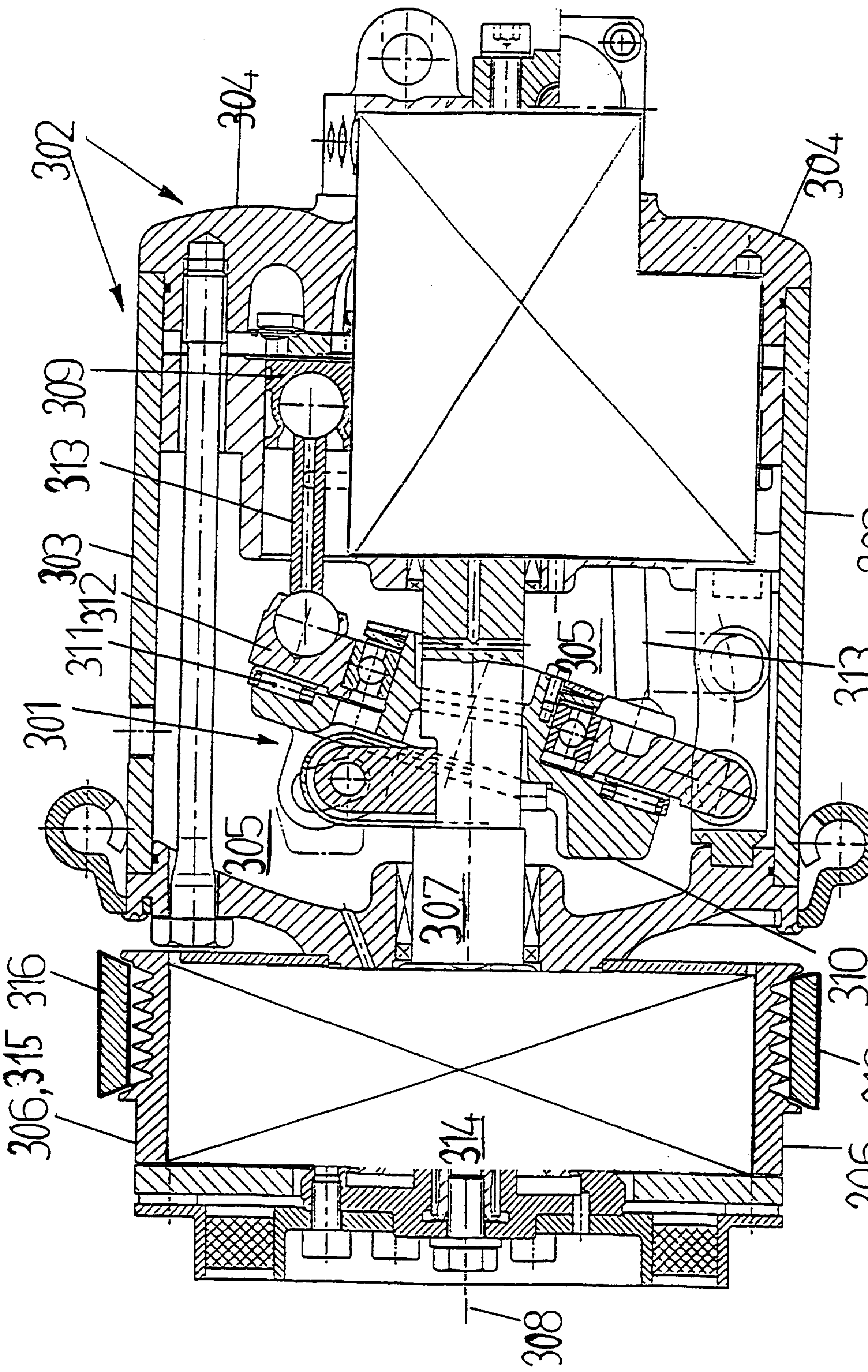
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Fig. 12

