(12) INTERNATIONAL APPLICATION PUBLISHED UNDER THE PATENT COOPERATION TREATY (PCT)

(19) World Intellectual Property Organization
International Bureau

(43) International Publication Date
22 April 2010 (22.04.2010)

PCT

(10) International Publication Number
WO 2010/043301 A1

(51) International Patent Classification:
F16F 15/134 (2006.01)

(21) International Application Number:
PCT/EP2009/006898

(22) International Filing Date:
24 September 2009 (24.09.2009)

(25) Filing Language: English

(26) Publication Language: English

(30) Priority Data:
61/196,541 17 October 2008 (17.10.2008) US

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(81) Designated States *(unless otherwise indicated, for every kind of national protection available)*: AE, AG, AL, AM, AO, AT, AU, AZ, BA, BB, BG, BH, BR, BW, BY, BZ, CA, CH, CL, CN, CO, CR, CU, CZ, DE, DK, DM, DO, DZ, EC, EE, EG, ES, FI, GB, GD, GE, GH, GM, GT, HN, HR, HU, ID, IL, IN, IS, JP, KE, KG, KM, KN, KP, KR, KZ, LA, LC, LK, LR, LS, LT, LU, LY, MA, MD, ME, MG, MK, MN, MW, MX, MY, MZ, NA, NG, NI, NO, NZ, OM, PE, PG, PH, PL, PT, RO, RS, RU, SC, SD, SE, SG, SK, SL, SM, ST, SV, SY, TJ, TM, TN, TR, TT, TZ, UA, UG, US, UZ, VC, VN, ZA, ZM, ZW.

(84) Designated States *(unless otherwise indicated, for every kind of regional protection available)*: ARIPO (BW, GH, GM, KE, LS, MW, MZ, NA, SD, SL, SZ, TZ, UG, ZM, ZW), Eurasian (AM, AZ, BY, KG, KZ, MD, RU, TJ, TM), European (AT, BE, BG, CH, CY, CZ, DE, DK, EE, ES, FI, FR, GB, GR, HR, HU, IE, IS, IT, LT, LU, LV, MC, MK, MT, NL, NO, PL, PT, RO, SE, SI, SK, SM, TR), OAPI (BF, BJ, CF, CG, CI, CM, GA, GN, GQ, GW, ML, MR, NE, SN, TD, TG).

Published:

— *with international search report (Art. 21(3))*

— *before the expiration of the time limit for amending the claims and to be republished in the event of receipt of amendments (Rule 48.2(h))*

(54) **Title**: DOUBLE PATH TORSIONAL DAMPER

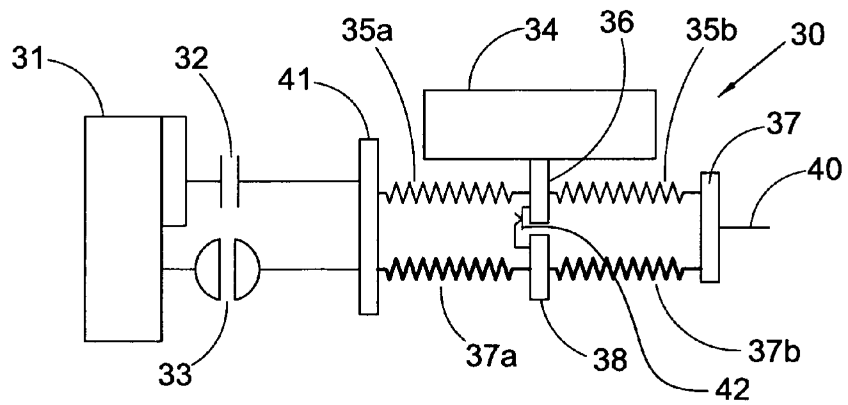

Fig. 3

(57) **Abstract**: The present invention is a double path torsion isolator (30) for use in conjunction with a torque converter. Each of the two vibration paths (35a, 35b, 36; 37a, 37b, 38) includes a flange (36, 38) with a spring connection to each of the torque converter cover (41) and an output connection (37) attached to the transmission shaft (40). The torque converter turbine (34) is connected to one of the flanges (36) to provide a different frequency to the associated vibration path (35a, 35b, 36). When the lockup clutch(32) is engaged, the engine vibration is divided along the two vibration paths toward the common output connection (37). The vibration frequency of the paths (35a, 35b, 36; 37a, 37b, 38) are adjusted so that the frequencies of the two paths (35a, 35b, 36; 37a, 37b, 38) are 180° out of phase at the output connection providing (37) a vibration cancellation effect to the output connection (37).

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Double path torsional damper

[0001] The present invention relates to the field of vehicle drive trains. More specifically, the invention relates to reduction of the conduction of engine vibration to the vehicle transmission and more particularly to the dampening of engine vibration through the torque converter.

[0002] Clutches are used to provide a mechanical by-pass for the hydrodynamic coupling in torque converters in order to improve fuel economy in vehicles. The clutch is engaged and stays locked as soon as torque multiplication is no longer required. During clutch engagement, the engine vibration will be transferred via the clutch to the drivetrain which causes excess wear on components of the drivetrain as well as passenger discomfort. To reduce the transmitted torsional vibrations, torsional isolators or dampers are placed in clutches between the engine output torque and the transmission input shaft. Torsional dampers typically comprise an arrangement of springs and friction plates serving as an elastic member to reduce the engine vibration amplitude. To isolate today's engine with more and more power, many complex spring arrangements are being designed. However, they all can introduce extra undesirable resonant frequencies that cause or increase unwanted vibrations.

[0003] The problem of undesirable resonant frequencies is often addressed by using a plurality of series torsional isolators in torque converters. This trend is also driven by the push for improved fuel consumption and reduced lugging limits. To maintain or improve torsional isolation, lower spring rates are required. Cylinder shutoff applications and a general trend toward a decreased number of engine cylinders further enforce the need for lower spring rate (increased isolator spring volume). Due to envelope constraints, the most efficient way to increase spring volume is to connect two concentric rows of springs together in series (hence the so called series isolator). This arrangement provides the maximum spring volume for the typical envelope available for the torsional damper.

[0004] This arrangement requires the use of one or more plates to serve as a torsional connection between the outer and inner row of springs (the so called "floating flange"). Unfortunately, the inertia of floating flange(s) is substantial for any concentric spring series arrangement. This inertia introduces an addition degree of freedom to the torsional drivetrain system that causes an objectionable vibration in the vehicle. One method of overcoming the additional degree of freedom is to introduce a friction package across one of the concentric spring packages to absorb the energy of the flange mode. However, this approach has the distinct disadvantage of degrading the isolation the damper provides at all frequencies other than the flange mode.

