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(71) Applicant: **MASSACHUSETTS INSTITUTE OF TECHNOLOGY** [US/US]; 77 Massachusetts Avenue, Cambridge, MA 02139 (US).

(72) Inventors; and
(71) Applicants *(for US only)*: **WINTER, Amos, G.** [US/US]; 280 Broadway, Unit 7, Somerville, MA 2145 (US). **DORSCH, Daniel, Scott** [US/US]; 23 Speridakis Ter., Cambridge, MA 02139 (US).

(74) Agent: **TONG, Jonathan** et al.; Smith Baluch LLP, 376 Boylston St., Suite 401, Boston, MA 02116 (US).

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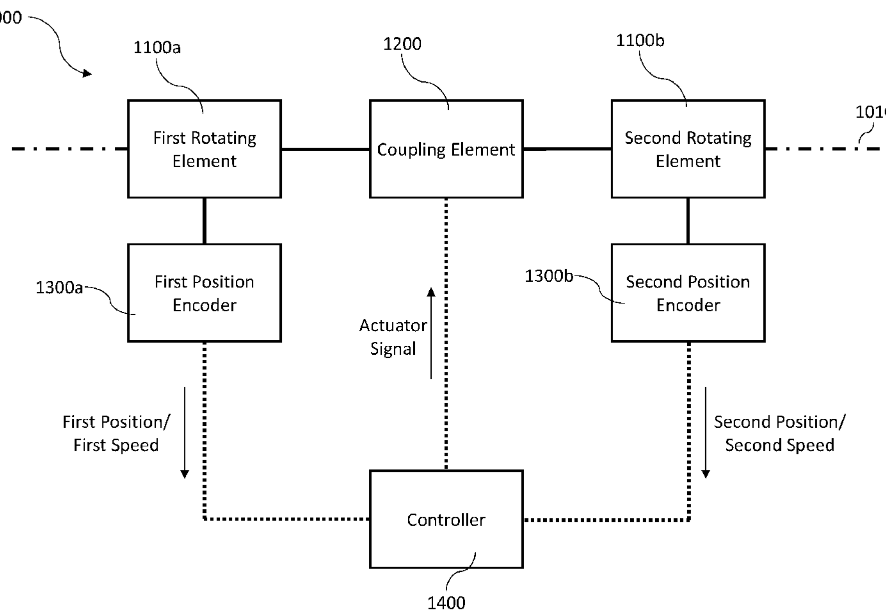

FIG. 2A

(57) Abstract: A transmission may include an input shaft, a gear that rotates relative to the input shaft, a first position encoder to measure a position and speed of the input shaft, and a second position encoder to measure a position and speed of the gear. The gear and the input shaft may be coupled using a combination of a coupling mechanism (e.g., a synchronizer) and an actuator. The transmission may further include a controller to (1) receive the position and speed from the first and second position encoders, (2) determine a relative position and a relative speed between the input shaft and the gear, and (3) output an actuator signal to activate the actuator, thus initiating a gear shift. The controller may use the position and speed to avoid undesirable collisions between the respective dog teeth of the synchronizer and input gear.

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POSITION-BASED ACTUATED COUPLING IN A MECHANICAL POWER TRANSMISSION SYSTEM

CROSS-REFERENCE TO RELATED PATENT APPLICATION(S)

[0001] This application claims priority to U.S. Provisional Application No. 62/723,303, filed on August 27, 2018, entitled "EXPERIMENTAL SETUP TO CHARACTERIZE SHIFT TIME FOR HIGH PERFORMANCE HYBRID TRANSMISSIONS," which is incorporated herein by reference in its entirety.

BACKGROUND

[0002] A vehicle's transmission connects a power source (e.g., an internal combustion engine (ICE), an electric motor (EM)) to one or more wheels on the vehicle, thus enabling mechanical energy produced by the power source to rotate the wheel(s) and, hence, move the vehicle. The transmission typically includes multiple pairs of meshed gears where one gear is mounted to an input shaft driven by the power source and the other gear is mounted to an output shaft coupled to the wheel(s). The multiple pairs of meshed gears are typically dimensioned to have different gear ratios (defined as the ratio of the diameter of the gear on the output shaft divided by the diameter of the gear on the input shaft) in order to adjust the amount of torque applied to the wheel(s) and/or the top speed of the vehicle. For example, a low gear in the transmission typically has a larger gear ratio, which imparts a higher torque to the wheels thus accelerating the vehicle faster, but limits the top speed of the vehicle. A high gear in a transmission typically has a smaller gear ratio, which produces less torque, but allows the vehicle to reach higher speeds.

[0003] Various types of transmissions have been developed and used in vehicles including, but not limited to a fully manual transmission (e.g., with a clutch and stick shift), an automatic transmission (e.g., with a torque converter), an automated manual transmission (e.g., with hydraulic actuation for the clutch and gear selection), and various hybrid configurations. The type of transmission deployed in a vehicle depends, in part, on the desired characteristics of the vehicle. For example, a high performance vehicle may utilize a lighter, smaller transmission configured to provide faster gear shifts during operation. In another example, a high efficiency vehicle may utilize a transmission that exhibits a longer lifetime, greater reliability, and a higher efficiency in transferring mechanical energy from the power source to the wheel(s).