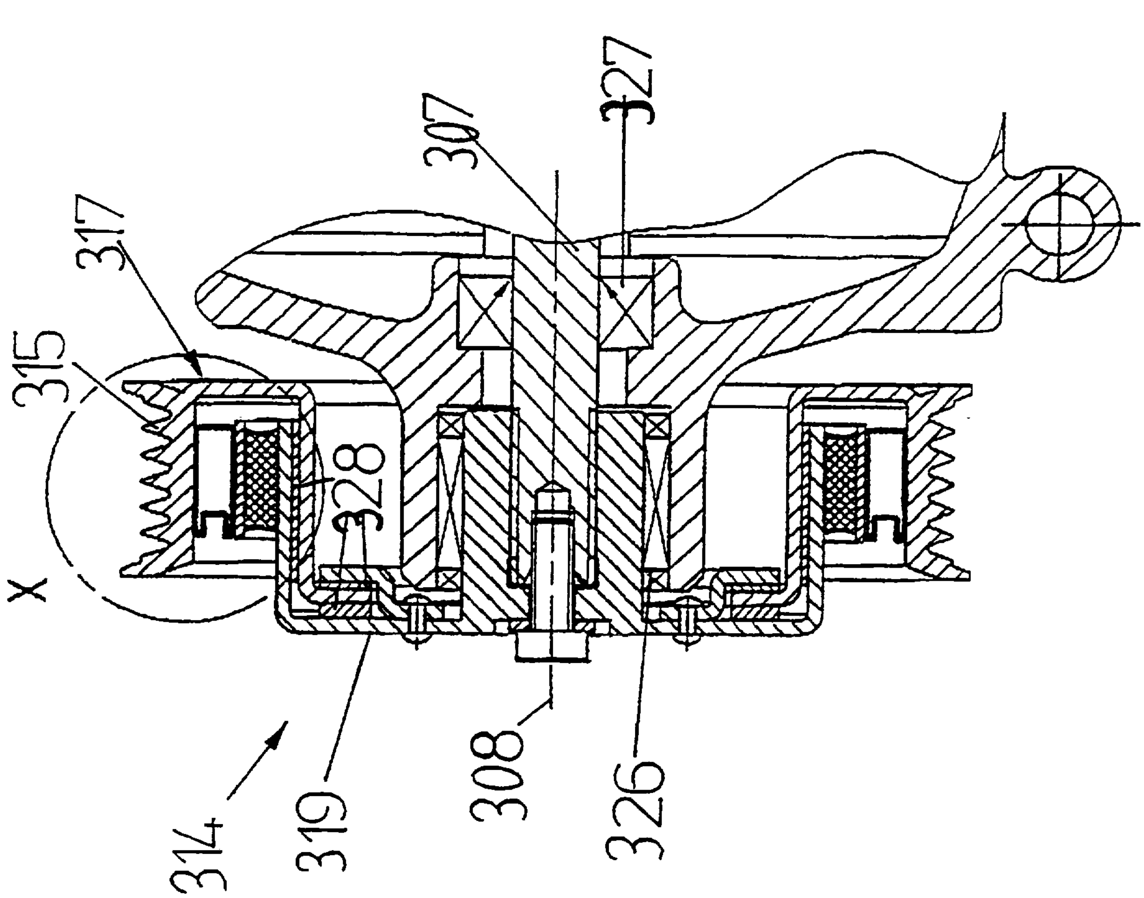

Fig. 13

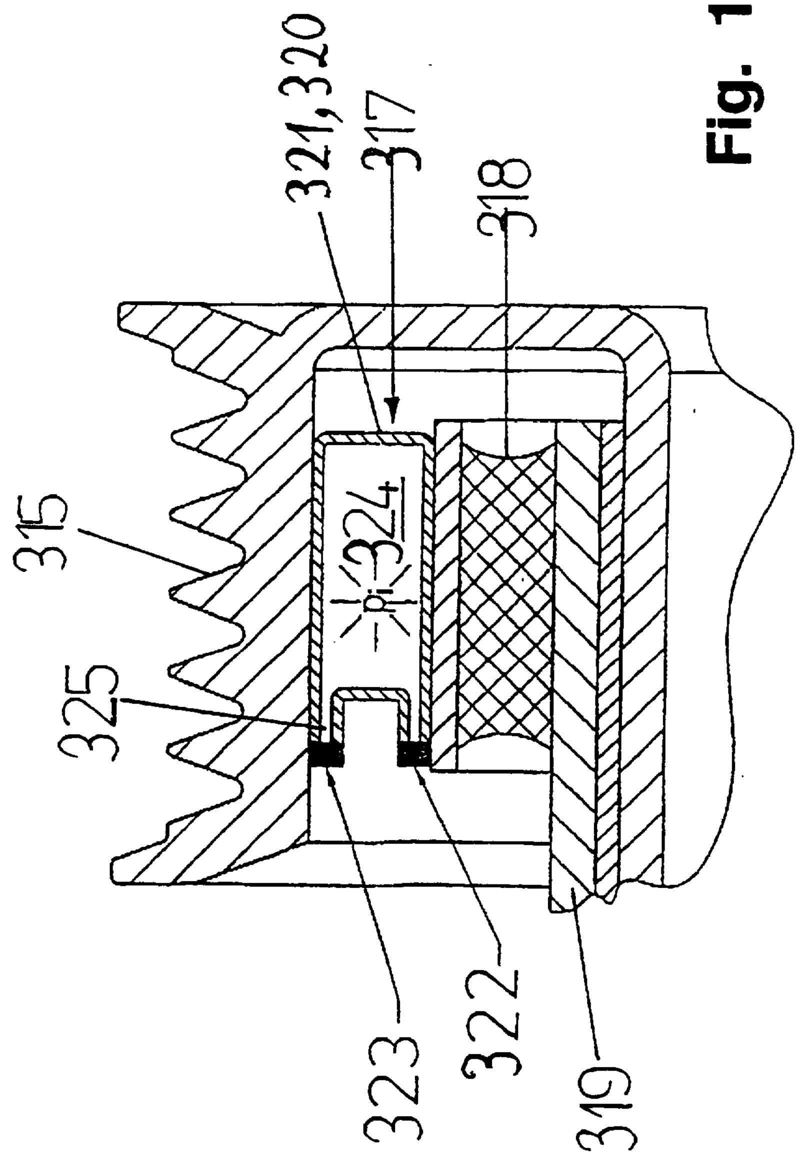

Fig. 14

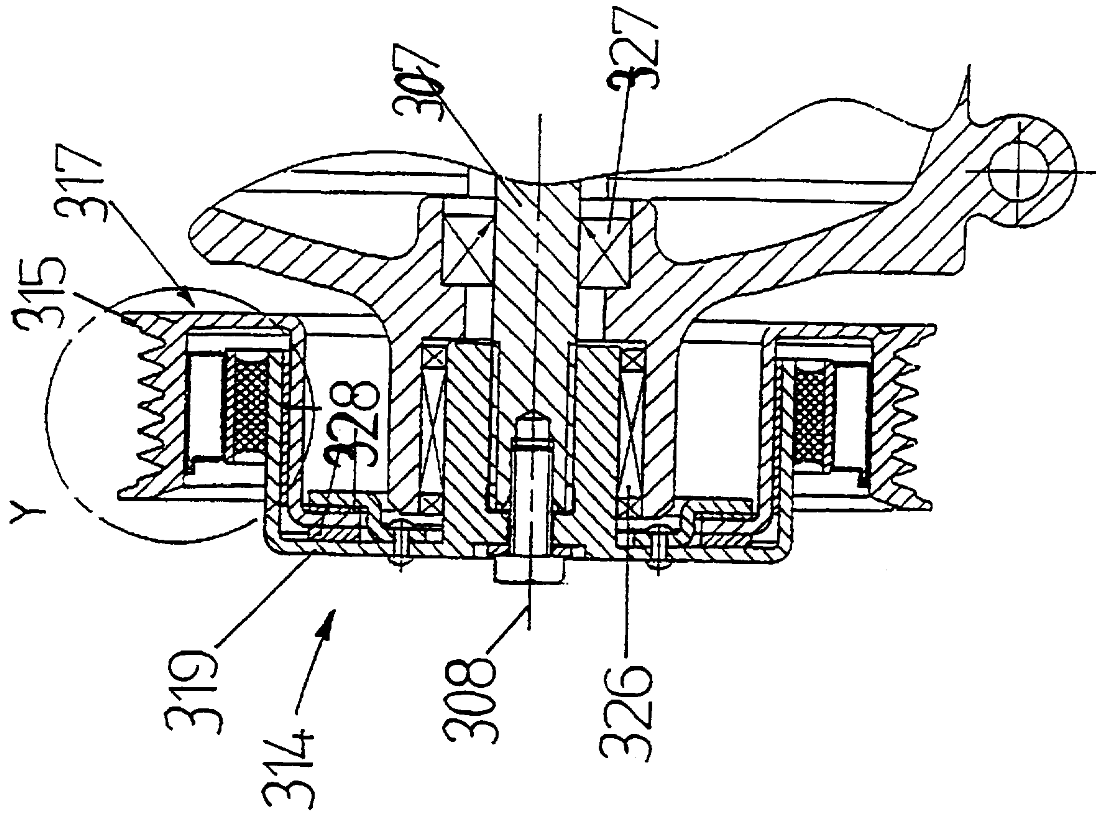

Fig. 15

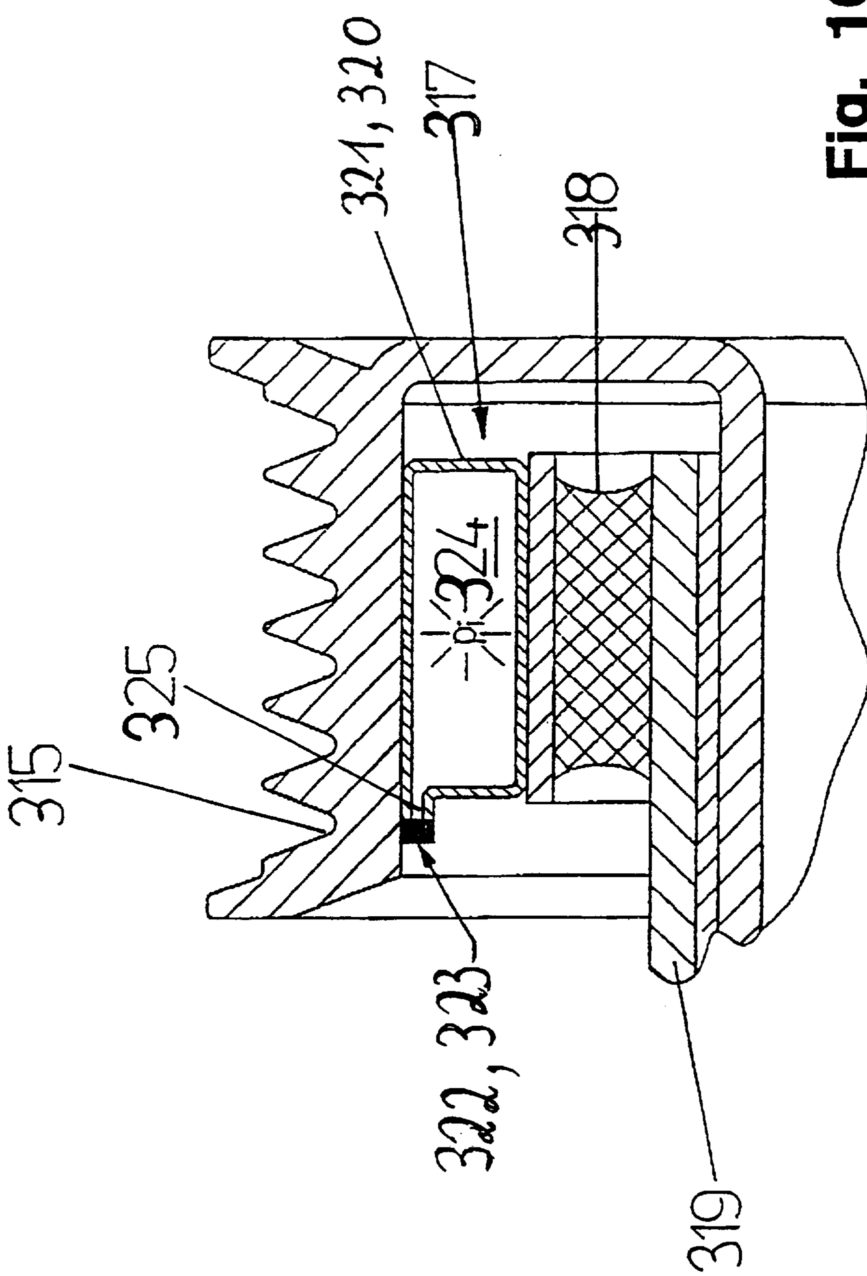

Fig. 16

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SWASH PLATE COMPRESSOR

abutment serves to absorb the torque that is transmitted from the rotating swash plate to the receiving disk.

Furthermore, in the case of the known compressors of the kind under discussion, it is essential that the angle of inclination of the swash plate or pivot disk, hereafter drive disk, can be varied by means of a special coupling mechanism, namely for reciprocating the pistons. However, with respect to the longitudinal axis of the piston, the drive disk tilts due to its geometric arrangement or due to the pivot axis there in such a manner that the dead center "travels" above the stroke position. This results in a tilting error or a damage area, which has a negative effect on the efficiency of the compressor.

Also known from practice are beginnings of reducing the damage area, namely in terms of constructing extremely expensive solutions with internal coupling mechanisms. To this end, reference may be made, for example, to DE 35 45 200 C2. Despite the constructional expenditure incurred therein, it has so far not been possible to eliminate the previously discussed problems, so that a resultant loss in efficiency is accepted in the case of the so-far known compressors of the kind under discussion.

Furthermore, the invention relates to a compressor, in particular for an air conditioning system of an automobile, with a housing and a compressor unit accommodated in the housing for taking in and compressing a refrigerant, the refrigerant flowing from an intake area preferably formed in a front-end housing cover through the compressor unit into a discharge area preferably likewise formed in the housing cover.

Air-conditioning compressors of a large variety of types operate with a refrigerant. Besides conventional refrigerants, whose use appears to become more and more critical in the light of an increasing awareness of the environment, it is possible to use as a refrigerant an inert gas, such as, for example, CO_2, which is noncritical under environmental aspects. However, the use of such a refrigerant leads to higher pressures within the compressor, thereby necessitating quite special constructional measures, for example, with respect to the selection of material and wall thickness of the housing.

The use of a high-strength material for the housing of the compressor makes it easy to absorb as early as in the intake state the high pressures necessary or occurring in the case of a refrigerant having a high density. For example, it is thus necessary to withstand bursting pressures of up to 30 MPa at discharge temperatures ranging from 160°C to 170°C.

As aforesaid, compressors of the kind under discussion comprise an intake area and a discharge area. Whereas on the suction side -- in the intake area -- the refrigerant flows in at a temperature mostly ranging from 30°C to 40°, the temperatures on the pressure side, i.e., discharge area, are in a range from 80°C to about 170°C.

Normally, the compressor housings are made of metal, for example, aluminum, of high-quality steel, or of a high-tensile steel. Consequently, the high temperature in the discharge area is bound to become effective on the intake area to the extent that same is heated via the housing material coming into contact with the refrigerant, as well as the "interiors" of the compressor. As a result, the gaseous refrigerant is heated on the intake side, whereby its density decreases. This again leads to a loss in delivery or a reduction of the mass flow of the refrigerant and, thus, to a loss in output of the compressor. Because of the temperature influence by the discharge area on the intake area, the efficiency of a conventional compressor is quite considerably reduced.

Likewise, the invention relates to a compressor, in particular for the air conditioning system of an automobile, with a housing and compressor unit arranged in the housing for taking in and compressing a refrigerant, wherein a belt drives the compressor unit via a drive shaft and a drive wheel coupled with the drive shaft, and wherein the drive wheel comprises a belt pulley body engaging the belt, the belt pulley body being coupled via a coupling device directly or indirectly with the drive shaft.

The compressors under discussion are driven via a belt, which is guided over a belt pulley hereafter drive wheel. The belt in turn is driven via the crank shaft of the internal combustion engine of an automobile.

Malfunctions may occur in the operation of the compressor. Thus, for example, the compressor unit or the drive shaft may block. If the belt loops about the drive wheel at a very small angle, the belt is expected to slip on the drive wheel or the belt pulley. In this instance, the drive wheel will heat up very considerably. This leads already after a short time to damage and finally to the destruction of the belt, so that even the subassemblies that are also driven by the belt, such as, for example, the water pump or alternator, can no longer be operated. As a result, the automobile is no longer operable.

If the belt loops about the drive wheel or belt pulley at a larger angle, for example, more than 180°, it is hardly possible that the belt slips on the drive wheel or belt pulley. This leads either to a tearing of the belt or to a choking of the engine. Likewise, in such an instance, the automobile is no longer operable.

To avoid the above-described problems, an electromagnetic clutch has already been integrated in the drive wheel of the compressor. If the belt slips or the clutch halves slip, the clutch will undergo a very considerable heating. If a predetermined temperature is reached, a safety fuse will interrupt the coil current, and the clutch disengages the compressor, so that the belt can continue to move with the belt pulley body of the drive wheel. This ensures the operation of safety-relevant components of the automobile, such as, for example, the water pump and/or the alternator, which are likewise driven by this belt.