[0005] Thus, a problem exists in the field concerning the reduction of vibration produced in the engine that is transmitted through the drive train when a torque converter clutch is engaged with the vehicle engine.

[0006] The present invention broadly comprises a double path torsional vibration damper for a torque converter having a lockup clutch, the isolator comprising a first flange and a second flange, with the first flange with the first flange attached to the turbine of the torque converter, a first pair of springs, the first one of the first pair of springs extending between an outer hub of the torque converter and the first flange and the second one of the first pair of springs extending from the first flange to an output connection, the output connection operatively connected to a transmission input drive. The invention also comprises a second pair of springs, with the first one of the second pair of springs extending between the outer hub and the second flange and the second one of the second pair of springs extending from the second flange to the output connection. The first pair of springs forms a first vibration path including the outer hub, the first pair springs, the first flange and the output connection and the second pair of springs forms a second vibration path including the outer hub, the second pair of springs, the second flange and the output connection. In a preferred embodiment, the spring rate of the first pair of springs is less than or equal to the spring rate of the second pair of springs. In a more preferred embodiment, the spring rate of each of the vibration paths is tunable (adjustable). In some embodiments, the outer hub may be the torque converter cover.

[0007] One object of the invention is to provide a device for reducing the transmission of engine vibration when a torque converter clutch is engaged.

[0008] A second object of the invention is to supply a device that divides the vibration path between the engine and the transmission.

[0009] A third object of the invention is to provide a vibration reducing device that is adjustable for various sized drive train components.

[0010] A fourth object of the invention is to provide a vibration reducing device in which two vibration paths are adjustable so that the frequency of one path is 180 degrees out of phase with the other path when the vibration waves reach the ends of the paths.

[0011] The nature and mode of the operation of the present invention will now be more fully described in the following detailed description of the invention taken with the accompanying drawing Figures, in which:

[0012] Figure 1 is a schematic view of a single path torsional vibration damper;

[0013] Figure 2 is a schematic view of a single path vibration damper in which a single flange is attached to a torque converter turbine;

[0014] Figure 3 is a schematic of the double path torsional damper of the present invention;

[0015] Figure 4 is a schematic drawing of the double path damper showing changes in the amplitude in vibration along the vibration paths;

[0016] Figure 5A is a cross section view of double path damper of the present invention showing the structural relation of its component parts;

[0017] Figure 5B is a side perspective of the two flanges of the double path damper of the present invention;

[0018] Figure 6 graphically portrays the damper characteristic for the V/4-V/8 engine.

[0019] Figure 7 is a graph of the damper effects of the double path damper at different spring rate ratios;

[0020] Figure 8 is a graph comparing the damper effects of the double path damper of the present invention with prior art dampers; and,

[0021] Figure 9 is a schematic view of an alternate embodiment of the double path damper of the present invention.

[0022] At the outset, it should be appreciated that like drawing numbers on different drawing views identify identical structural elements of the invention. It also should be appreciated that figure orientations, proportions, and angles are not always to scale in order to clearly portray the attributes of the present invention.

[0023] While the present invention is described with respect to what is presently considered to be the preferred embodiments, it is understood that the invention is not limited to the disclosed embodiments. The present invention is intended to cover various modifications and equivalent arrangements included within the spirit and scope of the appended claims.

[0024] Adverting to the drawings, Figure 1 is a schematic view of a single path torsional vibration damper **10**, typical of the prior art. Engine **11** is operatively connected to turbine **14** through clutch **12** and fluid coupling **13**. By operatively connected is meant that a component or device is connected either directly or indirectly to a second component and causes that second component to function. For example, clutch **12** and fluid coupling **13** separately operatively connect engine **11** to turbine **14** as both act to transmit the movement of the engine (or engine crankshaft) to turbine **14**. The assembly of outer spring **15**, floating flange **16**, and inner spring **17**, is connected to isolator hub **18** which acts as an output connection to transmission input shaft **19**. It will be recognized that outer spring **15** and inner spring **17** may be reversed and that flanges in the various embodiments of the dampers discussed *infra*, are one form of inertial elements that may form vibration pathways.

[0025] Figure 2 is a schematic view of a series double damper **20** found in the prior art in which a single flange **26** is attached to turbine **24**. Engine **21** is attached to hub **29** through clutch **22** and fluid coupling **23**. Flange **26** is connected to turbine **24**. Outer spring **25a** and inner spring **25b** connect flange **26** to hub **29** and isolator hub **27**. Similar to the embodiment seen in Figure 1, isolator hub **27** acts as an output connection to transmission input shaft **28**.

[0026] Figure 3 is a schematic of the double path torsional damper **30** of the present invention. Engine **31** is operatively attached to outer hub 41 through clutch **32** and fluid coupling **33**. It is well known that fluid coupling **33** of damper **30** transmits rotational motion from engine **31** through the pump (not shown) and turbine **34** of the associated torque converter, while clutch **32** transmits this rotational motion when engaged with engine **31**. Engine vibration will also be transmitted though clutch **32**. Clutch **32** is operatively connected to outer hub **41**.

[0027] Flange **36** is attached to turbine **34**. A pair of springs **35a** and **35b** attach flange **36** to outer hub **41** and isolator hub **39**, respectively. Together springs **35a**, **35b**, and flange **36** form a first vibration path connecting outer hub **41** and isolator hub **37**. A second pair of springs **37a** and **37b** attach flange **38** to outer hub **41** and isolator hub **37**, respectively. Together springs **37a**, **37b**, and flange **38** form a second vibration path connecting outer hub **41** and isolator hub **37**. In a preferred embodiment, the first vibration path and the second vibration path are substantially parallel. It should be noted that both the first and second vibration paths begin on the outer hub **41** and terminate on isolator hub **37**.

[0028] Figure 5A is a cross section view of double path damper **30** showing the structural relation of outer hub **41**, flanges **36** and **38**, springs **35** and **37**, and output connection **39**. Figure 5B is a side perspective view of one possible structure of flanges **36** and **38**.