[0004] The transmission architecture may also vary based on the type and number of power sources used in the vehicle. For example, a vehicle may be powered by only an ICE (e.g., a traditional vehicle), only an EM (e.g., an electric vehicle), or a combination of an ICE and an EM (e.g., a hybrid electric vehicle). Among these various vehicle types, hybrid electric vehicles (also referred to herein as "HEVs" or "hybrids") have become more common over the past several decades due, in part, to the higher fuel economy afforded to users, stricter emission requirements and fuel standards adopted by various countries to curb carbon dioxide emissions, and the longer range when compared to electric vehicles.

[0005] In a hybrid vehicle, the energy produced by the ICE and the EM is transferred directly or indirectly to the wheel(s). However, the manner in which this energy is delivered to the wheel(s) and the resulting impact on the design of the transmission has varied due to differing approaches in integrating the EM and the ICE into the vehicle drivetrain. Various drivetrain architectures have been deployed in hybrid vehicles including a parallel hybrid, a series hybrid, a through-the-road hybrid, and a combination of the foregoing.

[0006] In a parallel hybrid, the ICE and EM are coupled together, and may work to simultaneously power the wheel(s) of the vehicle. An engine control unit (ECU) may be used to continuously monitor and adjust the amount of power delivered by the ICE and the EM to increase the efficiency and, hence, the range of the vehicle. A transmission in a parallel hybrid, however, is typically large and heavy due, in part, to the inclusion of various gears, shafts, and clutches that allow propulsion from either or both the ICE and the EM. A mild hybrid architecture, which is similar to the parallel hybrid, has also been demonstrated where the ICE is the primary power source and the EM is only activated to provide additional torque to increase the acceleration of the vehicle when the ICE is unable to output sufficient torque.

[0007] In a series hybrid, the ICE is used solely to drive a generator (also referred to as a "range extender"), which in turn powers the EM and/or charges batteries disposed in the vehicle to later drive the EM when the ICE is not in use. Thus, the EM is the sole power source of the vehicle. Unlike conventional vehicles where the ICE is used as a power source and, thus, operates with a variable efficiency based on variations to the rotations per minute (RPM) of the ICE, the ICE in a series hybrid may primarily operate at a higher efficiency or, in some instances, at the peak efficiency corresponding to a particular RPM when in operation. A transmission in a series hybrid is typically simpler mechanically and smaller in size. This architecture also provides greater flexibility in terms of how the various components of the drivetrain (e.g., the generator) is packaged.

[0008] In a through-the-road hybrid, each axle of the vehicle has a dedicated power source. For a vehicle having two axles and, hence, two power sources, the vehicle may be powered by one or both power sources simultaneously. This architecture provides greater flexibility and ease of integration since a previously developed vehicle may be hybridized by replacing a conventional axle with an e-axle (i.e., an axle with an integrated EM and transmission). The remaining portions of the powertrain may remain unchanged. This architecture is becoming more common as e-axles are becoming more widely available.

[0009] Amongst these various types of architectures, the transmission typically includes a synchronizer to engage or disengage a gear to a shaft. FIG. 1A shows an exemplary synchronizer 100. The synchronizer 100 comprises a hub 110 that rotates with the shaft 102 and a sleeve 120 that (1) rotates with the hub 110 and (2) is slidably adjustable along a longitudinal axis of the shaft 102 relative to the hub 110. The transmission may include an actuator (not shown) to move the sleeve 120 during a gear shift. The sleeve 120 and the gear 130 may each include complementary sets of dog teeth 122 and 132 that mesh when engaged.

[0010] During a gear shift, the synchronizer 100 and the gear 130 may be initially rotating at sufficiently different speeds to prevent the engagement of the respective sets of dog teeth 122 and 132. In order to synchronize the speeds of the gear 130 and the synchronizer 100, the synchronizer 100 (or the gear 130) typically includes a conical friction element 140 (also referred to herein as "friction element") comprising a conical protrusion 142 on the gear 130 that contacts a corresponding conical recess 144 on the sleeve 120. When contact occurs, a frictional torque is imparted on the gear 130 and/or the sleeve 120, thus changing the relative speeds of the gear 130 and the synchronizer 100. Once the gear 130 and the synchronizer 100 are at a sufficiently similar speed, the sleeve 120 may then be slid along the hub 110 by the actuator until the respective sets of dog teeth of the sleeve 120 and the gear 130 are engaged. In this manner, the synchronizer 100 is used to rigidly couple a gear 130 to the shaft 102 during a gear shift.

[0011] FIGS. 1B-1E show an exemplary gear shift using the synchronizer 100. FIG. 1B shows the sleeve 120 being initially coupled to the gear 130a. FIG. 1C shows the sleeve 120 is initially disengaged from the gear 130a and moved to a neutral position between the gears 130a and 130b. FIG. 1D shows that prior to engaging the gear 130b, the sleeve 120 is first moved into a position where the friction element is engaged to reduce the speed difference between the synchronizer 100 and the gear 130b. Once the speeds of the synchronizer 100 and the gear 130b sufficiently match, the sleeve 120 is then moved into engagement with the gear 130b as shown in FIG. 1E.