However, the electromagnetic clutch as known from practice, is problematic, inasmuch as it is constructed relatively large, expensive with respect to its individual components, and thus represents a quite considerable cost factor. Primarily, due to its complexity, such an electromagnetic clutch causes an extraordinary weight load, which is diametrically opposite to a weight reduction as is constantly sought in the current automobile construction. Because of its enormous overall size, the compressor is not suitable for installation in small engine compartments.

From practice alone, it is already known to provide an overload clutch with a disk-shaped rubber body having an external gear tooth system, whose gear teeth shear in the event of excessive stress. In this case, the clutch is a purely mechanical overload clutch, whose disengagement behavior can be defined only in a certain bandwidth. At any rate, such an overload clutch is barely reliable.

It is the object of the present invention to improve and further develop a compressor of the initially described kind such that its efficiency becomes more favorable in comparison with conventional compressors, that the compressor is constructed smaller, that its weight is reduced, and that accordingly it is easier and less expensive to manufacture the compressor. Furthermore, it is an object to increase output and to ensure at a lesser expense at least the same safety, in particular a protection of the belt drive and the internal combustion engine, as in the case of compressors known until now.

The compressor of the present invention accomplishes the foregoing object by the characteristic features of claim 1. Accordingly, a compressor of the initially described kind is characterized in that the drive disk is supported such that the center line of the bearing mount (pivot or bearing axis) forms a tangent to a reference circle that defines the stroke, so that the angle of inclination of the drive disk is variable without displacing the upper dead center above the stroke position.

To begin with, it has been recognized by the invention that is possible to increase the efficiency of a compressor of the described kind by optimizing the damage area. It has further been recognized that an optimization of the damage area is possible by realizing a special bearing mount of the drive disks, i.e., the swash plate or pivot disk. According to the invention, the drive disk is mounted such that the center line of the bearing mount, i.e. the pivot or bearing axis of the drive disk forms a tangent to the reference circle that defines the stroke, so that the angle of inclination of the drive disk is variable, without displacing the upper dead center above the stroke. In other words, the pivot or bearing axis of the drive disk is positioned such that the drive disk tilts exactly above the longitudinal axis of the piston. This precludes the dead center from changing in the stroke position of the piston, or from shifting. Instead, it is held constant above the stroke position.

Finally, the measure of the present invention accomplishes that the instantaneous pole lies on the diameter of the reference circle. Accordingly, the instantaneous pole is stationary, when related to the rotating reference system in the case of the drive shaft. In this instance, it is easily possible to realize pivotal movements of the drive disk up to about 20°.

Concretely, the drive disk is operatively connected to the drive shaft via a coupling mechanism associated to its edge region. To this end, it is possible to joint the drive disk, via the coupling mechanism, to an entraining member that is nonrotatably connected to the drive shaft, so as to be able to perform there the pivotal movement. In a further advantageous manner, the entraining member surrounds the drive disk at least in part. In this instance, the entraining member may be made annular, preferably cylindrical. Likewise, it is possible to design and construct the entraining member as a ring segment. At any rate, the entraining member may be both a separate component and an integral component of the drive shaft.

In a further advantageous manner, the entraining body forms a stop for the pivotal movement of the drive disk. In this connection, the stop may be both for a maximal and for a minimal pivotal movement of the drive disk. Accordingly, during the movement, the drive disk or a there-provided shoulder or the like comes into contact within the entraining member. To this end corresponding contact surfaces, steps, cants, or the like are provided.

The drive disk is supported via a pivot bearing operative between the drive disk and the entraining member. This pivot bearing may be designed in many different ways. For example, the pivot bearing may comprise spherical segments or bearing pins that are operative between the drive disk and the entraining member.

According to the foregoing description, the coupling mechanism is associated to the edge region of the drive disk. In this arrangement, it is of further advantage that the here concretely selected pivot bearing engages the entraining member according to its width in such a positioned manner that the drive disk tilts exactly above the longitudinal axis of the piston.

As likewise described before, the drive disk serves to move the pistons. To this end, coupling means are operative between the drive disk and the pistons, which comprise an embracing element associated to the piston and preferably operative in the fashion of a slideway, and a spherical body associated to the drive disk and preferably operative in the fashion of a slide shoe. In other words, the coupling means are a jointed arrangement, which permits a movement of the pistons along their longitudinal axis free of deviations, wherein the drive -- via the drive disk -- is operative at constantly changing angles.

As regards a concrete development of the coupling means, it is very important that the embracing element of the piston extend through the drive disk on the side facing the drive shaft, and enclose it outwardly at least in part, preferably with the inclusion of the slide shoe or spherical body. Consequently, in this instance, an encircling engagement occurs from the inside, whereas in the previously known compressors of the kind under discussion, an encircling engagement is realized exclusively from the outside. With a corresponding dimensioning of the cylinder block and the pistons, it will be possible to reduce the overall size, at any rate, however, the diameter of a semicircle.

Furthermore, it will be of advantage, when the cylinder block comprises an antirotation device for the pistons. The antirotation device may be realized in many different ways. Thus, it would be possible to provide a support surface between the cylinder block and piston, so that insofar the piston is prevented from rotating. Likewise, it would be possible to realize the antirotation device as a support surface between the piston and drive shaft. Likewise possible is a support surface between the piston and entraining member. As an alternative, it would be possible to make the pistons "noncircular," so that an antirotational protection is realized with a corresponding configuration of the cylinder block. Finally, it would be possible to associate to the piston a guide element as an antirotation device, which may be a pin or the like.

It is also possible to modify the drive disk itself, namely in that it operates in the fashion of a centrifugal governor as is sufficiently known from practice.

As regards a further, concrete realization of the teaching according to the invention, it will be of advantage, when the drive disk is operatively connected to the drive shaft via a guide arm that is rigidly joined to the drive shaft and supported for sliding movement in the drive disk, preferably in its edge region. In addition to absorbing axial forces, the guide arm could also transmit the torque of the drive shaft. At any rate, the guide arm is rigidly connected to the drive shaft, with the sliding mount of the guide arm in the edge region of the drive disk enabling a tilting of the drive disk.

In particular with respect to a stable construction of the guide arm, same could be designed and constructed as a square bar, and in a further advantageous manner, it could be mounted in the drive shaft orthogonally to its longitudinal axis. In this connection, it is possible that the guide arm is pressed into a corresponding recess of the drive shaft. The rigid arrangement of the guide arm on or in the drive shaft is ensured by frictional engagement.

For a sliding support of the guide arm in the drive disk, the free end of the guide arm mounts a guide member, with which the guide arm engages the drive disk. The guide member could be made substantially cylindrical crosswise to the longitudinal axis of the guide arm, with the guide member being directly or indirectly supported in the drive disk.

Thus, it would be possible to use the guide member with its surface for a sliding contact with opposite inside walls of the drive disk. To this end, the substantially cylindrical configuration of the guide member will be especially suited. The necessary engagement of the guide arm in the drive disk makes it possible that the inside wall of the drive disk defines the tilting angle of the drive disk, namely in that it forms a stop for the guide arm, which is rigidly connected to the drive shaft.

As regards the operative connection between the pivoting or wobbling drive disk and the pistons, different constructions are possible. Thus, it would be possible to provide in the effective range of the guide member, on both sides of the drive disk, spherical segments for pivotally engaging the piston. These spherical segments operate between the outside wall of the drive disk that consists of one or more parts, and corresponding slide surfaces in the piston. In this arrangement, the center of the ball formed by the two spherical segments lies in the center of the cylindrical guide member or on its longitudinal axis, so that a displacement of the upper dead center above the stroke position is effectively avoided.

To realize the slide surface that is to be provided at the free end of the piston, the piston could extend with its connecting region about the free end of the drive disk to engage the spherical elements. This embracing engagement could be made approximately C-shaped. In this connection, corresponding surfaces of spherical segments are formed on both sides of the drive disk -- for guiding or receiving the spherical segments. At any rate, it will be easy to guide the piston via the guide arm and the here-proposed joint connection.

As an alternative to the foregoing development, the drive disk could be operatively connected, via a friction bearing, to the piston that mounts at its end a slide shoe. To this end, the piston could be constructed as a cylindrical solid body with a movable slide shoe, with the latter connecting to the piston by means of a spherical joint. The pivotal movement of the drive disk is thus compensated via the spherical joint.

For an operative connection between the drive disk and the piston, one could provide between the drive disk and the piston a special depressor, which pushes the slide shoe onto the drive shaft. At any rate, this depressor is nonrotatably mounted. Between the depressor, the slide shoe, and the drive disk, a friction bearing is operative. In a very advantageous manner, and while realizing a simple type of construction, this friction bearing could comprise a spacer ring extending between the drive disk and the depressor, and a depressor guideway following the spacer ring and extending at least in part over the depressor. In this construction, the surface of the drive disk facing the spacer ring ultimately forms a part of the friction bearing. By all means, it must be ensured that the drive disk is able to rotate, and that a sliding movement of the drive disk relative to the piston is possible.

The depressor pushing the slide shoe onto the drive disk could be constructed preferably as a circular disk. In this connection, an adaptation to the drive disk will be of advantage. Corresponding to the number of the pistons, passageways are provided for the piston or for a connection between the spherical joint and the slide shoe, so that the piston or this connection can extend through the depressor. The passageways may be constructed as slots terminating at the edge of the depressor or as elongate holes. In the case that the passageways are made as -- laterally closed -- elongate holes, one will obtain a higher rigidity of the depressor and, thus, a greater degree of operational reliability.

For axially guiding the drive disk and, in particular, likewise for transmitting the torque, a guide pin rigidly connected to the drive disk engages an elongate hole provided in the drive shaft, or a corresponding passageway, there being an adequate play between the guide pin and the elongate hole. The guide pin could unilaterally extend into the elongate hole and end therein. In a very advantageous manner, however, the guide pin extends through the elongate hole and connects to the drive disk on both sides of the drive shaft. This ensures a reliable, axial guidance between the drive shaft and the drive disk, in particular, however, also for transmitting the torque.

For an axial guidance of the drive disk, it would likewise be possible that a guide member rigidly connected to the drive disk engages a guideway formed on the drive shaft. However, this guideway is formed outside of the drive shaft. Likewise, it would be possible that for an axial guidance, the drive disk engages the drive shaft by means of a guide sleeve. At any rate, it is essential that the axial guidance of the drive disk occur by an operative connection to the drive shaft and possibly not by an external joint connection of the drive disk.

The compressor of the present invention accomplishes a part of the object by the characteristic features of claim 47. Accordingly, a compressor of the initially described kind is characterized in that the components coming into contact with the refrigerant, preferably the walls forming the flow path between the intake area and the discharge area are thermally insulated against the refrigerant at least slightly and in areas of contact.

