DECELERATING SYSTEM FOR TOURING VEHICLES

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Filed: Sept. 16, 1969

Appl. No.: 858,321

Foreign Application Priority Data

Sept. 17, 1968 France..........................166527
Mar. 13, 1969 France..........................6907189
May 20, 1969 France..........................6916401

U.S. Cl.........................................188/296, 192/3 TR
Int. Cl..........................................F16D 57/02
Field of Search.................................188/90, 90 A, 296; 192/3, 3 TR; 138/45, 46

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ABSTRACT

This system is intended for touring vehicles driven at a high speed corresponding to an engine shaft speed above 3,000 r.p.m. and preferably above 5,000 r.p.m. The engine is cooled by forced water circulation in a circuit comprising in series a pump, the engine and a radiator having a high heat dissipation capacity. The system comprises a hydraulic decelerator whose rotor runs permanently at a speed of at least the same order as engine shaft speed and whose diameter is less than 20 centimeters. The decelerator is connected in parallel to a part of the engine cooling circuit by means of a three-way two-position valve which when in one position sends all the water it receives to said part of the circuit isolating the decelerator, while when in its other position the valve sends all the water it receives to the decelerator inlet, isolating said circuit part.

19 Claims, 9 Drawing Figures
Fig. 5.

Fig. 6.

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DECELERATING SYSTEM FOR TOURING VEHICLES

The invention relates to systems adapted to decelerate a moving vehicle by applying a decelerating torque to one of the rotating elements of the transmission line connecting the vehicle engine to the vehicle wheels; amongst such systems, the invention relates more particularly to those which are fitted to touring vehicles adapted to travel at high speeds corresponding to engine shaft speeds above 3,000 r.p.m., preferably above 5,000 r.p.m., and where the engine of the vehicle to be decelerated is normally cooled by forced circulation of water in a circuit comprising in series a pump and the maximum speed attainable by modern touring vehicles increase.

It is conventional for heavy vehicles (trucks and coaches) to be fitted with decelerators which help to decelerate the vehicles without stopping them completely; devices of this kind are useful more particularly on long downgrades where conventional brakes would be likely to overheat. The decelerators operate without solid friction, being either hydraulic or electric (eddy current brakes); their weight is greatly reduced. Unfortunately, devices of this kind, which are usually interposed between the clutch and the rear axle, have been too bulky and costly for use in touring vehicles, the main reason for the bulkiness being the relatively slow speeds of the vehicle's engine.

The conventional brakes used for lightweight and touring vehicles must decelerate the same from any speed right down to a standstill, if required, and they achieve this by solid friction whether they are of the drum or disc kind. Brakes of this kind are very efficient and virtually essential, but at high speeds they heat up rapidly and become unsatisfactory, particularly when applied repeatedly for prolonged periods of time. This disadvantage is becoming increasingly widespread as motorways increase in number and the maximum speeds attainable by modern touring vehicles increase.

It is conventional for heavy vehicles (trucks and coaches) to be fitted with decelerators which help to decelerate the vehicles without stopping them completely; devices of this kind are useful more particularly on long downgrades where conventional brakes would be likely to overheat. The decelerators operate without solid friction, being either hydraulic or electric (eddy current brakes); their weight is greatly reduced. Unfortunately, devices of this kind, which are usually interposed between the clutch and the rear axle, have been too bulky and costly for use in touring vehicles, the main reason for the bulkiness being the relatively slow speeds of the vehicle's engine. The use of such decelerators at low speeds and for long durations, whether they are of the drum or disc kind, would be a disadvantage for the braking of a vehicle travelling at very high speed, for a vehicle travelling at 150 km./h. travels more than 40 m./sec.

More particularly, the cooling water may boil, since the heat-removal capacity of the radiators normally used for heavy vehicles has no wide margins available for use other than merely cooling the engine, more particularly for dissipating heat evolved for a prolonged period of time by a heat source other than the engine.