SUMMARY

[0012] The Inventors have recognized and appreciated the importance of matching the speeds of the shaft and the gear in achieving a smoother gear shift. However, the Inventors have also recognized the inclusion of friction elements increases the size and weight of the transmission, especially when considering a transmission often includes multiple gears and synchronizers. A heavier transmission directly affects the performance of the vehicle by reducing the range and acceleration rate. A larger transmission may also impose constraints on the design of the vehicle design by increasing the wheelbase, which reduces vehicle maneuverability, particularly in confined spaces such as an urban environment. Furthermore, the actuator used to move the sleeve of the synchronizer during a shift event may also be correspondingly larger and/or heavier to ensure a sufficient force is provided to move the heavier elements of the transmission (e.g., the sleeve of the synchronizer), which may also increase the amount of time to shift between gears. The Inventors also recognize conventional friction elements do not ensure the dog teeth of the shaft and/or gear are aligned. When a gear shift is initiated, the dog teeth may collide or clash, producing an undesirable shock and/or acoustic noise in the vehicle.

[0013] The present disclosure is thus directed to various inventive implementations of a method and apparatus for engaging and disengaging rotating elements in a mechanical power transmission using, in part, a power source for speed matching and a position encoder to measure respective positions of the rotating elements for engagement. In some implementations, the rotating elements being engaged may be a gear and/or a shaft. The transmission may couple a gear to a shaft, a gear to another gear, or a shaft to another shaft via a coupling mechanism. The coupling mechanism may be various types of mechanisms including, but not limited to a synchronizer, a friction clutch (e.g., a single plate, multiplate, or cone clutch), a centrifugal clutch, a semi-centrifugal clutch, a conical spring clutch (e.g., a tapered finger or a crown spring clutch), a diaphragm clutch, a positive clutch (e.g., a dog or spline clutch), a hydraulic clutch, an electromagnetic clutch, a vacuum clutch, and an overrunning clutch. As an exemplary demonstration, a transmission may use a combination of speed matching and position matching to engage a gear and a shaft.

[0014] In one aspect, the power source (e.g., an EM, an ICE) may be used to adjust the relative speed of the shaft and the gear during a gear shift. The power source may output more torque compared to a conventional friction element, thus the speed difference between the shaft and the gear may be reduced more quickly. The direction and magnitude of the torque produced by the power source may also be adjustable, allowing greater control over the alignment of the shaft and the gear. In some implementations, the power source may also be used to propel the vehicle or may be an auxiliary power source (e.g., a second EM) used solely to synchronize the speeds of the various synchronizers and gears in the transmission.

[0015] In another aspect, the transmission may include a position encoder for each synchronizer and/or gear to monitor the relative position and alignment between said synchronizer and/or gear as a function of time. In some implementations, the position encoder may track the position of each dog tooth amongst a plurality of dog teeth coupled to the gear or the shaft (e.g., a sleeve of a synchronizer).

[0016] In some implementations, the position encoder may comprise an index ring and a sensor. The index ring may be coupled to rotate with a shaft and/or gear. In implementations where the shaft and the gear are engaged via dog teeth, the index ring may include a discrete number of spatial features to index the dog teeth such that the position of each individual dog tooth may be monitored. For example, the position encoder may be a magnetic encoder where the index ring comprises a plurality of magnetic pole pairs with each pole pair corresponding to a dog tooth. The sensor may be a magnetic sensor configured to measure a waveform based on the magnetic pole pair.

[0017] A controller, coupled to the position encoder, may receive the position data recorded by each position encoder to determine a relative position between the shaft and the gear. In some implementations, the controller may derive a shift window based on the relative position of the shaft and the gear. The shift window may be based on the relative positions of the gear and the shaft that provide sufficient alignment for engagement when a gear shift is initiated. For example, in implementations where a gear is engaged to a synchronizer on a shaft via complementary sets of dog teeth, the shift window may be based, in part, on predicted trajectories of said dog teeth when the synchronizer is moved into engagement with the gear. Said in another way, when a gear shift is initiated within the shift window, the actuator should move the synchronizer such that the respective dog teeth of the synchronizer and the gear successfully engage. Successful engagement may be defined based on several criteria including, but not limited to the degree of engagement between the respective dog teeth (e.g., the dog teeth may be preferably fully engaged), avoiding any undesirable collisions of the dog teeth, and ensuring any collisions (e.g., axial collisions of dog teeth when first making contact) do not generate acoustic noise that exceeds a desired threshold.

[0018] Unlike previous gear shifting methods that relied solely upon speed matching (e.g., using only a friction element), measuring the relative position of the shaft and the gear provides greater control over alignment and degree of engagement of the shaft and the gear. For instance, in past approaches where only the speeds of a synchronizer and a gear are matched, the respective dog teeth of the synchronizer and the gear may be misaligned. When attempting to engage the dog teeth, respective tip sections of each tooth may collide. Although the dog teeth may still engage after the collision, the collision may produce undesirable noise and vibration (e.g., exceeding the noise, vibration, and harshness (NVH) limits of the vehicle) especially if the actuator moves the sleeve of the synchronizer at a high velocity. Additionally, such collisions may wear down the dog teeth over time, thus reducing the operating lifetime of the transmission.

[0019] In contrast, measurements of the relative position of the dog teeth on the synchronizer and the gear may be used to align the respective dog teeth such that no undesirable collision occurs during engagement. By avoiding unwanted collisions between the dog teeth, the actuator may be allowed to move the sleeve of the synchronizer at a higher shift velocity to produce faster gear shifts. However, it should be appreciated the shift velocity may still be limited to ensure that contact between the dog teeth does not produce excessive acoustic noise.