It has further been recognized by the present invention that the high temperature difference between the discharge area and intake area of a conventional compressor of the kind under discussion leads to a decrease of efficiency, due to a heating of the intake area and, thus, likewise of the there-entering refrigerant because of the thermal conductivity of the compressor components.

Furthermore, it has been recognized by the present invention that the here-analyzed problems can be lessened, in that components coming into contact with the refrigerant are thermally insulated to a certain degree, so that the heating of the taken-in refrigerant is at least reduced. To this end, the walls forming the flow path between the intake area and the discharge area are thermally insulated against the refrigerant at least slightly -- in areas of contact. The definition "thermal insulation" does here not mean a complete insulation for avoiding a heat transfer.

Instead, it means a reduction of the thermal conductivity from components of the compressor to the refrigerant by means of passive measures. In this connection, a thermal insulation provided in areas of contact will already decrease heating of the taken-in refrigerant and, thus, increase the efficiency of the compressor.

Concretely, the thermal insulation could be realized as a lining of a material having a low thermal conductivity. Accordingly, the lining is applied to the walls forming the flow path within the compressor. In this connection, even a partial lining in the intake area will already turn out to be quite successful.

With respect to a particularly simple construction of the compressor, it would be possible to realize the thermal insulation as a coating of a material with a low thermal conductivity. In this instance, conventional coating techniques are considered, which require that the material of the coating, for example, Nikasil, be resistant to temperatures in a range up to 170°C.

Within the scope of a particularly simple construction of the compressor according to the invention, the thermal insulation is provided on the inside wall of the intake channel. This permits reducing the heating of the taken-in refrigerant already on the intake side. To further reduce a heating of the taken-in refrigerant, the thermal insulation is provided in a quite particularly advantageous manner on the inside wall of the entire intake area, thereby decreasing again a heating of the refrigerant sucked in in the intake area.

If one departs from the fact that both the intake area and the discharge area are formed in a housing cover, often also named pressure cover, it would be possible to provide the thermal insulation likewise on the inside wall of the discharge channel, or even on the inside wall of the entire discharge area. This means, one could provide the entire inside wall of the housing cover with a corresponding thermal insulation. To this extent, it would be possible to treat the thermal insulation uniformly on the entire inside wall of the housing cover, or apply it thereto uniformly in the case of a coating.

As previously mentioned, it would be possible to realize the thermal insulation in the form of a lining. To this extent, it is especially advantageous to realize the lining within the housing cover in the form of a loose insert, so that this insert defines the flow path for the refrigerant.

For a further reaching reduction of the thermal conductivity between the housing cover and the refrigerant, it would be possible to space the lining at least slightly from the inside wall of the housing cover, so that a space remains between the actual inside wall of the housing cover and the lining. This interspace decreases again the heat transfer between the housing cover and the refrigerant.

Concretely, the lining could lie against the inside wall of the housing cover with partially shaped, preferably integral spacers, so that the spacing between the insert and the inside wall of the housing cover is not decreased because of the inflowing refrigerant.

Within the scope of a further alternative, it would be possible to realize the lining or coating in the form of a porous foam, whereby a gaseous cushion -- within the foam -- leads to a reduction of the heat transfer between the inside wall of the housing cover and the refrigerant. It is desired that the foam have an open porosity, so that its structure is not destroyed in the event of occurring pressure differences.

As previously mentioned, it would also be possible to coat the inside wall of the housing cover as a whole, namely all over, where the flow path of the refrigerant is defined by the inside wall of the housing cover. Within the scope of such a coating, it is also possible to impart to same a surface structure that promotes the flow, such as, for example, to provide there a defined roughness, which may have the surface structure of a shark skin. At any rate, such a measure is able to promote the flow within the intake and discharge areas.

Likewise, it is possible to provide the inside wall of the housing cover only with a rough finish, and to realize the required surface structure by the coating. Thus, for example, it would also be possible to smooth by means of a suitable coating material the surface of the inside wall of the housing cover, which is too rough for the flow of the refrigerant.

In the case of the compressors of the kind under discussion, the housing cover comprising the intake and discharge areas abuts a valve plate, so that the flow path of the refrigerant is defined at least in part between the valve plate and the inside wall of the housing cover. To this extent, it will be of further advantage, when the thermal insulation is also provided on the valve plate. To this end, it would be possible to coat the valve plate on its side facing the housing cover, as may also be done with the inside wall of housing cover.

As an alternative, it is also possible -- as in the case of the inside wall of the housing cover -- to apply a loose insert to the valve plate on its side facing the housing cover. Moreover, this insert can also be effective in the way of a seal at least in the edge region and in transitional regions between the valve plate and the housing cover.

The foregoing description relates to a reduction of the thermal conductivity between the inside wall of the housing cover and refrigerant. However, it is possible to further reduce a heating of the refrigerant in that surfaces in the pump unit, which additionally form the flow path, or adjoin the flow path, are coated with a material of a low conductivity. To such a coating, it would be possible to attach directly two functions, namely, on the one hand the attempted reduction of a heat transfer between the components of the compressor and the refrigerant, and on the other hand the application of a protective coating against wear and tear for lengthening the service life of the compressor.

If the pump unit is designed and constructed as an axial piston pump, it will be possible to coat the working surface of the cylinder in the cylinder block with the material of a low conductivity. A thermal insulation applied thereto, which simultaneously serves as a protective coating against wear and tear, will be exceptionally advantageous due to the mechanical stress normally occurring in this region.

Furthermore, it is possible to coat the surface of the piston likewise with the low-conductivity material, where the coating serves at the same time as a protection against wear and tear.

Regardless of the foregoing measures for reducing a heat transfer between components of the compressor and the refrigerant by means of a lining or coating, a further measure for reducing the heat transfer could lie in that the housing cover itself is made of a material having a low thermal conductivity. In this connection, the housing cover could consist of a metal having a low thermal conductivity, for example, a high tensile steel, which exhibits a yet considerably lower thermal conductivity than aluminum. In a very advantageous manner, the housing cover is made of a ceramic material or ceramic composite, which reduces the heat transfer quite considerably, even without a coating or lining of the flow path.

Further parts of the object are accomplished by further characteristics of the invention in that in a compressor, whose compressor unit is driven by a belt via a drive shaft and a drive wheel coupled therewith, wherein the drive wheel comprises a belt pulley body engaging the belt, which is directly or indirectly coupled with the drive shaft via a coupling device, the coupling device automatically disengages, when a defined, thermal and/or mechanical load limit is exceeded.

To this extent, it has been recognized by the present invention that the disengagement has to occur automatically, namely upon exceeding a defined thermal load limit, a defined mechanical load limit, or upon exceeding selectively one of the two foregoing load limits. Ultimately, the intent here is to ensure that an automatic disengagement occurs by all means. In this connection, it is possible to predetermine the kind of the load limit to be exceeded as well as the amount of the load capacity.

With that, it is intended to avoid by all means that a blocking of the compressor unit or the drive shaft leads to a blocking or to damage of the belt. Lastly, it is intended to accomplish that even when the compressor unit or the drive shaft blocks, the belt is able to continue to run more or less unimpeded, with only the compressor being inoperative due to the occurred defect.

Concretely, the coupling device that engages in the normal operation of the compressor, could comprise a coupling element that is operative between the belt pulley body and a coupling disk engaging the drive shaft. This coupling element is responsible for the actual engagement and, thus, for the drive of the compressor unit. In particular, also with respect to a small overall size, it will of advantage, when the coupling element is arranged between the inside surface of the belt pulley body and the outside surface of the coupling disk. In such a case, the two surfaces -- inside surface of the belt pulley body and outside surface of the coupling disk -- are arranged in coaxial relationship with each other. In other words, the belt pulley body serving to receive the belt extends substantially annularly around the coupling disk. In this arrangement, both the belt pulley body and the coupling disk have two adjacent and parallel extending surfaces. Arranged therebetween is the coupling device with the coupling element.

Furthermore, it would be possible to provide between the coupling element and the coupling disk or a drive flange of the compressor unit, a vibration damper associated to the coupling device for damping rotary vibrations. This damper may be an elastomeric element or a rubber-metal element. With respect thereto, the coupling device may comprise the coupling element on the one hand and the vibration damper on the other. However, the component used for disengaging is the coupling element.

There exist numerous possibilities of concretely designing and constructing the coupling element, with the latter having to effect a defined disengagement, when a thermal and/or a mechanical load limit is exceeded. To this end, it would be possible to design and construct the coupling element as a spring, which loses at least in part its elasticity under a temperature influence above a predetermined limit value of the temperature, and disengages in this process. Thus, the coupling realized by elasticity is eliminated by a quasi "fatiguing" of the spring. In this connection, the spring may also easily assume a double function, insofar as the spring also permits a disengagement, when a mechanical load limit is exceeded, namely, it acts in the fashion of a slipping clutch. Both modes of operation are possible, namely a disengagement when a thermal and a mechanical load limit are exceeded.

Likewise, it is possible to construct the coupling element as a permanent magnet cooperating directly or indirectly with the magnetic material of the belt pulley body and coupling disk. This permanent magnet would have to loose its magnetic effect at least in part under a temperature influence above a predetermined limit value, and disengage in this process. In this respect, a disengagement would be ensured, when a definable, thermal load limit is exceeded.

The provision of a magnetically operating coupling device could also perform a formfitting engagement, namely in that the coupling element comprises magnetic coupling parts adapted for a formfitting engagement, as well as an at least weak electromagnet, which disengages the coupling parts, and decouples in this process, when a blocking of the drive shaft is detected. Such a provision presents itself at least when the compressor is electrically controlled, i.e., when it is easy to detect the blocking of the compressor.

Within the scope of a very advantageous development, the coupling element is designed and constructed in the way of an annular pressure body for a frictional engagement between the belt pulley body and the coupling disk or drive shaft. Due to its pressed state between the components, this pressure body effects a frictional engagement. In the case that a vibration damper is provided in addition, the coupling element and, thus, the pressure body are arranged between the belt pulley body and the vibration damper. At any rate, the pressure body is operative between the belt pulley body and the drive shaft, in each case via those components that are arranged therebetween from a functional viewpoint.

Concretely, the pressure body could be realized in the form of bellows, preferably thin-walled metal bellows. A thin-walled construction would be of advantage, inasmuch as same could be spatially expanded by a flow medium. For a frictional engagement, the pressure body could be filled with a flow medium under a predeterminable pressure. The flow medium may be a gas, a liquid, or at least in part a liquid and otherwise gas. At any rate, in its pressure-biased state, the pressure body effects an automatic engagement, so that the compressor is rotatably driven via the belt pulley body.