The applicants have observed first that the very high speeds attainable by the engines of high-speed touring vehicles (engine speeds of more than 3,000 r.p.m. and possibly reaching and frequently exceeding 6,000 r.p.m.) are suitable for producing relatively high braking torques by means of very small hydraulic decelerators provided that the high speeds are applied directly to the decelerator rotors. For instance, at a speed of 6,000 r.p.m. a braking torque of 15 m.kg. can be produced by a hydraulic decelerator having a rotor diameter of as little as 15 cm. Since small decelerators of this kind take up little space—and are therefore of reduced cost—they can be fitted below the hoods of touring vehicles, inter alia at one end of the vehicle engine crankshaft.

The applicants have also found that a system of this kind can with advantage be supplied by the cooling water of the engine, for the two disadvantages mentioned in the foregoing are greatly reduced or even obviated for the following reasons:

Since the internal volume of the decelerator is small, it can be filled rapidly, more particularly if, as will be assumed hereinafter, the entire water flow is forced through it to give a brief response time, and

more particularly, there is no risk of the cooling water boiling since the "brake applications" are much shorter than the periods of running with deceleration," and more particularly because, for a given engine power and given load, the cooling circuits of touring vehicles are devised to remove much more heat—than the cooling circuits of heavy vehicles, for the reason that:

the installed engine capacity per load unit is approximately 5–10 times greater in touring vehicles than in heavy vehicles, and

the use of petrol engines instead of diesel engines and the fact that engine shaft speeds are higher lead to greater heating in touring vehicles.
In short, using a hydraulic decelerator at the very high speeds mentioned provides a number of associated advantages (high torque, small size, low cost, short response time, possibility of using the engine cooling water to supply the decelerator without risk of the water boiling) which make it very attractive to use a system of this kind to brake a high-speed touring vehicle. The system can very readily reduce the high speeds of such vehicles by very effective progressive decelerations down to a medium speed at which the conventional brakes can be used satisfactorily.

The safety provided by this form of deceleration at high speeds is outstanding, particularly since, although it is so effective, there is no risk of skidding, for the decelerating torque provided by a hydraulic decelerator is relatively low at low speeds of rotation and should theoretically drop to zero when the wheels lock; the device therefore has a self-regulating action, the decelerating torque automatically decreasing immediately the wheels lock due to excessive deceleration and vice versa.

FIG. 1 shows an embodiment of a hydraulic decelerator which is of use according to the invention but which does not limit the same. The decelerator comprises a rotor forming a centrifugal pump and comprising a semitoroidal shell 1 whose base is open at a place 1, and which is braced by blades 2 which are radial or inclined in the direction of rotation to increase the braking effect. The decelerator also includes a stator comprising a semitoroidal socketed shell 3 disposed axially opposite the rotor; and a cover 4 around the rotor. The rotor is rigidly secured to a shaft portion 5 disposed at the ends of the engine vehicle shaft, as a rule, the crankshaft; the shaft portion 5 is centered relatively to the engine frame 6 by a roller bearing 7 sealed by two cup seals 8, beyond which the shaft portion 5 otherwise. A rotating gasket 9 sliding on a stationary ring 10 provides sealing-tightness between the rotor and the engine frame 6.

The sockets of the stator shell 3 are designed to collect the liquid streams emitted by the tips of the blades 2 and to inject such streams into the small-diameter zones of the blades 2 so as to produce vortices tending to brake the rotor.

The cover 4 comprises a liquid inlet 11 which is offset from the rotor axis, so as to reduce the overall axial size of the decelerator, and an outlet 12 for the liquid which escapes radially between the blades 2 and the sockets 3 to an annular chamber 13 of the stator. The cover inside surface extends very close to the external profile of the rotor so as to reduce leakages of liquid going directly from the inlet 11 to the chamber 13 without having passed through the toroidal enclosure (2, 3) where the decelerating torque is produced.