[0020] In some implementations, measurements of the relative position between the shaft and the gear as a function of time may enable gear shifts to occur even when the shaft and the gear are not rotating at the same speed. For example, the duration of the shift window may vary based on the relative speed difference between the synchronizer and the gear. Generally, as the speed difference between the synchronizer and the gear decreases, the shift window increases in duration, but occurs at a lower frequency since the period of time where the synchronizer and the gear are unable to be engaged also increases. In some implementations, a gear shift may thus preferably occur when the duration of the shift window is such that the actuator can move the sleeve of the synchronizer at a sufficiently fast velocity for the synchronizer and the gear to be engaged.

[0021] In some implementations, the position encoder may be used in a closed feedback loop to adjust the torque applied to the shaft by the power source and/or the actuator velocity in order to achieve a desired alignment of the shaft and the gear during a gear shift. A closed feedback loop may be used, in part, to reduce the effects of systematic noise and alterations to vehicle operating conditions. For example, the ICE may produce speed irregularities in the input shaft that varies in amplitude and frequency as a function of the input shaft speed. The position encoders may monitor the effects of the speed irregularities on the relative position and motion of the respective dog teeth and the controller may modify the shift window accordingly. In some implementations, the torque produced by the power source (e.g., an EM) may also be dynamically adjusted to reduce the effects of the speed irregularities during a gear shift.

[0022] In another example, the traction of a vehicle may vary due to various effects including, but not limited to inclement weather, the type of terrain (e.g., a paved road, a dirt path, a muddy path), and the type and condition of the wheels (e.g., a racing slick, a snow tire, a tire with worn treads). Changes to vehicle traction may affect the load on the output shaft of the transmission, which in turn may cause fluctuations in the speed of the gears. These fluctuations may again be monitored by the position encoders and the controller may modify the shift window and/or to adjust the output torque produced by the power source to reduce the effect of these fluctuations when aligning and engaging the gear to the shaft.

[0023] In some implementations, the combination of the power source (e.g., an EM) and the position encoder for speed and position matching of the gear and the shaft may allow for transmissions that have no friction elements. The combination of removing friction elements (including the blocker rings) from the transmission and potential reductions to the size of the synchronizer (e.g., a thinner hub and sleeve may be used) may reduce the overall size and weight of the transmission. A lighter and smaller transmission may (1) improve vehicle performance by increasing the range and acceleration rate of the vehicle, (2) shorten the wheelbase of the vehicle to increase maneuverability, and/or (3) reduce the size and weight of the actuator allowing for faster gear shifts.

[0024] A lighter and smaller transmission may be especially beneficial for vehicles where an EM is used for propulsion. Typically, the synchronizer is subjected to large inertias on both sides. On one side of the synchronizer is the inertia of the transmission. On the other side is the inertia of the electric motor, which is typically several times larger than the inertia of a typical layshaft in an ICE vehicle. In order to complete a gear shift quickly, the EM should be used to match speeds with the new gears (and gear ratios) since conventional friction synchronizers alone would result in a longer shift time. Faster shifts may be accomplished with a friction synchronizer by increasing the size of the synchronizer such that the torque and energy dissipation limits of the synchronizer are not exceeded. However, a larger synchronizer results in a larger and heavier transmission.

[0025] Although the removal of the friction elements from the transmission has several benefits to vehicle performance, in some implementations, position matching may still have benefits even for vehicle transmissions with friction elements. For instance, a position encoder may be integrated into a conventional transmission to ensure the respective dog teeth of a synchronizer and a gear are properly aligned for engagement. In this manner, acoustic noise caused by misaligned dog teeth may be reduced.

[0026] The use of the position encoder and the controller to align and engage the respective dog teeth of the synchronizer and the gear without the respective tip portions of each dog tooth colliding may allow for reductions in the size of the dog teeth. For example, each dog tooth in a conventional set of dog teeth typically includes a tip portion that is tapered to facilitate engagement of the dog teeth when said tip portions clash with respective tip portions of another set of dog teeth. By preventing the respective tip portions of each dog tooth from colliding, the tip section of each dog tooth may be reduced in size or, in some instances, eliminated entirely, thus reducing the overall height of the dog tooth. A shorter dog tooth reduces the actuation distance to engage said dog teeth, thus reducing the gear shift time as well as the length of the transmission.

[0027] The use of the power source for speed matching and the position encoder for position matching may also be readily adapted and implemented into various drivetrain architectures including, but not limited to a single shaft transmission, a dual shaft transmission, and a transmission with a multi-speed electric drive. Additionally, the inventive embodiments described herein are not limited to vehicles (e.g., an ICE only vehicle, an electric vehicle, and/or a conventional hybrid vehicle), but may be applied to any machine with a power transmission including, but not limited to industrial equipment (e.g., extruders, crushers, conveyors) and power systems (e.g., a windmill, a turbine).

[0028] In one exemplary implementation, a mechanical power transmission includes an input shaft that rotates about a first rotation axis, a rotating element that rotates about the first rotation axis relative to the input shaft, a first position encoder, and a second position encoder. The first position encoder includes a first index ring that rotates with the input shaft and a first sensor proximate to the first index ring to measure a first position of the first index ring. The second position encoder includes a second index ring that rotates with the rotating element and a second sensor disposed proximate to the second index ring to measure a second position of the second index ring.