The provision of the above-described pressure body thus permits a disengagement, when a defined thermal and/or mechanical load limit is exceeded. To this end, the flow medium could exhibit such a high thermal coefficient of expansion that, upon exceeding a predetermined temperature, it opens the pressure body at least in sections due the then prevailing inside pressure, or even causes it to explode, and disengages it in this process. At any rate, this presumes that the belt slips over the belt pulley body, thereby heating it, in practice, to about 300°C , and that it transfers the temperature to the pressure body directly adjacent the inside surface of the belt pulley body. At any rate, the temperature increase leads to such an expansion of the flow medium that the pressure body leaks or even explodes, thereby allowing the pressure to escape, and discontinuing in the end the contact pressure that is applied by the pressure body and required for a forced engagement between the belt pulley body and drive shaft or coupling disk. Thus, a disengagement is realized, when a thermal load limit is exceeded.

Likewise, it is possible that the pressure body has at least one predetermined mechanical breaking point that is used for relieving pressure and, thus, for disengagement. To this end, a predetermined breaking point could tear, so that the pressure prevailing within the pressure body or the flow medium therein is able to escape. However, this will require that the force necessary for tearing be smaller than that force, which holds, by static friction, adhesion, or the like, the pressure body in its position relative to the coupling disk or to the vibration damper arranged therebetween. This would ensure a disengagement, when a mechanical load limit is exceeded.

Likewise, it would be possible that for purposes of relieving pressure and thus for disengaging, the pressure body comprises at least one safety fuse, which melts regardless of a possible pressure rise within the pressure body, when a predeterminable temperature is reached, and releases the pressure medium. In a particularly advantageous manner a plurality of safety fuses are provided along the circumference of the pressure body, so that irrespective of the angular position of the belt pulley or belt pulley body, at least one safety fuse is arranged in the vicinity of the region of the belt pulley, which overheats due to a slipping of the belt. At any rate, this also creates an automatic disengagement, namely by exceeding a defined thermal load limit.

Since a malfunction of the compressor can lead both to a slipping of the belt and, thus, to an enormous heating of the belt pulley body and to a total blocking of the belt and, thus, to a choking of the drive engine, or even to a destruction of the belt, it will be of advantage to combine the two previously described disengagement mechanisms, namely to the extent that the pressure body may comprise for disengagement at least one predetermined mechanical breaking point serving to relieve pressure, and at least one safety fuse serving to relieve pressure. In this connection, it should be made sure that the pressure body be held in its position in a material-engaging manner, at least, however, with a high coefficient of static friction, so that the predetermined breaking point tears indeed under a corresponding mechanical load.

As regards a concrete realization of the pressure body, it will be of further advantage, when same extends annularly, preferably in the way of a hollow cylinder, between the inside surface of the belt pulley body and the outside surface of the coupling disk or a vibration damper. In other words, the pressure body extends annularly between the belt pulley body and the coupling disk, namely between the two inside surfaces of the components under discussion that are to be engaged. The vibration damper provided in this location may serve as a quasi intermediate element, but has nothing to do with engagement or disengagement in a functional respect.

In the longitudinal section of the compressor or the pressure body, same could have a substantially rectangular pressure chamber. Adjacent to this rectangular pressure chamber are outwardly directed separating regions of the pressure body, which narrow relative to the pressure chamber in the longitudinal section thereof. These separating regions have closely adjacent walls, which, due to their vicinity, are closed as a whole, in zones, or point by point, with a safety fuse. A connection of the walls in the separating regions can also be realized in the way of a predetermined breaking point.

In particular, when combining an automatic disengagement upon exceeding both a defined thermal and a defined mechanical load limit, it will be of advantage, when two opposite separating regions are formed, which adjoin the pressure chamber with their arms in U-shape in the longitudinal section of the pressure body. In this arrangement, the one separating region will serve for disengagement, when a defined thermal load limit is exceeded, and the other separating region, when a defined mechanical load limit is exceeded. In this connection, the one separating region is a safety fuse, and the other separating region a predetermined mechanical breaking point. Both the safety fuse and the preset breaking point may be provided along the entire circumference of the pressure body continuously, in zones, or only point by point.

There now exist various possibilities of improving and further developing the teaching of the present invention. To this end, reference may be made on the one hand to the claims dependent from claim 1, on the other hand to the following detailed description of embodiments of the invention with reference to the drawing. In conjunction with the description of the preferred embodiments of the invention with reference to the drawing, generally preferred embodiments and further developments of the teaching will also be explained. In the drawing:

Figure 1 is a schematic, longitudinally sectioned view of an embodiment of a compressor according to the invention, wherein a drive disk constructed as a pivot disk is inclined or pivoted by a minimum amount that is defined by a stop;

Figure 2 illustrates the embodiment of Figure 1, wherein the pivot disk is inclined according to a stop defining a maximum pivotal movement;

Figure 3 is a cross sectional view of the embodiment of Figure 1 along line B-B;

Figure 4 is a schematic, longitudinally sectioned view of a further embodiment of a compressor according to the invention, wherein a compressor unit is merely indicated;

Figure 5 is a schematic detail view of an operative connection between the drive shaft and piston, wherein the pivot disk is maximally pivoted;

Figure 6 illustrates the embodiment of Figure 5, wherein the pivot disk is inclined or pivoted by the minimal amount;

Figure 7 is a partial, schematic top view of the jointed engagement between the drive shaft and piston including the pivot disk;

Figure 8 is a detail view according to the illustration of Figure 6 showing a further embodiment of the compressor according to the invention, wherein a friction bearing is operative between the piston and a swash plate;

Figure 9 is a schematic top view of a depressor as is used in the embodiment of Figure 8;

Figure 10 is a partial and sectional schematic side view of a further embodiment of a compressor according to the invention, the Figure showing only an intake area and a discharge area in a housing cover;

Figure 11 is an enlarged sectional side view of the intake area in the housing cover, wherein a lining is provided as a thermal insulation in spaced relationship with the inside wall of the housing cover;

Figure 12 is a schematic, longitudinally sectioned view of an embodiment of a species-forming compressor with the essential components;

Figure 13 is a partial, longitudinally sectioned schematic view of an embodiment of a compressor according to the invention with a special belt pulley arrangement;

Figure 14 is an enlarged view of detail "X" of Figure 13;

Figure 15 is a partial, longitudinally sectioned schematic view of a second embodiment of a compressor according to the invention with a belt pulley arrangement; and

Figure 16 is an enlarged view of detail "Y" of Figure 15.

Figures 1-3 show a compressor for an air conditioning system of an automobile. The compressor comprises a housing **1** and a compressor unit **2** accommodated in the housing **1** for taking in and compressing a refrigerant. The refrigerant may preferably be CO_2.

The refrigerant flows from an intake area **4** formed in a front-end housing cover **3** through the compressor unit **2** into a discharge area **5** likewise formed in the housing cover **3**.

Figures 1 and 2 show very clearly the compressor unit **2** accommodated in housing **1** for taking in and compressing the refrigerant. The compressor unit **2** comprises pistons **7** reciprocating in a cylinder block **6** and a drive disk driving pistons **7**. Concretely, the drive disk is a pivot disk **8**.

In accordance with the invention, the pivot disk **8** is supported such that a center line **9** of the bearing mount, i.e. a pivot or bearing axis of pivot disk **8** forms a tangent to a reference circle **10** defining the stroke, so that an angle of inclination **11** of pivot disk **8** is variable without displacing the dead center above the stroke position. At any rate, Figures 1 and 2 show jointly that the pivot disk **8** tilts exactly above the longitudinal axis **12** of the piston, namely that the instantaneous pole lies on the reference circle **10**, and is stationary when related to the rotating reference system of a drive shaft **13**.

As clearly shown in Figure 3, the pivot disk **8** is operatively connected to the drive shaft **13** via a coupling device **14** arranged in its edge region. To this end, the pivot disk **8** is jointed to an entraining member **15** that is nonrotatably connected to drive shaft **13**. The entraining member **15** is a cylindrical body, which simultaneously forms a stop **16**, **17** for the pivotal movement of pivot disk **8**. This is best seen in particular in Figures 1 and 2, with Figure 1 showing the stop **16** for a minimum pivotal movement and Figure 2 the stop **17** for a maximum pivotal movement of pivot disk **8** -- with the respective position of pivot disk **8**.

As further shown in Figure 3, the pivot disk **8** is supported by means of pivot bearing **18** that is operative between the pivot disk **8** and entraining member **15**. This pivot bearing **18** comprises bearing pins **19**, which engage the entraining member **15** for pivotally jointing the pivot disk **8**.

As can be noted from Figures 1-3 as a whole, coupling means **20** are operative between the pivot disk **8** and pistons **7**. These coupling means **20** comprise a brace **21** associated to piston **7** and a slide shoe or spherical body **22** associated to pivot disk **8**. In this arrangement, the brace **21** of piston **7** extends through the pivot disk **8** on the side facing the drive shaft **13**, whence it extends outward with inclusion of spherical body **22**, surrounding the latter at least in part.

Figures 1-3 only indicate that the cylinder block **6** comprises an antirotation device for the piston **7**, namely in the form of a support surface **23** between the cylinder block **6** and piston **7**.

Figure 4 shows a compressor for the air conditioning system of an automobile. The compressor comprises a housing **101** and a compressor unit **102** accommodated in the housing **101** and merely indicated in Figure 4 for taking in and compressing a refrigerant. This refrigerant may preferably be CO_2.

The refrigerant flows from an intake area **104** formed in a front-end housing cover **103** through the compressor unit **102** into a discharge area **105** likewise formed in the housing cover **103**.

Figure 4 only indicates the compressor unit **102** accommodated in the housing **101** for taking in and compressing the refrigerant. The compressor unit **102** comprises pistons **107** reciprocating in a cylinder block **106**, and a drive disk driving the pistons **107**. The drive disk is not shown in Figure 4. Concretely, the drive disk is a pivot disk **108** (note Figures 5-8).

In accordance with the invention, the pivot disk **108** shown in Figures 5-8 is supported such that a center line **109** of the bearing mount, i.e. the pivot or bearing axis of pivot disk **108** forms a tangent to a reference circle **110** defining the stroke (note Figure 7), so that an angle of inclination **111** of pivot disk **108** is variable, without displacing the upper dead center above the stroke position. At any rate, Figures 5-8 show jointly that the pivot disk **108** tilts exactly above the longitudinal axis **112** of the piston, namely that the instantaneous pole lies on the reference circle **110**, and is stationary when related to the rotating reference system of drive shaft **113**.