The connection between the inlet 11 and such enclosure is by way of passages 2, bounded by the upstream tips of the blades 2 and extending through the open part 1.

A decelerator of this kind does not require accurate machining operations and is rugged and cheap.

To operate the decelerator as and when required, the same is supplied with the entire water flow used to cool the vehicle, such water normally being circulated by a pump in a closed circuit through the engine and a cooling radiator consecutively.

FIG. 2 shows the hydraulic decelerator 14, engine 15 of the vehicle and a radiator 16 cooled by a fan 17. As in conventional cooling circuits the water is moved by a pump 18 seriatim from the engine to the radiator, after heating, through a line 20, and from the radiator to the engine, after cooling, through a line 19. In the present case the line 20 has a three-way two-position valve 21 which in one position completes the line 20 and which in the other position diverts all the flow through the valve to supply a decelerator inlet line 22.

A decelerator outlet line 23 is connected to the line 20 at a place 24 slightly downstream of valve 21. If that section of the line 20 which separates the valve 21 from the place 24 is given the reference 29, it can be stated that the decelerator 14 and its inlet and outlet lines 22, 23 are shunted across the section 29.

To prevent any flowback of water from the place 24 to the decelerator, the line 23 joins the interior of the line 20 in the direction of normal water flow in the line 20. Of course, other means could be used to achieve the same result, such as a non-return valve or a second valve coupled with the first valve, but the suggestion made here is very rugged and economical.

The circuit also comprises: a narrow line 25 connecting the decelerator to the normal cooling circuit at a place 26 disposed upstream of the pump 18, the connection being such that the normal cooling water flow has an aspirator effect on the contents of the line 25, and thus helps to empty the decelerator when the valve 21 is in its normal nonbraking position; and a narrow line 27 connecting the decelerator to the top of an expansion tank 28—if the ordinary cooling circuit is a closed circuit and has an expansion tank—to enable gas from the expansion tank to help drain the decelerator of water and to help fill the decelerator with water by removal from the decelerator of the gas therein to the expansion tank.

A circuit of this kind operates as follows:

When the valve 21 is in the position in which it completes the line 20, the water normally in a closed circuit through the radiator and the engine. The aspirator effects set up at the places 24, 26 empty the decelerator completely so that the same produces no disturbing residual torque.

To decelerate the vehicle, the valve 21 is changed over to the position which is shown in FIG. 2 and in which all the cooling water goes through the decelerator so that the same is connected in series in the normal cooling circuit.

Most of the water leaving the decelerator then passes through the radiator and is cooled therein, but the water flowing through the line 25 is not cooled in this way. This is not a disadvantage in practice since the latter flow, although ensuring rapid decelerator emptying, is very small in relation to the total flow.

A control circuit of this kind has many advantages for small high-speed decelerators used in light vehicles. More particularly, the control response time is very short, for when the valve 21 is placed in its operative position the delivery from the circulating pump is compelled to flow through the decelerator, in contrast to systems in which the decelerator is just connected in parallel to the cooling circuit. Also, the cooling water delivery is maximum since it is driven not just by the ordinary circulating pump but also by the decelerator which is devised as a centrifugal pump. This maximum delivery (e.g. of the order of from 80 to 100 liters/min.) always achieves production of the maximum decelerating torque corresponding to engine speed at the particular time concerned and maximum heat removal. In other words, when a brief application is made at high speed, the resulting decelerating torque is not limited by water temperature, because of the heat inertia of the total volume of circulating water, and, in prolonged deceleration the decelerating torque is limited only by the heat removal capacity of the radiator and, as already stated, this capacity is very high in the case of touring vehicle radiators.