[0029] In another exemplary implementation, a method of shifting a mechanical power transmission comprises the following steps: (1) measuring a first position of an input shaft, (2) while measuring the first position, measuring a second position of a rotating element that is rotatably coupled to the input shaft, (3) determining a relative position and a relative speed between the input shaft and the rotating element using the first and second positions, and (4) in response to the relative position and the relative speed satisfying a criterion, actuating a coupling mechanism so as to engage the input shaft to the rotating element thereby causing the rotating element to rotate with the input shaft.

[0030] In another exemplary implementation, a mechanical power transmission includes an input shaft and a gear rotatably coupled to the input shaft with a first plurality of dog teeth. The transmission further includes a synchronizer coupled to the input shaft with a sleeve that rotates with the input shaft and has a second plurality of dog teeth disposed at a first end of the sleeve. An actuator, coupled to the sleeve, is used to slide the sleeve along the input shaft to at least one of engage or disengage the second plurality of dog teeth to the first plurality of dog teeth. The sleeve causes the gear to rotate with the input shaft when the first and second pluralities of dog teeth are engaged. The transmission also includes a first position encoder with a first index ring that rotates with the input shaft and a first sensor, disposed proximate to the first index ring, to measure a first position of the first plurality of dog teeth. The transmission also includes a second position encoder with a second index ring that rotates with the sleeve and a second sensor, disposed proximate to the second index ring, to measure a second position of the second plurality of dog teeth. A controller, coupled to the first and second position encoders and the actuator, is used to (1) receive the first and second positions as input, (2) determine at least one of a relative position or a relative speed between the input shaft and the gear based on the first and second positions, and (3) output an actuator signal that activates the actuator so as to engage the gear to the input shaft when the at least one of a relative position or a relative speed satisfies a criterion.

[0031] In another exemplary implementation, a mechanical power transmission system includes a first rotating element to rotate at a first speed about a first axis passing through the first rotating element, a first position encoder coupled to the first rotating element to measure a first angular position of the first rotating element about the first axis as a function of time and the first speed of the first rotating element, a second rotating element to rotate at a second speed about the first axis, wherein the first axis passes through the second rotating element, a second position encoder coupled to the second rotating element to measure a second angular position of the second rotating element about the first axis as a function of time and the second speed of the second rotating element, a coupling element to mechanically couple the first rotating element to the second rotating element to transfer mechanical power from the first rotating element to the second rotating element, and a controller, coupled to the first position encoder, the second position encoder, and the coupling element, to actuate the coupling element to mechanically couple the first rotating element to the second rotating element. The controller actuates the coupling element based at least in part on: (1) the first angular position and the first speed of the first rotating element measured by the first position encoder and (2) the second angular position and the second speed of the second rotating element measured by the second position encoder.

[0032] It should be appreciated that all combinations of the foregoing concepts and additional concepts discussed in greater detail below (provided such concepts are not mutually inconsistent) are contemplated as being part of the inventive subject matter disclosed herein. In particular, all combinations of claimed subject matter appearing at the end of this disclosure are contemplated as being part of the inventive subject matter disclosed herein. It should also be appreciated that terminology explicitly employed herein that also may appear in any disclosure incorporated by reference should be accorded a meaning most consistent with the particular concepts disclosed herein.

BRIEF DESCRIPTION OF THE DRAWINGS

[0033] The skilled artisan will understand that the drawings primarily are for illustrative purposes and are not intended to limit the scope of the inventive subject matter described herein. The drawings are not necessarily to scale; in some instances, various aspects of the inventive subject matter disclosed herein may be shown exaggerated or enlarged in the drawings to facilitate an understanding of different features. In the drawings, like reference characters generally refer to like features (e.g., functionally similar and/or structurally similar elements).

[0034] FIG. 1A shows an exploded of a typical synchronizer.

[0035] FIG. 1B shows a perspective view of the typical synchronizer of FIG. 1A.

[0036] FIG. 1C shows a side view of the synchronizer of FIG. 1A where the sleeve is at a neutral position.

[0037] FIG. 1D shows a side view of the synchronizer of FIG. 1A where the sleeve is engaged with a blocking ring of a gear.

[0038] FIG. 1E shows a side view of the synchronizer of FIG. 1A with the sleeve is engaged with dogteeth of a gear.

[0039] FIG. 2A shows a diagram of an exemplary mechanical power transmission.

[0040] FIG. 2B shows a cross-sectional diagram of an exemplary mechanical power transmission with a shaft and a gear for transferring power.

[0041] FIG. 2C shows a diagram of a dual actuator used to shift between gears in the transmission of FIG. 2B.

[0042] FIG. 3A shows an exemplary index ring of a position encoder.

[0043] FIG. 3B shows a magnified view of the index ring of FIG. 3A.

[0044] FIG. 4A shows exemplary index rings of two position encoders used to determine a relative position and/or phase.

[0045] FIG. 4B shows a control flow diagram, a logic statement for when to perform a gear shift, and a graphical user interface (GUI) for operating a transmission with position matching feedback.

[0046] FIG. 5A shows the respective positions of dogteeth, an index ring, and a signal waveform, corresponding to measurements of the index ring, of two gears that are engaged.