As best seen in Figures 5-8, the drive disk or pivot disk **108** is operatively connected to drive shaft **113** via a guide arm **114** rigidly connected to drive shaft **113** and mounted for sliding movement in the edge region of pivot disk **108**. As further shown in Figures 5-8 together, the guide arm **114** is constructed as a square bar and arranged orthogonally to the longitudinal axis **115** of drive shaft **113**. In the illustrated embodiments, the drive arm **114** is pressed into the drive shaft **113**.

As very clearly illustrated in Figures 5, 6, and 8, a guide member **116** is formed at the free end of guide arm **114** for a sliding support of the guide arm **114** in pivot disk **108**. The guide member **116** is constructed substantially cylindrical crosswise to the longitudinal axis of guide arm **114**.

In the illustrated embodiments, the guide member **116** is directly supported in pivot disk **108**. An indirect bearing mount via other bearing/slide mechanisms is possible. At any rate, the guide member **116** with its surface **117** serves for a sliding contact with opposite inside walls of the pivot disk **108**, as can very clearly be noted from Figures 5, 6, and 8.

In the embodiment shown in Figures 5, 6, and 7, spherical segments **118** are provided in the effective range of guide member **116** on both sides of pivot disk **108** for a pivotal engagement in piston **107**. The center **119** of the ball formed by the two spherical segments **118** lies in the center of the cylindrical guide member **116** or on its longitudinal axis **120**.

As further shown in Figures 5 and 6, the piston **107** extends with its connecting region **121** over the free end of pivot disk **108** to engage spherical segments **118**. The connecting region **121** has a C-shaped cross section, so that it is possible to engage over the free end of pivot disk **108**.

Figures 8 and 9 show a further embodiment of a compressor according to the invention. These Figures show only details, as are also illustrated in Figures 5 and 6 with respect to the foregoing embodiment. Accordingly, the pivot disk **108** is operatively connected via a friction bearing **122** to the piston **107** that mounts at its end a slide shoe **123**. The piston **107** is constructed as a cylindrical solid body, with the slide shoe **123** connecting to the piston **107** by means of a spherical joint **124**, which permits a tilting of slide shoe **123**.

Concretely, the slide shoe **123** is held or pushed onto pivot disk **108** by means of a depressor **125**. As shown in Figure 9, the depressor **125** is constructed in the form of a disk and nonrotatably mounted. Between the depressor **125**, the slide shoe **123**, and the pivot disk **108**, friction bearing **122** is operative. To this end, the friction bearing **122** comprises a spacer ring **126** extending between the pivot disk **108** and depressor **125** and a depressor guideway **127** adjacent spacer ring **126** and extending in part over the depressor **125**.

Figure 9 only indicates that the depressor **125** comprises for the piston **107** or for a connection **129** between the spherical joint **124** and slide shoe **123**, passageways **128** in an amount corresponding to the number of pistons **107**. The passageway **128** may be designed and constructed as a slot **130** terminating at the edge of depressor **125** or as a laterally closed, elongate hole **131** that increases the rigidity of depressor **125**.

As further clearly shown in Figures 5-8, a guide pin **131** rigidly connected to pivot disk **108** engages an elongate hole **132** formed in pivot disk **108** for an axial guidance thereof. In so doing, the guide pin **131** extends through the elongate hole **132** and connects to pivot disk **108** on both sides thereof, as best seen in Figure 7.

Figure 10 shows a compressor for an air conditioning system of an automobile. The compressor comprises a housing **201** and a compressor unit **202** accommodated in the housing **201** for taking in and compressing a refrigerant. The refrigerant may be CO_2.

The refrigerant flows from an intake area **204** formed in a front-end housing cover **203** through the compressor unit **202** into a discharge area **205** likewise formed in the housing cover **203**.

In accordance with the invention, components of the compressor that come into contact with the refrigerant, namely the walls that form a flow path **206** between intake area **204** and discharge area **205**, are thermally insulated against the refrigerant at least in areas of contact.

In the embodiment illustrated in Figure 10, a thermal insulation **207** is realized as a coating of a material having a low thermal conductivity. The thermal insulation **207** is provided both on the inside wall **208** of an intake channel **209** and on the inside wall **210** of a discharge channel **211**. More specifically, the entire inside walls **208**, **210** of the intake area **204** are coated in a thermally insulating manner. Ultimately, the entire inside walls **208**, **210** of the housing cover **203** are coated to this end.

In the embodiment partially and schematically illustrated in Figure 11, it is indicated for the intake area **204** that the inside walls **208**, **210** of housing cover **203** are there lined in the form of a loose insert **212**. This insert **212** is slightly spaced from the inside walls **208**, **210**. This spaced relationship is implemented by integral spacers **213**, which lie directly against the inside walls **208, 210** of housing cover **203**.

As can further be noted from Figure 10, the housing cover **203** adjoins a valve plate **214**. A thermal insulation is likewise provided on valve plate **214**, the latter being coated on its side facing the housing cover **203**, preferably with the same material as the inside walls **208, 210** of housing cover **203**. In this respect, the flow path formed between housing cover **203** and valve plate **214** is totally coated and thus thermally insulated.

With respect to the thermally insulating and wear-resistant coating of further components of the compressor, the general part of the specification is herewith incorporated by reference for purposes of avoiding repetitions. The same applies to the material of housing cover **203**.

The embodiment of a species-forming compressor as shown only by way of example in Figure 12, is an axial piston compressor. In this embodiment, a compressor unit **301** not described in greater detail is accommodated in a housing **302**. The housing **302** essentially comprises two housing parts **303, 304**, with the housing part **303** forming a so-called drive chamber **305** that accommodates the compressor unit **301**.

For example, an internal combustion engine drives the compressor unit **301** via a belt pulley **306**. From there, the drive is effected via a drive shaft **307** that rotates about an axis of rotation **308**. The drive shaft **307** is rotatably supported in the housing **302** and in the region of the belt pulley **306**.

To drive pistons **309**, a swash plate **310** is provided which acts via bearings **311** upon a receiving disk **312** nonrotatably mounted in housing **302**. The receiving disk **312** engages, via a connecting rod **313**, the piston or pistons **309**. According to this arrangement, the piston **309** reciprocates, upon a rotation of swash plate **310**, via receiving disk **312**, in direction of its longitudinal axis, with the illustrated embodiment comprising a plurality of pistons **309**.

Since the compressor illustrated in the Figure is a compressor for an air conditioning system of an automobile, it is driven by the internal combustion engine of an automobile not shown. In this instance, a driving moment is introduced to a drive wheel **314** of the compressor via a suitable belt pulley that connects to the crankshaft of the internal combustion engine. This drive wheel **314** comprises a belt pulley body **315**, which guides a belt **316**.

As indicated in the illustration of Figure 12, the belt pulley body **315** is put into rotation by belt **316**. In the embodiments shown in Figures 13-16, the torque introduced into belt pulley body **315** is transmitted via a coupling device **317** to the drive shaft **307**. In the selected embodiments, the coupling device **317** comprises a vibration damper **318**.

In accordance with the invention, the coupling device **317** is designed such that upon exceeding a defined thermal and/or mechanical load limit, it automatically disengages, so that the belt pulley body **315** is able to rotate unimpeded in its disengaged state.

As best seen in Figures 13-16, the coupling device **317** comprises a coupling element **320** that is operative between the belt pulley body **315** and drive shaft **307** or a coupling disk **319**. In this arrangement, the coupling element **320** extends between the inside surface of belt pulley body **315** and the outside surface of coupling disk **319**, and the two surfaces -- inside surface of belt pulley body **315** and outside surface of coupling disk **319** -- are arranged in coaxial relationship with each other.

As previously stated, a vibration damper **318** for damping rotational vibrations, which is associated to the coupling device **317**, is provided between the coupling element **320** and coupling disk **319**, or between the two adjacent surfaces of these components. This vibration damper **318** has nothing to do with the actual engagement and disengagement procedures.

As further shown in Figures 13-16, the coupling element **320** is designed and constructed in the fashion of an annularly constructed pressure body **321** for frictionally engaging the belt pulley body **315** and coupling disk **319**. Concretely, the pressure body **321** is a thin-walled metal bellows, which is filled for a frictional engagement with a flow medium under a predeterminable pressure. The clamping effect caused by the pressure body **321** effects the engagement between belt pulley body **315** and coupling disk **319**. In this connection, the pressure body **321** can be secured in its position by gluing, soldering, spot welding, or the like.

As can be noted from the enlarged view of Figure 14, the pressure body **321** comprises predeterminable mechanical breaking points, which are used for relieving pressure and, thus, for disengagement. Likewise at these points, safety fuses **323** are provided, so that in the embodiment shown in Figures 13 and 14, a combination of a coupling device **317** is realized, which automatically disengages when both a defined thermal load limit and a defined mechanical load limit are exceeded.

In the embodiment shown in Figures 15 and 16, the predetermined mechanical breaking points **322** and the safety fuses **323** are not located opposite to each other as in the embodiment shown in Figures 13 and 14, but are formed only on the side of pressure body **321** that faces the inside surface of belt pulley body **315**.

Furthermore, as jointly shown in Figures 13-16, the pressure body **321** comprises a pressure chamber **324** substantially rectangular in its longitudinal section, and adjoining, outwardly directed separating regions **325** that cross sectionally narrow relative to the pressure chamber **324**. In the embodiment shown in Figures 13 and 14, two separating regions are arranged in facing relationship, which connect in the longitudinal section of pressure body **321,** with their arms in U-shape to the actual pressure chamber **324**.

In the embodiment shown in Figures 15 and 16, only one side of the pressure body **321** -- the side facing the inside surface of belt pulley body **315** -- is provided with a separating region **325**, which comprises both the predetermined mechanical breaking points **322** and the safety fuses **323**.

Furthermore, it should be noted that, as shown in Figures 13 and 15, needle bearings **326** are provided, which support the drive shaft **307** outside of drive chamber **305**. Moreover, the needle bearings **326** are used to support the coupling disk **319**. Since the needle bearings **326** are arranged outside of the drive chamber **305**, they operate under ambient atmospheric pressure, with seals **327** sealing against the drive chamber **305**.

Finally, as can be noted from Figures 13 and 15, friction linings **328** are provided between the belt pulley body **315** and the coupling disk **319**, which serve as "safety bearings" or as a bearing surface or bearing coating, inasmuch as, should the compressor block and a disengagement occur according to the foregoing description, the foregoing needle bearings **326** will also be inoperative. To the end that the belt pulley body **315** is able to rotate unimpeded over an acceptable period of time, for example for at least some hours, a kind of safety bearing is needed for the belt pulley body **315.** This safety bearing is provided by the friction lining **328** that extends both radially and transversely. For this purpose, it is possible to use temperature-resistant materials.

As regards the operation of the realized coupling device, the general part of the specification is herewith incorporated by reference for purposes of avoiding repetitions.