[0029] Figure 7 is a schematic view of double path damper **30** adapted for use without attaching a torque converter turbine to a flange in one of the vibratio pathways. Clutch **32** provides the operative connection of damper **30** to engine **31**. In contrast to the embodiment seen in Figure 3, the natural frequency of the upper pathway is made lower than that of the lower pathway by adding mass to flange **36a** rather than attachment to a torque converter turbine. Such an embodiment may be used with drivetrains with manual transmissions which do not normally utilize a torque converter. In addition, in this embodiment, first cover **41** and second cover **37** may be used to flank or surround the springs and flanges of damper **30**.

[0030] Figure 4 is a schematic drawing of the double path damper **30** of the present invention. Vibration path **A** extends from the engine and clutch (not shown in Figure 4) to output hub **41**. Vibration path **A** is divided into two parallel vibration paths **B** and **C** respectively including the two separate intermediate flanges **36** and **38**, respectively. Additional mass, for example the mass of turbine **34,** is attached to flange **36** to lower the natural frequency of that flange. Additional mass may be added by increasing the size of inertial element or flange **36** of pathway **A**. This creates two separate natural frequencies for each vibration pathway **B** and **C** rather than one single frequency. These two separate natural frequencies can be selected or "customized" for a specific drivetrain by determining the spring rate distribution between paths **B** and **C** as well as adjusting the inertia of the flanges individually based on the specific elastic elements and inertial elements used to form the vibration pathways. Examples of elastic elements include rubber slugs, compression springs, and ball ramps combine with diaphragm springs. The original single vibration **A** from the engine enters into the two parallel vibration paths **B** and **C** each with the same wave phases but different amplitudes. During engine speed ramp up or acceleration, when the engine vibration **A** having a frequency (ω) reaches to the path **B** (turbine+flange) having a natural frequency ($\omega_{nt,}$), only the vibration phase in turbine+flange path $\mathbf{B_2}$ will be shifted 180° opposite of the original wave. The natural frequency of the second flange (path **C**) is much higher than for path **A**. Therefore, the vibration wave continues to pass along vibration path **C** without any shift in the phase. Because the frequencies of paths **B** and **C** can be determined based on the specific components chosen for a particular vibration path, they can be adjusted so that the two vibration waves $\mathbf{B_2}$ and **C** reach output hub **39** 180° out of phase with each other, producing a vibration cancellation or reduction effect. In other words when the turbine+flange path $\mathbf{B_2}$ starts to resonate 180° out of phase with the original path **C**, it serves somewhat like a mass absorber. This results in an output vibration **D** having a much lower amplitude.

[0031] The inertia of a floating flange in a series torsional isolator is substantial for any concentric spring series arrangement as seen in Figure 1. This inertia introduces an additional degree of freedom to the torsional drivetrain system (flange mode). Since the spring rate and the flange inertia are dictated by the design requirement of the flange – spring assembly, the natural frequency of the flange itself cannot be adjusted. The construction of a double path damper **30** with two parallel vibration paths **B** and **C** with two separate series spring arrangements **35** and **37,** respectively, provides two individual flanges **36** and **38** with two natural frequencies instead of one. The effective spring rate (K) in a parallel configuration is $K=K_1+K_2$, so by variably distributing the spring rate between these two paths one path can become softer as the other path becomes stiffer. The flange with softer spring rate (flange **36**) will have a lower natural frequency while flange **38** with stiffer spring rate would have a higher natural frequency than the natural frequency from an isolator having one vibration path. In addition, if more inertia is added to flange **36**, the natural frequency of flange **36** will decrease further. Since the turbine in a torque converter has no acceleration or ramp-up function when the clutch is locked to the engine, the inertia of the turbine **34** can be added to the inertia of flange **36** when it is attached to it, which further lowers the natural frequency of flange **36.** (See Figures 3 and 4 for a schematic construction of the attached turbine.) Figure 8 displays a graph of the comparison between a prior art Series Turbine Damper (STD **10**) (see in Figure 1), a prior art Series Double Damper (SDD **20**) (see in Figure 2), and the Double Path Damper (DPD **30**) with variant distribution spring rate of the present invention (see in Figure 3).

[0032] For an engine with a deactivation system (V4-V8 – four cylinders are shut down when idling) a series damper with characteristic like Figure 6 is needed. Figure 6 graphically portrays the damper characteristic for the V/4-V/8 engine. During the V4 mode, the engine is running in the first stage and during the V8 mode the engine runs in the second stage. For a conventional series damper (Figure 1) we need a spring rate of 18.25 Nm/° for the first stage and 53.69 Nm/° for the second stage. The only results shown are for the V4 mode, which is the worst case, in which Turbine Inertia =0.03 kgm2/rad, Floating flange Inertia =0.0135 kgm2/rad, and Isolator Hub Inertia = 0.002 kgm2/rad.

[0033] The simulation results for two conventional dampers are shown in Figure 8. Series Double Damper **20** has a damper mode at about 1500rpm and Series Turbine Damper **10** has an inner flange mode at about 2500rpm. In V8 mode the turbine mode in SDD **20** is about 750rpm and inner flange mode in STD **10** is about 1250rpm. Therefore both prior art series dampers have a resonance made at about 1200~1500rpm either in V8 or V4 mode which is well within the normal driving conditions for several types of vehicles. Due to design limitations, these resonance frequencies cannot be shifted to a less critical region. However if the vibration path is divided along two parallel paths and split the floating flange inertia to half (0.0135/2) and attach the turbine mass to one of the flanges as in the DPD **30** (Figure 3), two separate natural frequencies are formed. Since it is preferred to keep the damper characteristic the same as the original design, the spring rate is distributed in two different paths. Different trials were used to determine an optimum spring ratio. A 50/50 spring rate ratio distribution means the spring ratio is 50% for path **B** and 50% for path **C** (Figure 7 line 2). Note that the damper characteristic would not change. If the spring ratio is changed to 30% for path **B** (with turbine mass) and 70% for path **C**, we see the results in line 3 of Figure 7. By running simulations for different spring ratios, it was found that an optimum spring ratio can be determined for each design. In the present example, the ratio of 24% for path **B** and 76% for path **C** has the best result as seen in Figure 7 line 4. If the spring ratio is decreased further to 20% for path **B** and 80% for path **C**, the drivetrain mode (vibration) returns at about 800 rpm (Figure 7 line 5). Therefore, by adjusting the spring rate of the two vibration paths of DPD **30**, vibrations normally present during normal driving modes can be reduced or eliminated from the vehicle drive train.