The valve 21 can be operated either by being directly coupled to a special control pedal or lever or to the conventional brake pedal or accelerator pedal. Unfortunately, connections of this kind are difficult to embody and unreliable in operation; preferably, therefore, the power source for this control, in the preferred case in which the vehicle engine is an internal combustion engine, is the negative pressure in the engine induction pipe downstream of the throttle member or butterfly in such pipe, the bringing of this negative pressure into operation being dependent on a driver-operated control. Consequently, the decelerator comes into operation only when permitted to do so deliberately by the driver and when the negative pressure has a high enough absolute value, a factor presupposing a reduced fuel supply to the engine (butterfly valve closed completely or almost completely) and a high enough vehicle speed. Clearly, therefore, this particular form of control is very suitable for the purposes of the invention, since the main requirement is that the control be satisfactory at very high speeds.
A control of this kind is shown in FIGS. 3 and 4, where valve 21 is controlled by a system 31 responding to the negative pressure in the engine induction pipe 51 at a place 53 downstream of throttle member 52, communication between systems 31 and pipe 51 being by way of means 32 under the driver's control. More particularly, the system 31 is such as to cut in the decelerator, under the control of the means 32, only when the absolute value of such negative pressure exceeds a predetermined threshold corresponding to closure of the butterfly 52 and to an engine speed above a predetermined threshold.

In the embodiment shown in FIGS. 3 and 4, the three-way two-position valve 21 has a chamber 34 in permanent communication with the water supply line 20, the lines 29, 22 terminating in chamber 34 by way of coaxial seats 35, 36 respectively, the seats cooperating with respective lids 37, 38 mounted on a single rod 39.

The system 31 comprises a variable-volume chamber 40 bounded by a diaphragm 41 (or other moving or deformable member adapted to close the chamber 40 hermatically) connected to rod 39. A cover 42 crimped to the casing of chamber 40 clamps the periphery of diaphragm 41 and is formed with an aperture 43 via which atmospheric pressure is operative on that side of diaphragm 41 which is remote from chamber 40. A spring 50 acting on rod 39 and/or diaphragm 41 opposes the action of the negative pressure in chamber 40.

The means 32, which comprise a nonreturn valve 44 or some other moving closure member, are disposed in a line 45 adapted, when the valve 44 is open, to connect chamber 40 to place 53 (the direction of flow in induction pipe 51 is diagrammatically represented by an arrow in each of FIGS. 3 and 4).

The driver-controlled means 32 can be either entirely independent of any other vehicle control or can be subject to some other control such as the accelerator pedal operating the butterfly 52, the conventional brake pedal or the clutch pedal. In the case of accelerator pedal control, the means 32 can be such that the driver can operate the valve 44 directly by hand (or foot), e.g., via a push button (not shown) placed directly on the vehicle steering wheel (in the commonest case of a road vehicle), so that the driver does not have to release the wheel to operate the valve 44.

Preferably, however, the means 32 are so devised in both the cases hereinafter set forth so as to be operable by a solenoid valve in the form of assembly of an electromagnet 46 which operates valve 44 against the force of a return spring 47. Preferably, the circuit arrangement is such that the electromagnet 46, when energized by a power supply 48 via a contactor 49, opens the valve 44 and therefore connects chamber 40 to the place 53 in the engine induction pipe 51, whereas the spring 47 tends to isolate the chamber 40 when contactor 49 de-energizes electromagnet 46.

In the case of independently operated control means 32, the contactor 49 can be placed on the vehicle instrument panel so that the driver can permit or override operation of the electromagnet 46 and therefore of the decelerator.

In the case of control 32 subject to some other control of the vehicle, a first suggestion is to embody the contactor 49 by a microswitch disposed near the accelerator pedal and adapted to energize electromagnet 46 only when the accelerator pedal is in a position corresponding to minimum opening of butterfly 52. Consequently, when the vehicle is running on level ground or down a slight downgrade, the driver retains the possibility of lifting his foot operating the accelerator pedal almost completely without cutting in the decelerator, for the contactor 49 is not then operated by the pedal, and so valve 44 stays closed and prevents the negative pressure from reaching the diaphragm 41. However, if the driver releases the accelerator pedal completely and so decelerates the vehicle, the spring 47 is made continuous provided that engine speed is high enough for the negative pressure reaching the diaphragm 41 to be sufficient to overcome the force of the spring 50.