[0047] FIG. 5B shows the respective positions of dogteeth, an index ring, and a signal waveform, corresponding to measurements of the index ring, of the two gears when in position to initiate a gear shift.

[0048] FIG. 5C shows the respective signal waveforms corresponding to a top set of dogteeth in an ‘initiate shift’ position and an ‘engaged’ position relative to a bottom set of dogteeth.

[0049] FIG. 6A shows an exemplary Hall switch used as a sensor in a position encoder.

[0050] FIG. 6B shows an exemplary ratiometric Hall sensor used as a sensor in a position encoder.

[0051] FIG. 6C shows an exemplary magnetic encoder used as a sensor in a position encoder.

[0052] FIG. 7A shows an illustration of the magnetic poles of an index ring as well as the count resolution and index pulse of each dogtooth when using a magnetic encoder as the sensor.

[0053] FIG. 7B shows the index ring and the sensor of the position encoder of FIG. 7A.

[0054] FIG. 8A shows a front perspective view of an exemplary apparatus to fabricate a magnetic index ring in a position encoder.

[0055] FIG. 8B shows a top perspective view of the apparatus of FIG. 8A.

[0056] FIG. 8C shows an image of an exemplary magnetic index ring fabricated via the apparatus of FIG. 8A and the magnetic poles of the magnetic index ring.

[0057] FIG. 9A shows the characterization of the magnetic poles of an exemplary index ring using an optical encoder.

[0058] FIG. 9B shows a comparison of the counts of the optical encoder of FIG. 9A and a magnetic encoder.

[0059] FIG. 9C shows the circumferential distance error between the optical and magnetic encoders of FIG. 9B across the circumference of the index ring.

[0060] FIG. 10A shows the torque provided by an electric motor during a typical gear shift.

[0061] FIG. 10B shows the rotational speed of the electric motor during the gear shift of FIG. 10A.

[0062] FIG. 11A shows an illustration of an unwrapped gear for the analysis of dog teeth engagement.

[0063] FIG. 11B shows an illustration of an exemplary dog teeth trajectory during a gear shift and the differences between a successful and unsuccessful engagement.

[0064] FIG. 11C shows an illustration of how a relative phase between two sets of dogteeth changes as a function of time as the dogteeth move relative to each other.

[0065] FIG. 11D shows an exemplary shift window based on the trajectories of vectors V_1 and V_2.

[0066] FIG. 12A shows an illustration of the relative positions of top and bottom sets of dogteeth that result in an unwanted collision when a gear shift is initiated.

[0067] FIG. 12B shows an illustration of the relative positions of top and bottom sets of dogteeth that result in full engagement of the respective sets of dogteeth when a gear shift is initiated.

[0068] FIG. 12C shows an illustration of the relative positions of top and bottom sets of dogteeth that result in partial engagement of the respective sets of dogteeth when a gear shift is initiated.

[0069] FIG. 13 shows an exemplary state-machine method for performing a gear shift.

[0070] FIG. 14A shows a chart of the respective rotations per minute (RPM) of the electric motor and the gear to be engaged as a function of time.

[0071] FIG. 14B shows a chart of the RPM difference between the electric motor and the gear to be engaged of FIG. 14A.

[0072] FIG. 14C shows the relative phase between the top and bottom sets of dogteeth and the corresponding shift windows (labeled 'A', 'B', 'C', 'D', and 'E') based on the RPM difference of FIG. 14B.

[0073] FIG. 14D shows the shift window duration based on the shift windows of FIG. 14C.

[0074] FIG. 14E shows the shift time as a function of the speed difference range assuming a 0.5 m/s actuation velocity.

[0075] FIG. 14F shows the vertical engagement of the respective sets of dogteeth for the shift windows of FIG. 14C as a function of time.

[0076] FIG. 15A shows a chart of the RPM difference as a function of time for exemplary sets of top and bottom dogteeth.

[0077] FIG. 15B shows several exemplary trajectories of the top set of dogteeth at various points along the RPM difference curve of FIG. 15A.

[0078] FIG. 16 shows a chart of the RPM difference as a function of time for exemplary sets of top and bottom dogteeth and the use of the RPM difference to predict future trajectories.

[0079] FIG. 17A shows the RPM as a function of time for an electric motor that closely follows the speed irregularity of the internal combustion engine (ICE).

[0080] FIG. 17B shows the torque outputted by the electric motor of FIG. 17A.

[0081] FIG. 18 shows the torque of an electric motor configured to follow the speed irregularity of the ICE of FIG. 17A where the rate of change of torque of the electric motor is limited.

[0082] FIG. 19A shows an image of an exemplary benchtop setup to experimentally demonstrate the use of position matching in a transmission.

[0083] FIG. 19B shows an image of a synchronizer in the benchtop setup of FIG. 19A in a neutral position.

[0084] FIG. 19C shows an image of the synchronizer of FIG. 19B where the speed and position of the synchronizer are adjusted such that the dogteeth of the synchronizer are aligned with the dogteeth of the gear.

[0085] FIG. 19D shows an image of the synchronizer of FIG. 19C engaged with the gear.

[0086] FIG. 19E shows graphical code to control a field programmable gate array (FPGA) in the benchtop setup of FIG. 19A.

[0087] FIG. 19F shows graphical code to central processing unit (CPU) in the benchtop setup of FIG. 19A.