[0034] In some embodiments, the inertia of the flange **36** may be close to zero when the natural frequency of flange **38** approaches infinity. If needed, a small amount of friction **42** can be added internally between flange **36** and flange **38** in order to damp the natural frequency of the flange **38**. Since the flange has a small inertia and a high frequency, it would need a small amount of friction **42** to eliminate the resonance. In addition, since the added friction is just between internal flanges it does not diminish the quality of the flange in the other frequency.

[0035] Thus it is seen that the objects of the invention are efficiently obtained, although changes and modifications to the invention should be readily apparent to those having ordinary skill in the art, which changes would not depart from the spirit and scope of the invention as claimed.

Claims:

1. A damper for a vehicle drivetrain having a clutch comprising:

a first cover;

a second cover;

a first vibration pathway having a natural frequency, said first vibration pathway including a first inertial element disposed between a first pair of elastic elements, wherein one of said first pair of elastic elements is operatively connected to said first cover and the second of said first pair of said elastic elements is operatively connected to said second cover; and,

a second vibration pathway having a natural frequency, said second vibration pathway including a second inertial element disposed between a second pair of elastic elements, wherein one of said second pair of elastic elements is operatively connected to said first cover and the second of said second pair of said elastic elements is operatively connected to said second cover;

wherein said second vibration pathway is substantially parallel to said first vibration pathway; and,

wherein the natural frequency of said first vibration pathway is lower than said natural frequency of said second vibration pathway.

2. The damper for a vehicle drivetrain as recited in Claim 1 wherein the combined spring rate of said first pair of elastic elements is lower than the combined spring rate of said second pair of elastic elements.

3. The damper for a vehicle drivetrain as recited in Claim 1 wherein the mass of said first inertia element is greater than the mass of said second inertia element.

4. The damper for a vehicle drivetrain as recited in Claim 1 wherein the mass of said first inertia element is greater than the mass of said second inertia element and the combined spring rate of said first pair of elastic elements is lower than the combined spring rate of said second pair of elastic elements.

5. The damper for a vehicle drivetrain as recited in Claim 1 further comprising a friction connection between said first inertia element and said second inertia element.

6. The damper for a vehicle drivetrain as recited in Claim 1 wherein said first cover is operatively connected to an engine crankshaft of said vehicle drivetrain.

7. The damper for a vehicle drivetrain as recited in Claim 1 wherein said second cover is operatively connected to a transmission input shaft of said vehicle drivetrain.

8. The damper for a vehicle drivetrain as recited in Claim 3 wherein the mass of said first inertia element includes a turbine, said turbine fixedly attached to said first inertia element.

9. The damper for a vehicle drivetrain as recited in Claim 1 wherein at least one of said first and second pair of elastic elements comprises compression springs.

10. The damper for a vehicle drivetrain as recited in Claim 1 wherein at least one of said first and second pair of elastic elements comprises rubber slugs.

11. The damper for a vehicle drivetrain as recited in Claim 1 wherein at least one of said first and second pair of elastic elements comprises a ball ramp with diaphragm spring.

12. The damper for a vehicle drivetrain as recited in Claim 1 wherein at least one of the inertial elements is a floating flange.

13. The damper for a vehicle drivetrain as recited in Claim 1 wherein said first cover is an outer hub and said second cover is an inner hub.

14. A damper for a torque converter having a clutch comprising:

an outer hub;

an inner hub;

a first vibration pathway having a natural frequency, said first vibration pathway including a first inertial element disposed between a first pair of elastic elements, wherein one of said first pair of elastic elements is operatively connected to said outer hub and the second of said first pair of said elastic elements is operatively connected to said inner hub; and,

a second vibration pathway having a natural frequency, said second vibration pathway including a second inertial element disposed between a second pair of elastic elements, wherein one of said second pair of elastic elements is operatively connected to said outer hub and the second of said second pair of said elastic elements is operatively connected to said ;

wherein said second vibration pathway is substantially parallel to said first vibration pathway; and,

wherein the natural frequency of said first vibration pathway is lower than said natural frequency of said second vibration pathway.

15. The damper for a torque converter as recited in Claim 14 wherein the combined spring rate of said first pair of elastic elements is lower than the combined spring rate of said second pair of elastic elements.

16. The damper for a torque converter as recited in Claim 14 wherein the mass of said first inertia element is greater than the mass of said second inertia element.

17. The damper for a torque converter as recited in Claim 14 wherein the mass of said first inertia element is greater than the mass of said second inertia element and the combined spring rate of said first pair of elastic elements is lower than the combined spring rate of said second pair of elastic elements.

18. The damper for a torque converter as recited in Claim 14 further comprising a friction connection between said first inertia element and said second inertia element.

19. The damper for a torque converter as recited in Claim 14 wherein said outer hub is operatively connected to an engine crankshaft of said vehicle drivetrain.

20. The damper for a torque converter as recited in Claim 14 wherein said inner hub is operatively connected to a transmission input shaft of said vehicle drivetrain.

21. The damper for a torque converter as recited in Claim 14 wherein at least one of said first and second pair of elastic elements comprises compression springs.

22. The damper for a torque converter as recited in Claim 14 wherein at least one of said first and second pair of elastic elements comprises rubber slugs.

23. The damper for a torque converter as recited in Claim 14 wherein at least one of said first and second pair of elastic elements comprises a ball ramp with diaphragm spring.

24. The damper for a torque converter as recited in Claim 14 wherein a turbine is fixedly attached to said first inertia element.

25. The damper for a torque converter as recited in Claim 16 wherein a turbine is fixedly attached to said first inertia element.

26. The damper for a torque converter as recited in Claim 17 wherein a turbine is fixedly attached to said first inertia element.