A second suggestion is to embody the contactor 49 by a microswitch placed near the ordinary brake pedal so that the contactor allows electromagnet 46 to be energized only when the driver operates the brake pedal, preferably during the dead part of the brake pedal travel prior to application of the ordinary brake. This is advantageous in the case of a vehicle in which the decelerator is required to operate only when the driver operates the brake pedal, inter alia for town driving so that gear-changing operations are not disturbed.

A third suggestion is to embody the contactor 49 as a microswitch controlled by the clutch pedal and adapted to deenergize the electromagnet 46 and therefore prevent the decelerator from operating, when the driver operates the clutch pedal.

At least two of the controls just outlined for energizing the electromagnet can be combined to form the means 49. Advantageously, one such mixed control 49 comprises a first single control in the form first contactor connected to the accelerator pedal. The second single control can comprise a second contactor in series with the first contactor and controlled, as previously mentioned, by the clutch pedal; this ensures that when the driver lifts his foot off the accelerator for gear changing, the decelerator does not operate and gear changing is not impaired by the normal deceleration, more particularly in the case of changing up, although due to the slight delay of the accelerators in responding this deceleration is barely perceptible.

Other improvements in or relating to the decelerating facilities hereinbefore described will now be described, since it may be useful:

to further shorten response time—i.e., the time between operation of the three-way valve and production of an appreciable decelerating effect (this time is normally about 1 sec. in systems of the kind of interest here)

to arrange for the decelerators to be considerably effective even at low speeds, since in the absence of any special action the braking torque output by a hydraulic decelerator decreases considerably when the speed of rotation of its rotor and its internal pressure decrease.

In the present case in which the decelerator rotor and the circulating pump are engine-driven, the pressure in the hydraulic cooling circuit (19, 22, 23, 20) decreases with engine speed, and so the decelerator braking torque decreases very considerably when engine speed decreases. One way of increasing the torque at low speeds would be to increase the pressure in the hydraulic circuit by the provision of a constriction at the decelerator outlet. Unfortunately, this step would systematically increase pressure even at high speeds, with the risk of producing decelerating torques in excess of the limits of adhesion of the brake wheels or the limits of clutch slip at high speeds.

Consequently, a constriction 54 is provided at the output of the decelerator 14 and is so devised that the opening cross section which it presents to the flow of liquid varies automatically, continuously or intermittently, in the same sense as the pressure of the liquid or—which comes to the same thing—as the rate of flow of such liquid once the operating condition has been established. Consequently, when there is no liquid in the decelerator the constriction has a minimum opening cross section, possibly zero opening cross section, and it fills very rapidly immediately after the corresponding actuation via the three-way valve; it is found that the presence of this downstream constriction can readily double the speed of decelerator filling.

Also, the presence of the constriction increases the pressure of the volume of liquid disposed immediately upstream of the constriction—i.e., in the decelerator—in proportion as the opening cross section of the constriction is smaller; consequently, at high speeds, corresponding to a large opening cross section of the constriction, there is no risk of an excessive decelerating torque which might lock the vehicle wheels.
also, the decelerating torque produced at relatively low speeds, corresponding to a relatively small flow cross section through the constriction, is increased appreciably to an extent such that it becomes perceptible and effective, so that the factory works over a wide range of speeds and not just at very high speeds.

To show the usefulness of automatic variation of the constriction cross section, it can be stated that, with a constriction whose cross section remains at 65 mm², the decelerating torque drops from 15 to 0.8 m.kg. when the speed of decelerator rotor rotation drops from 5,000 to 2,500 r.p.m., ceteris paribus, whereas if the constriction cross section decreases from 65 to 20 mm² approximately during the same time that the speed drops from 5,000 to 2,500 r.p.m., the braking torque decreases only from 15 to 9.1 m.kg. in the same conditions.