[0088] FIG. 19G shows a portion of a graphical user interface (GUI) to control the benchtop setup of FIG. 19A.

[0089] FIG. 19H shows another portion of the GUI of FIG. 19G.

[0090] FIG. 20A shows representative charts of the torque, speed, and piston motion of an actuator during an exemplary gear shift using the benchtop setup of FIG. 19A.

[0091] FIG. 20B shows the wheel speed and motor speed during a gear shift.

[0092] FIG. 21A shows the speed difference as a function of time between the dogteeth of a gear and a synchronizer during a shift event.

[0093] FIG. 21B shows the position difference as a function of time between the dogteeth of a gear and a synchronizer during a shift event.

[0094] FIG. 22A shows the shift position of the synchronizer as a function of time during a disengagement shift event and an engagement shift event.

[0095] FIG. 22B shows a magnified view of the engagement shift event of FIG. 22A.

[0096] FIG. 22C shows the shift force during the shift event of FIG. 22A.

[0097] FIG. 23A shows frequency spectra of the RPM magnitude measured by the first position encoder.

[0098] FIG. 23B shows an illustration of the adaptive notch filtering approach.

[0099] FIG. 23C shows various notch filters being applied to the frequency spectra of FIG. 23A.

[0100] FIG. 23D shows raw and filtered time data recorded by the first and second position encoders.

[0101] FIG. 23E shows raw and filtered frequency spectra corresponding to the first and second position encoders of FIG. 23D.

[0102] FIG. 24A shows another exemplary benchtop setup to experimentally demonstrate the use of position matching in a transmission.

[0103] FIG. 24B shows an image of the benchtop setup of FIG. 24A.

[0104] FIG. 24C shows an image of an exemplary gearset in the benchtop setup of FIG. 24A.

[0105] FIG. 24D shows an image of an exemplary electric motor in the benchtop setup of FIG. 24A.

[0106] FIG. 24E shows a diagram of a pneumatic actuator.

[0107] FIG. 24F shows an image of an exemplary actuator in the benchtop setup of FIG. 24A.

[0108] FIG. 25A shows the position of the actuator as a function of time at various pressures ranging between 20 psi and 60 psi.

[0109] FIG. 25B shows the position of the actuator as a function of time at 60 psi.

[0110] FIG. 26 shows a chart of reluctance sensor data compared to the positions of dog teeth.

[0111] FIG. 27A shows an illustration of a conventional transmission in a vehicle with an ICE.

[0112] FIG. 27B shows an illustration of a single shaft transmission.

[0113] FIG. 27C shows an illustration of a dual shaft transmission.

[0114] FIG. 27D shows an illustration of a two-speed transmission.

[0115] FIG. 27E shows an illustration of another single shaft transmission.

DETAILED DESCRIPTION

[0116] Following below are more detailed descriptions of various concepts related to, and implementations of, a method and apparatus to transfer power between rotating elements in a mechanical power transmission based, in part, on using a power source (e.g., an EM) to reduce the speed difference between the rotating elements and a position encoder to monitor the relative position of the rotating elements. It should be appreciated that various concepts introduced above and discussed in greater detail below may be implemented in multiple ways. Examples of specific implementations and applications are provided primarily for illustrative purposes so as to enable those skilled in the art to practice the implementations and alternatives apparent to those skilled in the art.

[0117] The figures and example implementations described below are not meant to limit the scope of the present implementations to a single embodiment. Other implementations are possible by way of interchange of some or all of the described or illustrated elements. Moreover, where certain elements of the disclosed example implementations may be partially or fully implemented using known components, in some instances only those portions of such known components that are necessary for an understanding of the present implementations are described, and detailed descriptions of other portions of such known components are omitted so as not to obscure the present implementations.

[0118] In the discussion below, various examples of a transmission and methods for shifting thereof are provided, wherein a given example or set of examples showcases one or more particular features of a position encoder, controller, synchronizer, and dog teeth. It should be appreciated that one or more features discussed in connection with a given example may be employed in other examples according to the present disclosure, such that the various features disclosed herein may be readily combined in a given system according to the present disclosure (provided that respective features are not mutually inconsistent).

[0119] An Exemplary Mechanical Power Transmission

[0120] FIG. 2A shows a diagram of an exemplary mechanical power transmission 1000 (also referred to herein as “transmission 1000”). As shown, the transmission 1000 may exchange mechanical power between a first rotating element 1100a and a second rotating element 1100b. A coupling element 1200 may be used to control the transfer of mechanical power by either mechanically coupling the first and second rotating elements 1100a and 1100b to transfer said mechanical power or mechanically decoupling the first and second rotating elements 1100a and 1100b to prevent transfer of said mechanical power. In some implementations, the first and second rotating elements 1100a and 1100b may rotate about a common rotation axis 1010.

[0121] The transmission 1000 may also include a first position encoder 1300a to measure the position and/or speed of the first rotating element 1100a and a second position encoder 1300b to measure the position and/or speed of the second rotating element 1100b. In some implementations, the position encoders 1300a and 1300b may measure the respective positions as a function of time. The respective speeds may then be determined on the change in the position as a function of time. The position and speed data recorded by the position encoders 1300a and 1300b may then be transmitted to a controller 1400. The controller 1400 may then transmit an actuate signal to the coupling element 1200 to either engage or disengage the first and second rotating elements 1100a and 1100b based on the position and speed data received by the controller 1400.