27. The damper for a torque converter as recited in Claim 14 wherein at least one inertial element is a flange.

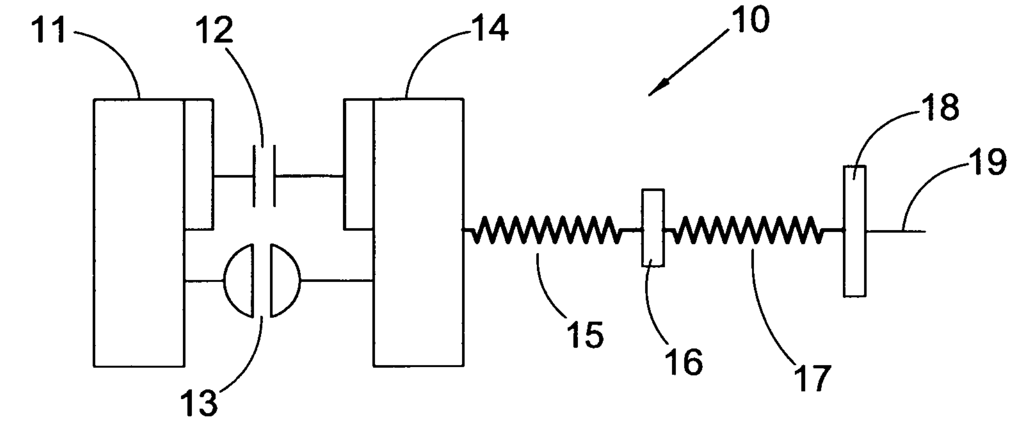


Fig. 1

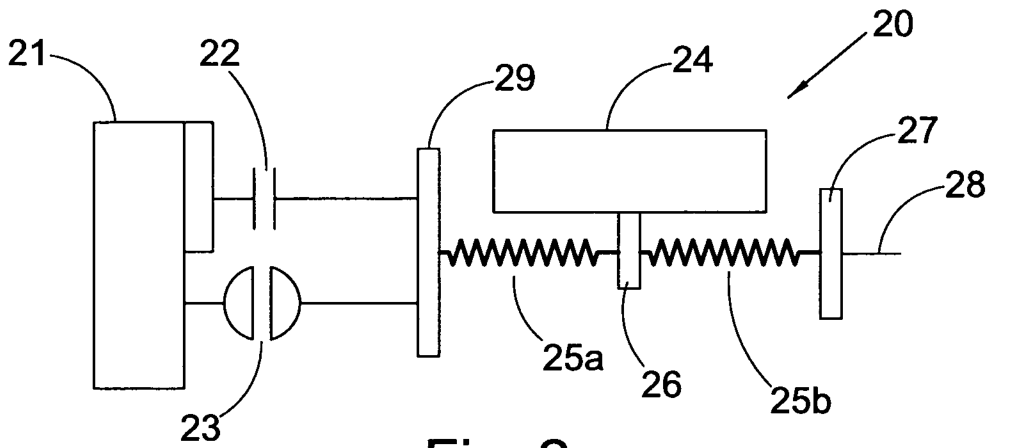


Fig. 2