These results can be gathered from the graph which is shown in FIG. 5 and in which the braking torque C expressed in m.kg. is plotted along the ordinates and the speed V in r.p.m. is plotted along the abscissae. Curve 55 of the graph shows variations of the torque C in dependence upon speed V for a fixed constriction cross section of 65 mm² (circular orifice of approximately 9 mm diameter) at the decelerator outlet; it can be seen in this case that the torque drops from about 15 m.kg. (point A) to less than 1 m.kg. (point B) for a speed reduction from 5,000 to 2,500 r.p.m. Curve 56 corresponds to a fixed constriction cross-section of 7 mm² (circular orifice of 3 mm diameter); in this case, the torque clearly becomes excessive at speeds above 3,500 r.p.m. Curve 57 corresponds to a variable constriction according to the invention whose cross section decreases from 65 mm² to about 20 mm² (circular orifice 5 mm in diameter) simultaneously as the speed drops from 5,000 to 2,500 r.p.m.; this curve shows clearly that the torque is still appreciable at 2,500 r.p.m., since it is still above 9 m.kg. (point C).

In more general terms, the system fitted with the variable constriction operates as follows:

At low engine speeds the delivery of water from the circulating pump is small and the opening of the constriction is at a minimum. As engine speed increases, the water delivery increases too and so, unless the opening of the constriction increased simultaneously so as to keep the pressure substantially constant, the pressure would tend to increase. An automatic regulation which largely compensates for speed-dependent torque variations is therefore provided.

Another considerable advantage of having a variable constriction at the decelerator outlet is that it reduces the decelerator response time; when the decelerator is out of operation the water pressure therein is zero and the cross section of the constriction at the decelerator outlet is minimum, and so when the three-way valve is changed over to energize the decelerator, the same fills very rapidly and the discharge orifice opens gradually as filling proceeds.

This result can be seen in the graph which is shown in FIG. 6 and in which the decelerating torque C expressed in percent- ages of its maximum operating value is plotted along the ordinates and the response time r in seconds calculated from the time at which the three-way valve is operated is plotted along the abscissae. The two curves 58, 59 both correspond to the setting up of a rated or operating torque corresponding to 4,000 r.p.m., curve 58 being for a constant cross section of the decelerator outlet while curve 59 is for a downstream constriction having an automatically varying cross section. Clearly, after 0.4 sec. the decelerating torque has risen to 16 percent of its rated value in the first case (point D) and to 40 percent of its rated value in the second case (point E)—a considerable advantage in cases in which the facility is required to decelerate a vehicle travelling at very high speed, since a fast-moving vehicle travels a considerable distance in half a second.

Another advantage is that increasing the constriction cross section helps to increase the water delivery and simultaneously to increase the rated torque and therefore the braking power, so that water delivery is adapted to required power at all engine speeds. This feature cuts out temperature variations due to operation of the decelerator, an advantage for the engine.

The constrictions having an automatically variable cross section as hereinbefore described can be embodied in any appropriate fashion, for instance, as a calibrated-spring non-return valve or as a pivoted flap biased resiliently towards its closed position and so devised that the liquid pressure on the upstream side tends to open it. Another possible form for such a constriction is a sliding lid whose position can vary in dependence upon the upstream pressure, the same acting on the lid through an appropriate sampling line; alternatively, the opening of the constriction could be controlled directly not by the liquid pressure but by some other parameter, such as engine speed, varying in the same sense as the liquid pressure.