The claims accompanying the application are proposed formulations without prejudice for obtaining a further reaching patent protection. Applicant reserves itself the right to claim still further characteristics that have so far been disclosed only in the specification and/or the drawings.

The dependencies claimed in the dependent claims refer to the further development of the subject matter of the independent claim by the characteristic features of the respective dependent claim; they should not be construed a waiver of obtaining an independent, valid protection for the characteristic features of the dependent claims.

However, the subject matters of these dependent claims also form independent inventions, which contain a development independent of the subject matters of the preceding dependent claims.

Likewise, the invention is not limited to an embodiment or the embodiments of the specification. Rather, within the scope of the invention, numerous changes and modifications are possible, in particular such variants, elements, and combinations, and/or materials, which are inventive, for example, by combining or modifying individual features, or elements, or method steps described in conjunction with the general specification and embodiments, as well as contained in the claims and drawing, and which result by combinable features in a new subject matter, or new method steps, or sequences of method steps, even to the extent that they relate to production, testing, and working methods.

As regards further advantageous developments of the present invention, which are not shown in the Figures, the general part of the specification is herewith incorporated by reference for purposes of avoiding repetitions.

Finally, it should be noted that the above-described embodiments merely given by way of example describe in greater detail only the teaching of the invention, without however limiting it to the embodiments.

CLAIMS

1. Compressor, in particular for an air conditioning system of an automobile, with a housing (1) and a compressor unit (2) accommodated in the housing (1) for taking in and compressing a refrigerant, wherein the compressor unit (2) comprises pistons (7) reciprocating in a cylinder block (6) and a drive disk -- swash plate or pivot disk (8) -- for driving the pistons (7),
characterized in that the drive disk (8) is supported such that a center line (9) of the bearing mount (pivot or bearing axis) forms a tangent to a reference circle (10) defining the stroke, so that an angle of inclination (11) of the drive disk (8) is variable, without displacing the upper dead center above the stroke position.

2. Compressor, in particular of claim 1, characterized in that the drive disk (8) operatively connects to a drive shaft (13) via a coupling device (14) associated to its edge region.

3. Compressor, in particular of claim 2, characterized in that the drive disk (8) is jointed via the coupling device (14) to an entraining member (15) that is nonrotatably connected to the drive shaft (13).

4. Compressor, in particular of claim 3, characterized in that the drive disk (8) is surrounded by the entraining member (15) at least in part.

5. Compressor, in particular of claim 3 or 4, characterized in that the entraining member (15) is

designed and constructed annular, preferably cylindrical.

6. Compressor, in particular of claim 3 or 4, characterized in that the entraining member (15) is designed and constructed as a ring segment.

7. Compressor, in particular of one of claims 3-6, characterized in that the entraining member (15) is an integral part of the drive shaft (13).

8. Compressor, in particular of one of claims 3-7, characterized in that the entraining member (15) forms a stop (16, 17) for the pivotal movement of the drive disk (8).

9. Compressor, in particular of claim 8, characterized in that the entraining member (15) forms a stop (16, 17) respectively for both the maximal and the minimal pivotal movement of the drive disk (8).

10. Compressor, in particular of one of claims 1-9, characterized in that the drive shaft (8) is supported via a pivot bearing (18) that is operative between the drive disk (8) and the entraining member (15).

11. Compressor, in particular of claim 10, characterized in that the pivot bearing (18) comprises a spherical segment.

12. Compressor, in particular of claim 10, characterized in that the pivot bearing comprises a bearing pin (19).

13. Compressor, in particular of one of claims 1-12, characterized in that between the drive disk (8) and pistons (7) coupling means (20) are operative, which comprise a brace (21) associated to the piston (7) and preferably operating in the fashion of a slideway, and a spherical member (22) associated to the drive disk (8) and preferably operating in the fashion of a slide shoe.

14. Compressor, in particular of claim 13, characterized in that the brace (21) of piston (7) extends through the drive disk (8) on the side facing the drive shaft (13), and surrounds it in outward direction at least in part, preferably with the inclusion of the slide shoe or spherical member (22).

15. Compressor, in particular of one of claims 1-14, characterized in that the cylinder block (6) comprises an antirotation device for the pistons (7).

16. Compressor, in particular of claim 15, characterized in that the antirotation device is realized as a support surface (23) between the cylinder block (6) and the pistons (7).

17. Compressor, in particular of claim 15, characterized in that the antirotation device is realized as a support surface (23) between the pistons (7) and the drive shaft (13).

18. Compressor, in particular of claim 15, characterized in that the antirotation device is

realized as a support surface (23) between the pistons (7) and the entraining member (15).

19. Compressor, in particular of claim 15, characterized in that "noncircular" pistons (7) are provided as an antirotation device.

20. Compressor, in particular of claim 15, characterized in that a guide element, preferably a pin is provided as an antirotation device in the piston (7).

21. Compressor, in particular of one of claims 1-20, characterized in that the drive disk (8) operates in the fashion of a centrifugal governor.

22. Compressor, in particular of one of claims 1-21, characterized in that the drive disk (108) is operatively connected to the drive shaft (113) via a guide arm (114) rigidly connected to the drive shaft (113) and supported for sliding movement in the drive disk (108), preferably in the edge region thereof.

23. Compressor, in particular of claim 22, characterized in that the guide arm (114) is designed and constructed as a square bar.

24. Compressor, in particular of one of claims 1-23, characterized in that the guide arm (114) is arranged orthogonally to the longitudinal axis (115) of the drive shaft (113).

25. Compressor, in particular of one of claims 1-24, characterized in that the guide arm (114) is pressed into the drive shaft (113).

26. Compressor, in particular of one of claims 1-25, characterized in that for a sliding support of the guide arm (114) in the drive disk (108), a guide member (116) is formed at the free end of the guide arm (114).

27. Compressor, in particular of claim 26, characterized in that the guide member (116) is designed and constructed substantially cylindrical crosswise to the longitudinal axis of the guide arm (114).

28. Compressor, in particular of one of claims 1-27, characterized in that the guide member (116) is supported directly or indirectly in the drive disk (108).

29. Compressor, in particular of claim 27 or 28, characterized in that the guide member (116) is used with its surface (117) for a sliding contact with opposite inside walls of the drive disk (108).

30. Compressor, in particular of one of claims 27-29, characterized in that in the effective range of the guide member (116), spherical segments (118) are provided on both sides of the drive disk (108) for pivotally engaging the piston (107).

31. Compressor, in particular of claim 30, characterized in that the center (119) of the ball

formed by the two spherical segments (118) lies in the center (119) of the cylindrical guide member (116) or on the longitudinal axis (120) thereof.

32. Compressor, in particular of claim 30 or 31, characterized in that the piston (107) extends with its connecting region (121) over the free end of the drive disk (108) to engage spherical segments (118).

33. Compressor, in particular of one of claims 1-29, characterized in that the drive disk (108) is operatively connected via a friction bearing (122) to the piston (107) that mounts at its end a slide shoe (123).

34. Compressor, in particular of claim 33, characterized in that the piston (107) is designed and constructed as a cylindrical solid body with a movable slide shoe (123).

35. Compressor, in particular of claim 34, characterized in that the slide shoe (123) connects to the piston (107) by means of a spherical joint (124).

36. Compressor, in particular of one of claims 33-35, characterized in that the slide shoe (123) is held on the drive disk (108) by means of a depressor (125).

37. Compressor, in particular of claim 36, characterized in that the depressor (125) is nonrotatably supported.

38. Compressor, in particular of claim 37, characterized in that a friction bearing (122) is operative between the depressor (125), the slide shoe (123), and the drive disk (108).

39. Compressor, in particular of claim 38, characterized in that the friction bearing (122) comprises a spacer (126) extending between the drive disk (108) and depressor (125), and a depressor guideway (127) adjoining the spacer (126) and extending at least in part over the depressor (125).

40. Compressor, in particular of one of claims 36-39, characterized in that the depressor (125) is designed and constructed preferably as a circular disk.

41. Compressor, in particular of one of claims 36-40, characterized in that the depressor (125) comprises passageways (128) for the piston (107) or for a connection between the spherical joint (124) and the slide shoe (123) in an amount corresponding to the number of pistons (107).

42. Compressor, in particular of claim 41, characterized in that the passageways (128) are designed and constructed as slots (130) terminating at the edge of the depressor (125) or as elongate holes (132).

43. Compressor, in particular of one of claims 1-42, characterized in that for the axial guidance of the drive disk (108), a guide pin (131) rigidly connected to the drive disk (108) engages for

sliding movement an elongate hole (132) formed in the drive shaft (113).

44. Compressor, in particular of claim 43, characterized in that the guide pin (131) extends through the elongate hole (132) and connects to the drive disk (108) on both sides of the drive shaft (113).

45. Compressor, in particular of one of claims 1-42, characterized in that for the axial guidance of the drive disk (108), a guide member rigidly connected to the drive disk (108) engages a guideway formed in the drive shaft (113).

46. Compressor, in particuar of one of claims 1-42, characterized in that for an axial guidance the drive disk (108) engages the drive shaft (113) by means of a guide sleeve.

47. Compressor, whose refrigerant flows from an intake area (204) preferable formed in a front-end housing cover (203) through a compressor unit (202) into a discharge area (205) likewise preferably formed in the housing cover (203), in particular of one of claims 1-46,
characterized in that components coming into contact with the refrigerant, preferably the walls forming a flow path (206) between the intake area (204) and discharge area (205) are thermally insulated against the refrigerant at least slightly and in areas of contact.

48. Compressor, in particular of claim 47, characterized in that a thermal insulation (207, 215) is realized as a lining of a material having a low thermal conductivity.

49. Compressor, in particular of claim 47, characterized in that the thermal insulation (207, 215) is realized as a coating of a material having a low thermal conductivity.

50. Compressor, in particular of one of claims 47-49, characterized in that the thermal insulation (207) is provided on an inside wall (208) of an intake channel (209).

51. Compressor, in particular of one of claims 47-49, characterized in that the thermal insulation (207) is provided on the inside wall (208) of the entire intake area (204).

52. Compressor, in particular of one of claims 47-51, characterized in that the thermal insulation (207) is provided on an inside wall (210) of a discharge channel (211).

53. Compressor, in particular of one of claims 47-50, characterized in that the thermal insulation (207) is provided on the inside wall (210) of the entire discharge area (205).

54. Compressor, in particular of one of claims 47-53, characterized in that the thermal insulation (207) is provided on all inside walls (208, 210) of the housing cover (203).

55. Compressor, in particular of claim 54, characterized in that the inside walls (208, 210) of the housing cover (203) are lined.

56. Compressor, in particular of claim 55, characterized in that the lining is designed and constructed in the form of a loose insert (212).