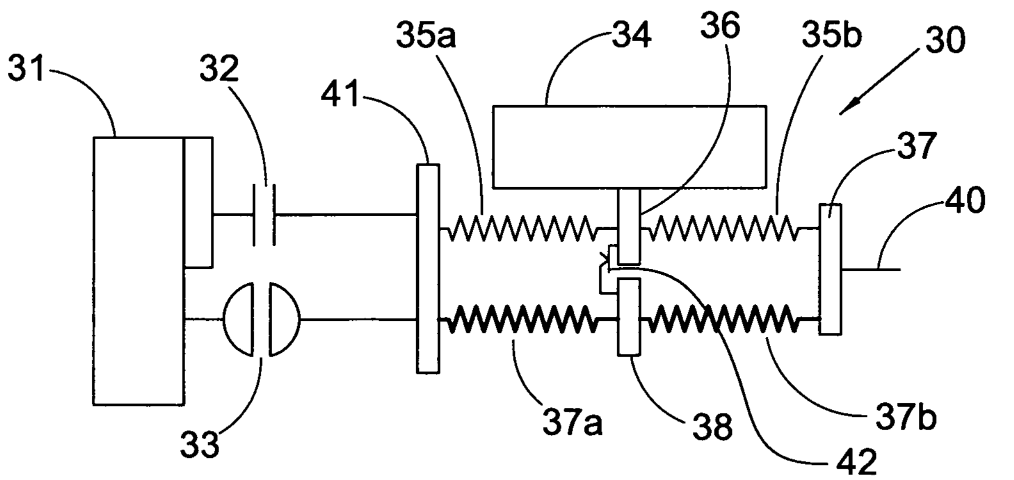


Fig. 3

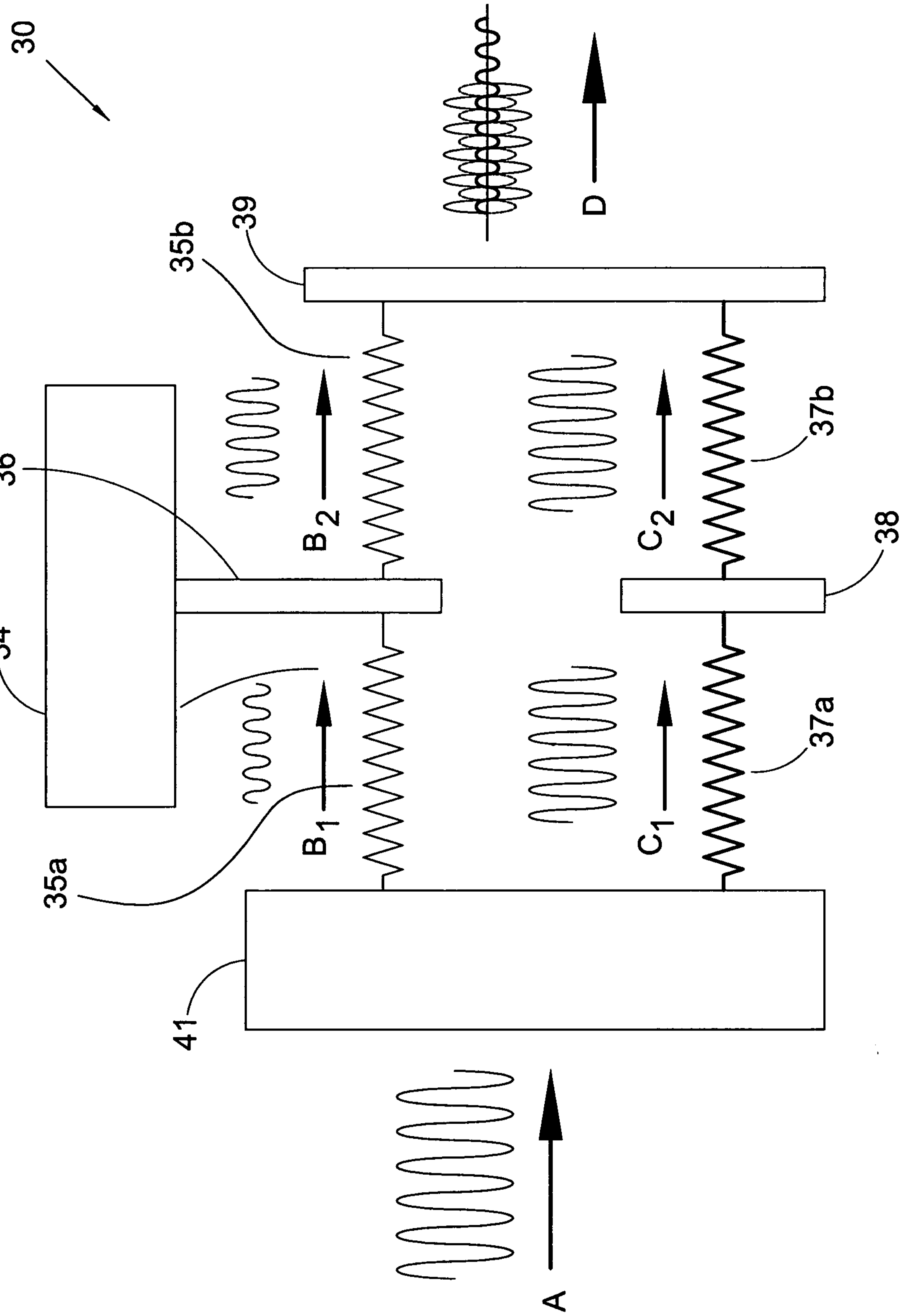


Fig. 4

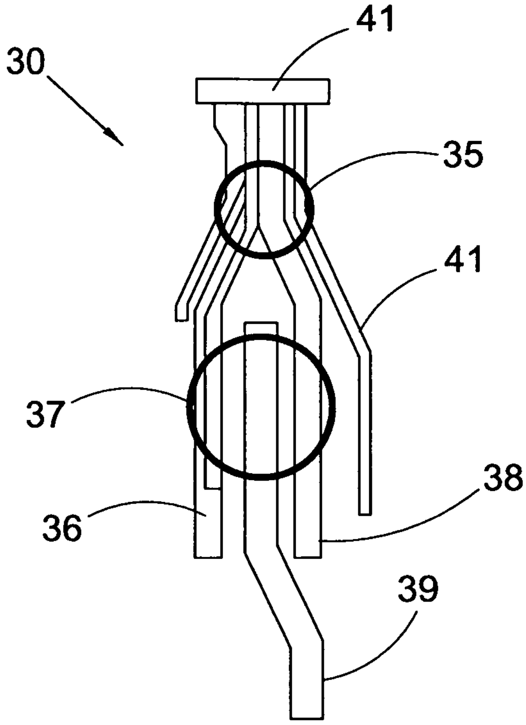


Fig. 5A

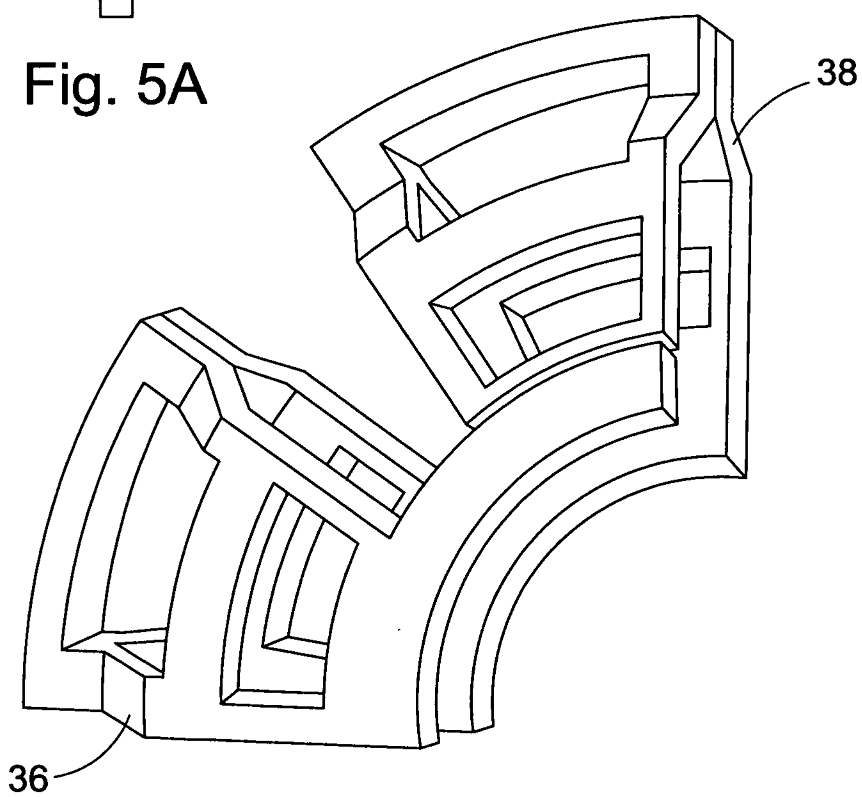
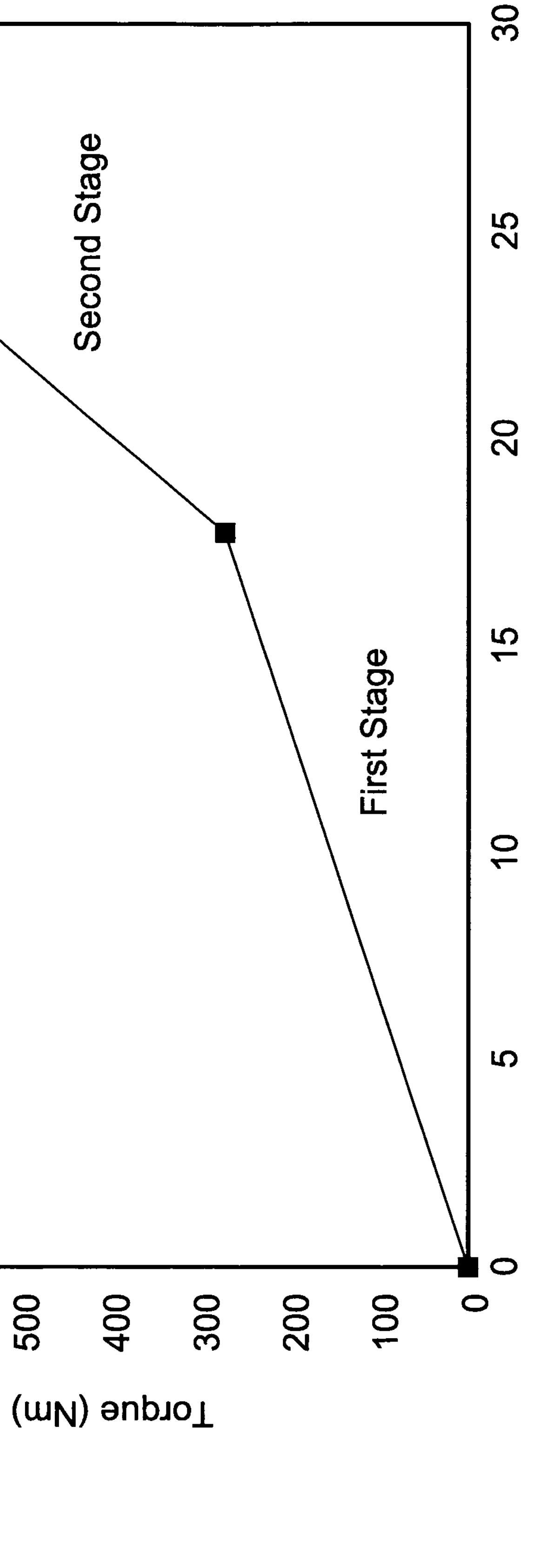