Very advantageously, however, the constriction is embodied by means of a resilient diaphragm 60 (FIG. 7) made of rubber or some other elastomeric substance and pierced with a calibrated orifice 61. Preferably, to facilitate the deformation of such a diaphragm, the edge of the calibrated orifice has the general shape of a nozzle converging towards its downstream end. The elastomeric substance must be able to undergo considerable stretch so that the orifice diameter can vary in operation from its ordinary size to three times its ordinary size or even more. The material must be temperature-resistant (often, the temperature is near the temperature of boiling water) and must be able to withstand chemical attack by the cooling liquid and must be non-tearing.

Conveniently, the decelerating systems hereinbefore described can be adjustable instead of being just two-step action devices, for even if the vehicle is travelling very fast the driver may require only a relatively small decelerating torque, for instance, sufficient to keep vehicle speed constant on a downgrade or to brake the vehicle very gently and gradually.

Accordingly, the variable constriction comprises a number—preferably two—of constrictions respectively associated with lines connected in parallel to the decelerator outlet, all the lines except one being adapted to be made inoperative by appropriate valves.

The advantage of such a feature will be readily apparent; the pressure in the decelerator is smaller in proportion as the total cross-section, and therefore the number, of the constrictions offering a passage to the liquid leaving the decelerator is larger.

Consequently, such pressure—and the corresponding decelerating torque—can be controlled as required by varying the number of constrictions in operation; more particularly, if there are two constrictions, the driver can choose between a first relatively gentle deceleration, similar to the deceleration provided by engine braking, when the two constrictions in parallel are used, and a second stronger deceleration, for which only one of the two constrictions is used, the other being rendered inoperative by closure of a valve.

The constrictions can have opening cross sections which are either fixed or automatically variable in the manner hereinbefore described; in the case of fixed cross sections, the same regulating effect as previously provided continuously by a single variable constriction is obtained but intermittently, the bringing into operation of an increasing number of fixed-section constrictions having the same result as the progressive opening of a single constriction—i.e., increased water circulation.

The various constrictions can be cut into and out of operation by any appropriate mechanical, electrical, pneumatic or hydraulic means.

The embodiment shown in FIGS. 8 and 9 comprises two constrictions 54, 56, connected in parallel to two lines 62, 63 respectively, the line 62 forming a part of the line 23 and the line 63 being closable by a valve 64. It is assumed in this embodiment that the engine is an internal combustion engine and the power source for operating the valve 64 is the negative pressure in the engine induction pipe 51 at the place 53 downstream of the throttle valve 52; this negative pressure can
close valve 64 by attracting a diaphragm 65 connected to valve 64 against the force of a return spring 66.

As previously, such negative pressure is also used to operate the valve 21 and is in fact used to operate the desired member (valve 21 or valve 64) only when the driver expresses the deliberate intention for this operation, inter alia by electrically energizing a solenoid valve (32, 67) disposed in a line (45, 68) for connecting the plate 53 to the required member.

Energization can be achieved very simply by placing the handle 69 (FIG. 9) of a switch to the appropriate position; the three positions diagrammatically shown as a, b and c in FIG. 9 for the handle 69 correspond to zero decelerating torque, to a reduced decelerating torque (energization only of solenoid valve 32, corresponding to operation of the decelerator and two constrictions 54, 54a) and maximum deceleration torque (energization of the two solenoid valves 32, 67, corresponding to operation of the decelerator with closure of valve 64—i.e., use solely of construction 54a), respectively.

In a variant the two solenoid valves can be controlled by different members, the first being, with advantage, energized just by release of the accelerator pedal and the second being energized by initiation of operation of the brake pedal.

Clearly, and as the foregoing shows, the invention is not limited to those of its embodiments and uses which have been more particularly described, but covers all variants.