57. Compressor, in particular of claim 56, characterized in that the lining is spaced at least slightly from the inside walls (208, 210).

58. Compressor, in particular of claim 57, characterized in that the lining lies with partially formed, preferably integral spacers (213) against the inside walls (208, 210) of the housing cover (203).

59. Compressor, in particular of claim 55, characterized in that the lining is realized as a porous foam.

60. Compressor, in particular of claim 59, characterized that the foam has an open porosity.

61. Compressor, in particular of claim 55, characterized in that the inside walls (208, 210) of the housing cover (203) are coated as a whole.

62. Compressor, in particular of claim 61, characterized in that the coating has a surface structure that promotes the flow.

63. Compressor, in particular of claim 62, characterized in that the surface structure has a defined roughness, preferably in the form of a shark skin.

64. Compressor, in particular of one of claims 1-63, wherein the housing cover (203) adjoins a valve plate (214), characterized in that the thermal insulation (215) is provided on the valve plate (214).

65. Compressor, in particular of claim 64, characterized in that the valve plate (214) is coated on its side facing the housing cover (203).

66. Compressor, in particular of claim 64, characterized in that the valve plate (214) is covered with a loose insert (212) on its side facing the housing cover (203).

67. Compressor, in particular of one of claims 1-66, characterized in that surfaces forming the flow path or adjoining the flow path (206) in the pump unit are coated with a material of a low thermal conductivity.

68. Compressor, in particular of claim 67, wherein the pump unit is designed and constructed as an axial piston pump, characterized in that the working surface of the cylinder in the cylinder block is coated with a material of a low thermal conductivity.

69. Compressor, in particular of claim 67 or 68, characterized in that the piston surface of the

piston is coated with a material of a low thermal conductivity.

70. Compressor, in particular of one of claims 1-69, characterized in that the housing cover (203) is made of a material with a low thermal conductivity.

71. Compressor, in particular of claim 70, characterized in that the housing cover (203) is made of metal with a low thermal conductivity.

72. Compressor, in particular of claim 71, characterized in that the housing cover (203) is made of a high tensile steel.

73. Compressor, in particular of claim 70, characterized in that the housing cover (203) is made of a ceramic material.

74. Compressor, whose compressor unit (301) is driven by a drive belt (316) via a drive shaft (307) and a drive wheel (314) coupled with the drive shaft (307), wherein the drive wheel (314) comprises a belt pulley body (315) engaging the belt (316), the belt pulley body (315) connecting directly or indirectly to the drive shaft (307) via a coupling device (317), in particular according to one of claims 1-73, **characterized in** that the coupling device (317) automatically disengages when a defined thermal and/or mechanical load limit is exceeded.

75. Compressor, in particular of claim 74, characterized in that the coupling device (317)

comprises a coupling element (320) that is operative between the belt pulley body (315) and a coupling disk (319) connected to the drive shaft (307).

76. Compressor, in particular of claim 75, characterized in that the coupling element (320) is arranged between the inside surface of the belt pulley body (315) and the outside surface of the coupling disk (319), the two surfaces being arranged in coaxial relationship with each other.

77. Compressor, in particular of claim 75 or 76, characterized in that a vibration damper (318) associated to the coupling device (317) for damping rotational vibrations is provided between the coupling element (320) and the coupling disk (319) or a drive flange of the compressor unit (301).

78. Compressor, in particular of claim 77, characterized in that the vibration damper (318) is an elastomeric element.

79. Compressor, in particular of claim 77, characterized in that the vibration damper (318) is a rubber-metal element.

80. Compressor, in particular of one of claims 75-79, characterized in that the coupling element (320) is designed and constructed as a spring, which loses its elasticity at least in part under the influence of a temperature above a predetermined temperature limit value, and disengages in this process.

81. Compressor, in particular of one of claims 75-79, characterized in that the coupling element (320) is designed and constructed as a permanent magnet cooperating directly or indirectly with the magnetic material of the belt pulley body (315) and coupling disk (319), the permanent magnet losing its magnetic effect at least in part under the influence of a temperature above a predetermined limit value, and disengaging in this process.

82. Compressor, in particular of one of claims 75-79, wherein the compressor is electrically controlled, characterized in that the coupling element (320) comprises formfittingly engaging, magnetic coupling parts as well as an at least weak electromagnet, which disengages the coupling parts and decouples same in this process, when a blocking of the drive shaft (307) is detected.

83. Compressor, in particular of one of claims 75-79, characterized in that the coupling element (320) is designed and constructed in the fashion of an annular pressure body (321) for frictionally engaging between the belt pulley body (315) and the coupling disk (319) or drive shaft (307).

84. Compressor, in particular of claim 83, characterized in that the pressure body (321) is designed and constructed as bellows, preferably as thin-walled metal bellows.

85. Compressor, in particular of claim 84, characterized in that for a frictional engagement, the

pressure body (321) is filled with a flow medium under a predeterminable pressure.

86. Compressor, in particular of claim 85, characterized in that the flow medium exhibits such a high thermal coefficient of expansion that is opens the pressure body (321) at least in zones, or even causes it to explode, thereby disengaging, when a predetermined temperature is exceeded because of the then prevailing inner pressure.

87. Compressor, in particular of claim 85 or 86, characterized in that the pressure body (321) comprises at least one predetermined mechanical breaking point (322), which is used for relieving pressure and, thus, for disengagement.

88. Compressor, in particular of claim 83 or 84, characterized in that the pressure body (321) comprises a least one safety fuse (323), which is used for relieving pressure and, thus, for disengagement.

89. Compressor, in particular of claims 87 and 88, characterized in that for a disengagement the pressure body (321) comprises at least one predetermined mechanical breaking point (322) serving to relieve pressure, and at least one safety fuse (323) serving to relieve pressure.

90. Compressor, in particular of one of claims 83-89, characterized in that the pressure body (321) extends annularly, preferably in the fashion of a hollow cylinder, between the inside surface of the belt

pulley body (315) and the outside surface of the coupling disk (319) or a vibration damper (318).

91. Compressor, in particular of one of claims 83-90, characterized in that the pressure body (321) comprises a pressure chamber (324) substantially rectangular in its longitudinal section, and adjacent thereto preferably outward directed separating regions (325), which narrow in the longitudinal section relative to the pressure chamber (324).

92. Compressor, in particular of claim 91, characterized in that two opposite separating regions (325) are designed and constructed, which adjoin the pressure chamber (324) with their arms in U-shape in the longitudinal section of the pressure body (321).

93. Compressor, in particular of claim 91 or 92, characterized in that the separating region or regions (325), i.e., the safety fuse (323) and/or the predetermined breaking point (322) extend along the circumference of the pressure body (321), point by point, in zones, or continuously.

INTERNATIONAL SEARCH REPORT

International Application No
PCT/DE 98/02561

A. CLASSIFICATION OF SUBJECT MATTER
IPC 6 F04B27/10

According to International Patent Classification (IPC) or to both national classification and IPC

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)
IPC 6 F04B

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched

Electronic data base consulted during the international search (name of data base and, where practical, search terms used)

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category °	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
X	DE 35 00 298 A (DIESEL KIKI CO) 14 November 1985 see page 7 - page 14; figures 1-5 ---	1-3,13, 26,28, 29,33-43
X	DE 24 15 206 A (BORG WARNER) 24 October 1974 see page 1, line 1 - line 8 see page 5 - page 8; figures 1,2 -----	1-9,13, 43,44

☐ Further documents are listed in the continuation of box C.

☒ Patent family members are listed in annex.

° Special categories of cited documents :

"A" document defining the general state of the art which is not considered to be of particular relevance

"E" earlier document but published on or after the international filing date

"L" document which may throw doubts on priority claim(s) or which is cited to establish the publication date of another citation or other special reason (as specified)

"O" document referring to an oral disclosure, use, exhibition or other means

"P" document published prior to the international filing date but later than the priority date claimed

"T" later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention

"X" document of particular relevance; the claimed invention cannot be considered novel or cannot be considered to involve an inventive step when the document is taken alone

"Y" document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents, such combination being obvious to a person skilled in the art.

"&" document member of the same patent family

Date of the actual completion of the international search	Date of mailing of the international search report
16 February 1999	17.02.99

Name and mailing address of the ISA	Authorized officer
European Patent Office, P.B. 5818 Patentlaan 2 NL - 2280 HV Rijswijk Tel. (+31-70) 340-2040, Tx. 31 651 epo nl, Fax: (+31-70) 340-3016	Jungfer, J

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INTERNATIONAL SEARCH REPORT

International application No.
PCT/DE98/02561

Box I Observations where certain claims were found unsearchable (Continuation of item 1 of first sheet)

This international search report has not been established in respect of certain claims under Article 17(2)(a) for the following reasons:

1. ☐ Claims Nos.:
   because they relate to subject matter not required to be searched by this Authority, namely:

2. ☐ Claims Nos.:
   because they relate to parts of the international application that do not comply with the prescribed requirements to such an extent that no meaningful international search can be carried out, specifically:

3. ☐ Claims Nos.:
   because they are dependent claims and are not drafted in accordance with the second and third sentences of Rule 6.4(a).

Box II Observations where unity of invention is lacking (Continuation of item 2 of first sheet)

This International Searching Authority found multiple inventions in this international application, as follows:

See Supplemental Sheet

1. ☐ As all required additional search fees were timely paid by the applicant, this international search report covers all searchable claims.

2. ☐ As all searchable claims could be searched without effort justifying an additional fee, this Authority did not invite payment of any additional fee.

3. ☐ As only some of the required additional search fees were timely paid by the applicant, this international search report covers only those claims for which fees were paid, specifically claims Nos.:

4. ☐ No required additional search fees were timely paid by the applicant. Consequently, this international search report is restricted to the invention first mentioned in the claims; it is covered by claims Nos.:

Remark on Protest

☐ The additional search fees were accompanied by the applicant's protest.

☐ No protest accompanied the payment of additional search fees.

PCT/ISA/210

The International Searching Authority has found that this international application contains several (groups of) inventions, namely:

1. Claims: 1-13, 21, 26-44, 46

   Drag bearing between the driving plate and the drag body

2. Claim: 14

   Suspension of the pistons on the housing side on the swash plate

3. Claims: 15-20

   Torsional fixing for the pistons in the cylinder head

4. Claims: 22-25, 45

   Drive shaft with a fixed guide arm

5. Claims: 47-73

   Heat insulation of a valve plate and a housing of a compressor for an air conditioning system

6. Claims: 74-93

   Coupling mechanism between the compressor and the drive motor

INTERNATIONAL SEARCH REPORT

Information on patent family members

Inte. .tional Application No
PCT/DE 98/02561

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