Fig. 5B



A DAMPER CHARACTERISTIC FOR WINDUP ANGLE (IN DEGREES) V/4 - V/8 EXAMPLE

Fig. 6

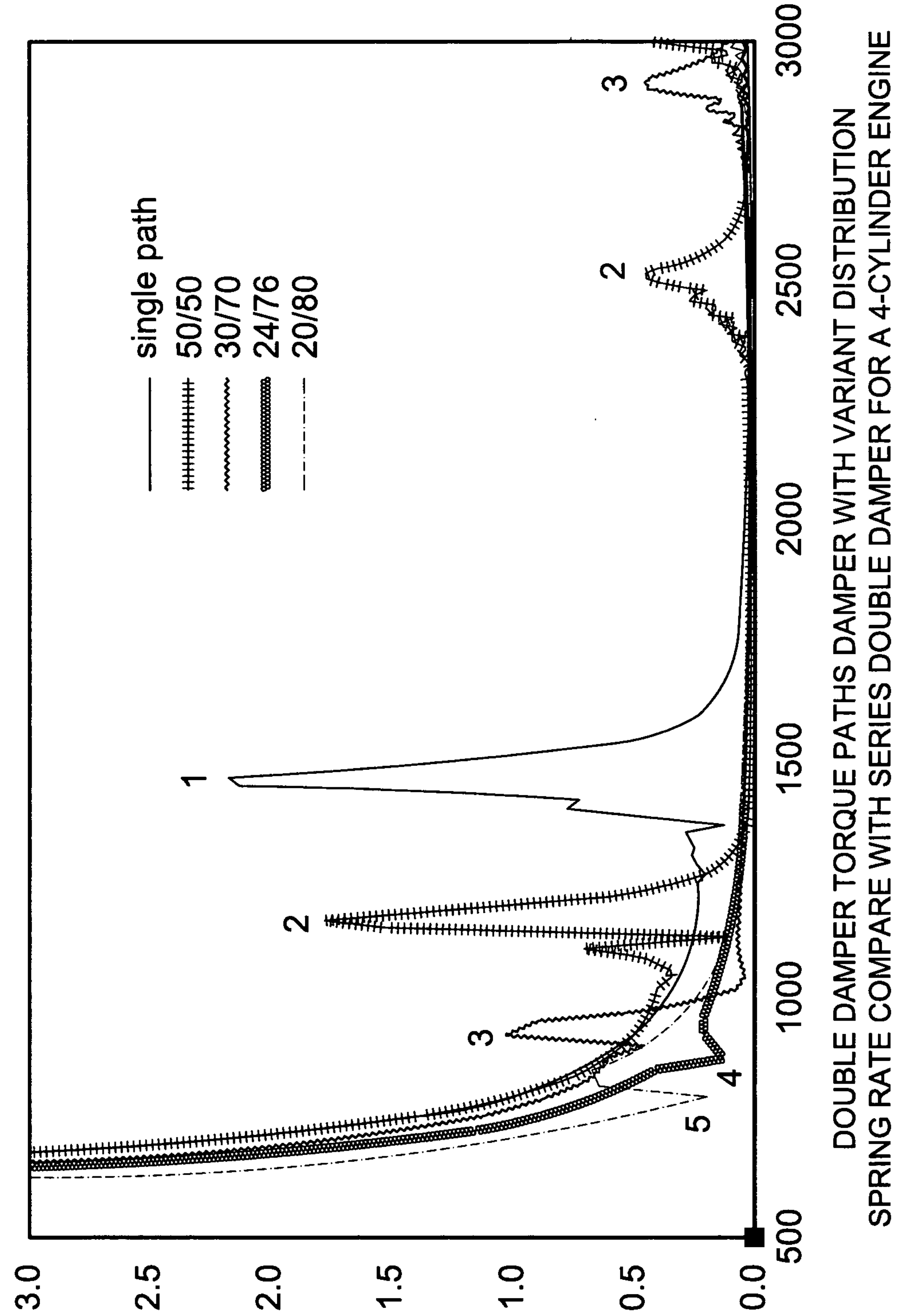
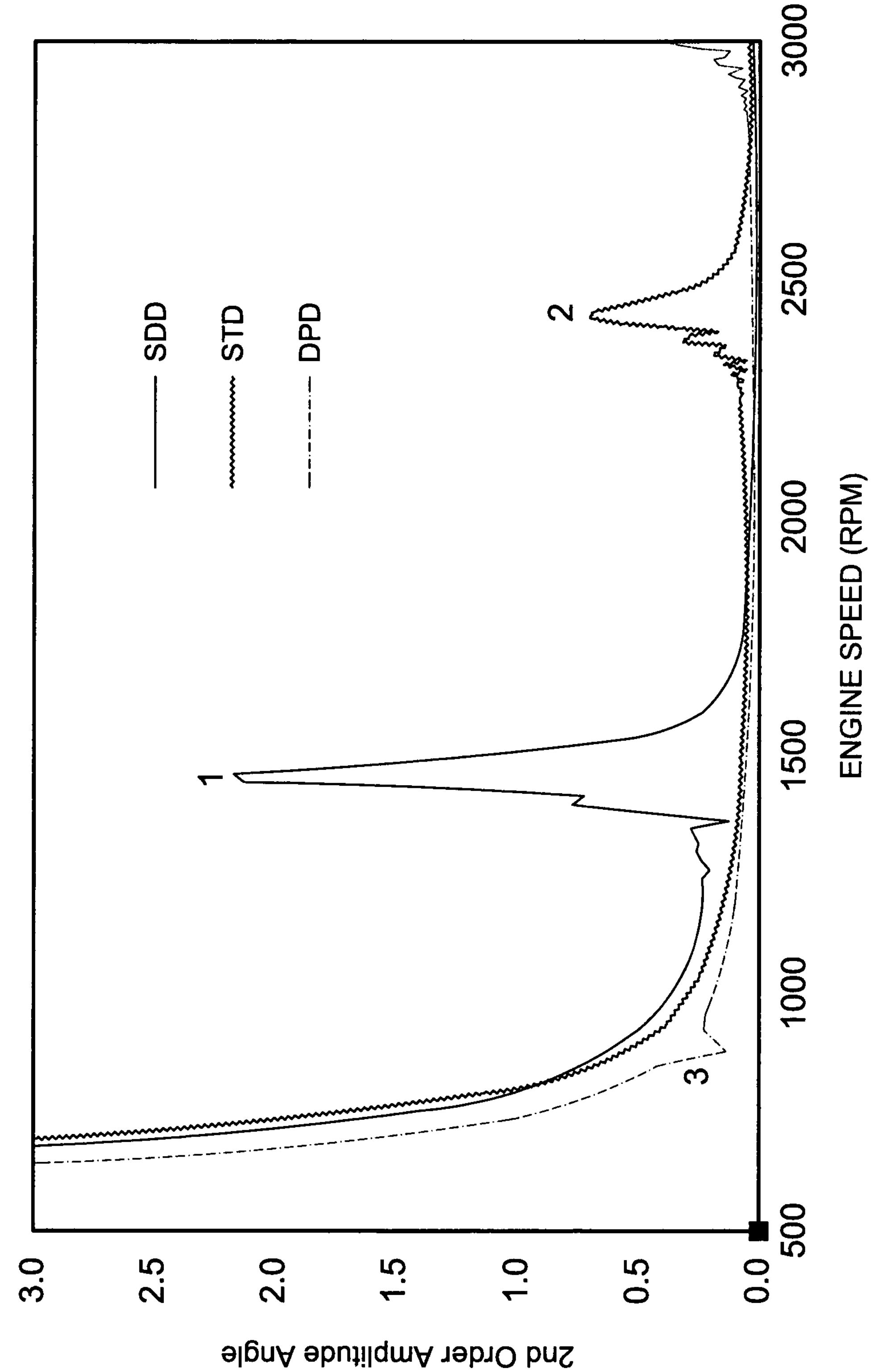