I claim:

1. A system for decelerating a touring vehicle driven at a high speed corresponding to an engine shaft speed above 3,000 r.p.m., the vehicle engine normally being cooled by forced liquid circulation in a circuit comprising in series a pump, the vehicle engine and a radiator having a high heat dissipation capacity, wherein the system comprises: a hydraulic decelerator having a rotor which is adapted to be connected to the engine shaft so as to run permanently at a speed at least of the same order as engine shaft speed, the diameter of said rotor being less than 20 cm., inlet and outlet means connecting the decelerator in parallel to a part of the engine cooling circuit, said means comprising upstream of said part of said engine cooling circuit a three-way two-position valve arranged when in one position to send all the liquid it receives to said part of the circuit, isolating the decelerator, and when in its other position to send all the liquid it receives to the inlet means, isolating said circuit part, and a constriction means in the outlet means for automatically varying the flow cross section which it presents to the liquid in the same sense as the pressure of the liquid.

2. A system according to claim 1, including control means for the three-way valve arranged to be automatically operated by the beginning of the instinctive movement of the right foot of the driver at the moment when he wishes to decelerate.

3. A system according to claim 2, wherein the control means are operated by the end of the release of the accelerator pedal.

4. A system according to claim 2, wherein the control means are operated by the beginning of the depression of the brake pedal.

5. A system according to claim 1, wherein said engine shaft speed is above 5,000 r.p.m.

6. A system according to claim 1, wherein said rotor-diameter is approximately 15 cm.

7. A system according to claim 1, the vehicle engine being an internal combustion engine including an induction pipe with an adjustable throttle member therein, said system further comprising power means for deriving power from the negative pressure in the engine induction pipe downstream of the adjustable throttle member wherein said throttle member is at least partly closed, and for using this derived power to operate the three-way valve.

8. A touring vehicle comprising a decelerating system according to claim 1.

9. A system as set forth in claim 1, the vehicle engine being an internal combustion engine including an induction pipe with an adjustable throttle member therein, said system further comprising power means for deriving power from the negative pressure in the engine induction pipe downstream of the adjustable throttle member therein when said throttle member is at least partly closed, and for using this derived power to operate the three-way valve; and driver-controlled means for connecting and disconnecting said power means to the induction pipe downstream of the adjustable throttle member wherein said throttle member is at least partly closed in order to bring said power means into and out of service respectively.

10. A system as set forth in claim 9, wherein said power means comprise a chamber connected via a line to a place in the engine induction pipe downstream of the throttle member, and a movable member mounted in said chamber to be moved by the pressure in said chamber, said movable member being operatively connected to said three-way valve; and said driver-controlled means comprise a driver-controlled solenoid valve disposed in said line.

11. A system as set forth in claim 10, wherein the driver-controlled means comprise a switch which is accessible to the driver and connected in the energizing circuit for the solenoid valve.

12. A system as set forth in claim 11, wherein the driver-controlled means comprise an electric switch which is connected in the energizing circuit for the solenoid valve and whose operation is controlled by a driver control different from said previously mentioned switch which is accessible to the driver.

13. A system as set forth in claim 12, wherein the driver-controlled means comprise at least two switches connected to two different control actions and connected in series in the energizing circuit for the solenoid valve.

14. A system as set forth in claim 1, wherein at least some of the last-mentioned part of the decelerator cooling circuit comprises a number of parallel-connected sections each having a constriction means and, means being provided for selectively closing and opening all the various sections except one.

15. A system as set forth in claim 14, wherein the number of parallel-connected sections is two.

16. A system as set forth in claim 15, the vehicle engine being an internal combustion engine having an induction pipe containing an adjustable throttle member wherein one of the two parallel-connected sections has a nonreturn valve adapted to be operated, under driver control, by the negative pressure which exists in the engine induction pipe downstream of the adjustable throttle member therein.

17. A system as set forth in claim 14, wherein the constriction means is embodied by a diaphragm in the form of an orifice in a membrane made of elastomer.

18. A system as set forth in claim 17, wherein the edge of the orifice has the shape of a nozzle converging in the downstream direction.

19. A system as set forth in claim 1, wherein control of the automatic variations of the passage cross section of the constriction is provided directly by the liquid pressure.