single path
50/50
30/70
24/76
20/80
1
2
3
4
5
3.0
2.5
2.0
1.5
1.0
0.5
0.0
500
1000
1500
2000
2500
3000
DOUBLE DAMPER TORQUE PATHS DAMPER WITH VARIANT DISTRIBUTION
SPRING RATE COMPARE WITH SERIES DOUBLE DAMPER FOR A 4-CYLINDER ENGINE

Fig. 7



DOUBLE TORQUE PATHS DAMPER COMPARE WITH SERIES DOUBLE DAMPER AND SERIES TURBINE DAMPER FOR A 4-CYLINDER ENGINE

Fig. 8

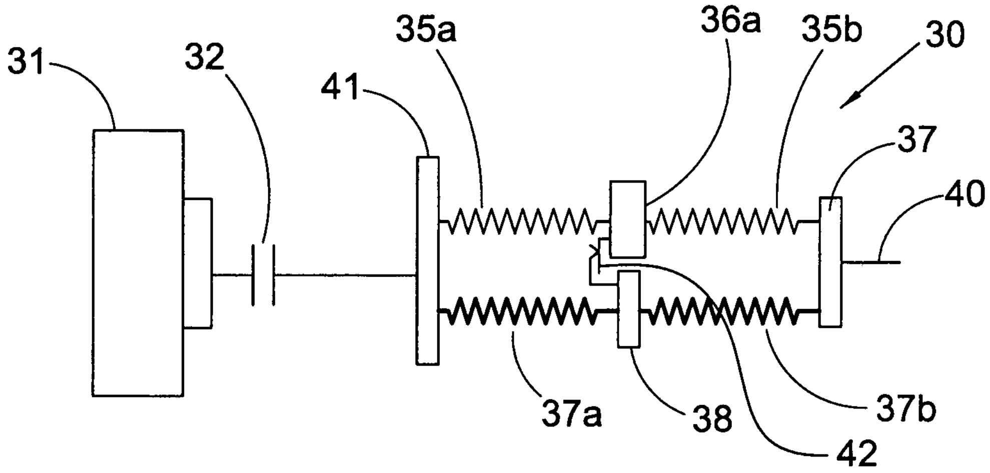


Fig. 9

INTERNATIONAL SEARCH REPORT

International application No
PCT/EP2009/006898

A. CLASSIFICATION OF SUBJECT MATTER
INV. F16F15/134

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)
F16F

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)

EPO-Internal

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
A	DE 38 23 384 A1 (FICHTEL & SACHS AG [DE]) 11 January 1990 (1990-01-11) the whole document -----	1-27
A	US 5 980 387 A (FRIEDMANN OSWALD [DE] ET AL) 9 November 1999 (1999-11-09) abstract; figures -----	1-27
A	EP 0 308 178 A2 (TOYOTA MOTOR CO LTD [JP]) 22 March 1989 (1989-03-22) abstract; figures -----	1-27
A	DE 198 39 528 A1 (DAIMLER CHRYSLER AG [DE]) 2 March 2000 (2000-03-02) abstract; figures -----	1-27
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[X] Further documents are listed in the continuation of Box C.

[X] See patent family annex.

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Date of the actual completion of the international search	Date of mailing of the international search report
25 February 2010	04/03/2010

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, Fax: (+31-70) 340-3016	Kramer, Pieter Jan

INTERNATIONAL SEARCH REPORT

International application No
PCT/EP2009/006898

C(Continuation). DOCUMENTS CONSIDERED TO BE RELEVANT

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INTERNATIONAL SEARCH REPORT

Information on patent family members

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