[0122] The first and second rotating elements 1100a and 1100b may be various types of elements including, but not limited to a shaft and a gear. The coupling element 1200 may also be various types of mechanisms including, but not limited to a synchronizer and a clutch. The following disclosure describes several inventive implementations where mechanical power is transferred between a shaft and a gear. However, it should be appreciated the various concepts described with respect to these implementations may be readily applied and/or otherwise adapted to transmissions 1000 that transfer power using other types of rotating elements.

[0123] FIG. 2B shows an inventive implementation of an exemplary mechanical power transmission 2000 where power is transferred using shaft(s) and gear(s). As shown, the transmission 2000 may include an input shaft 2100 that receives a torque input from a power source 2110 (e.g., an ICE, an EM) via a coupling 2010a. The input shaft 2100 may be supported by a support structure (not shown), such as a chassis, a frame, or a housing, via bearings 2020a and 2020b, which allow the input shaft 2100 to rotate relative to the support structure. The transmission 2000 may also include an output shaft 2200 that may receive at least a portion of the torque produced by the power source 2110 from the input shaft 2100. The output shaft 2200 may, in turn, deliver said portion of torque to the wheels of a vehicle (e.g., load 2210) for propulsion via a coupling 2010b. The output shaft 2200 may also be coupled to the support structure via bearings

2020c and 2020d, allowing the output shaft 2200 to rotate relative to the support structure.

[0124] The input shaft 2100 and the output shaft 2200 may be coupled via one or more sets of meshed gears. For instance, the input shaft 2100 may include a first input gear 2300a meshed to a first output gear 2400b and a second input gear 2300b meshed to a second output gear 2400b. In some implementations, the first and second input gears 2300a and 2300b may be rotatable with respect to the input shaft 2100 via a set of corresponding bearings while the first and second output gears 2400a and 2400b may be rigidly coupled to the output shaft 2200. The ratios of the first input and output gears 2300a and 2400a and the second input and output gears 2300b and 2400b dictate the amount of torque delivered to the output shaft 2200 as well as the speed of the output shaft 2200 relative to the input shaft 2100. The first and second input gears 2300a and 2300b are also referred to collectively as “the input gears 2300.” The first and second output gears 2400a and 2400b are also referred to collectively as “the output gears 2400.”

[0125] A synchronizer 2500 may be used to engage the input shaft 2100 to the first input gear 2300a or the second input gear 2300b. The synchronizer 2500 may include a hub 2510 that is rigidly coupled to the input shaft 2100 and a sleeve 2520 that slides along an axis parallel to the length of the input shaft 2100. The sleeve 2520 may be actuated by an actuator 2700 mounted to the support structure of the vehicle. In some implementations, the sleeve 2520, the first input gear 2300, and the second input gear 2300b may each have complementary sets of dog teeth such that the sleeve 2520 may either engage the first input gear 2300a or the second input gear 2300b depending on the position of the sleeve 2520.

[0126] It should be appreciated that other coupling mechanisms may be used to couple the sleeve 2520 and/or input shaft 2100 to the first and second input gears 2300a and 2300b (or more broadly the first rotating element 1100a to the second rotating element 1100b) including, but not limited to a friction clutch (e.g., a single plate, multiplate, or cone clutch), a centrifugal clutch, a semi-centrifugal clutch, a conical spring clutch (e.g., a tapered finger or a crown spring clutch), a diaphragm clutch, a positive clutch (e.g., a dog or spline clutch), a hydraulic clutch, an electromagnetic clutch, a vacuum clutch, and an overrunning clutch. The position encoder may generally be used to measure the position and/or speed of the respective rotating elements directly or indirectly (i.e., via the coupling mechanism).

[0127] The coupling element 1200 and/or the first and second rotating elements 1100a and 1100b should preferably have features that may be monitored by the position encoder 2600 such that the respective positions of the first and second rotating elements 1100a and 1100b (e.g., the input shaft 2100 and the input gear 2300) are measured at a resolution sufficient to perform a gear shift that meets a desired criteria. The criteria may include, but is not limited to preventing undesirable collisions within the coupling mechanism and reducing acoustic shock and/or noise.

[0128] The transmission 2000 may also include a first position encoder 2600a to monitor the position of the input shaft 2100, a second position encoder 2600b to monitor the position of the first input gear 2300a, and a third position encoder 2600c to monitor the second input gear 2300b. The first, second, and third position encoders 2600a-2600c may be collectively referred to as the "position encoder 2600." Each position encoder may include an index ring 2610 that rotates with the respective shaft or gear and a sensor 2620, mounted to the support structure, to monitor the position of the respective shaft or gear based on the index ring 2610. The transmission 2000 may further include a controller 2800 that receives the position of the input shaft 2100 and/or the input gear 2300 recorded by each position encoder 2600 as a function of time. The controller 2800, in turn, may process the respective positions to derive a relative position and speed between the input shaft 2100 and/or the input gear 2300. The controller 2800 may then initiate a gear shift by actuating the actuator 2700 via an actuator signal when the desired relative position and speed of the input shaft 2100 and/or the input gear 2300 occur.

[0129] It should be appreciated the power source 2110 coupled to the transmission 2000 may be various types of power sources including, but not limited to an internal combustion engine (ICE) and an electric motor (EM). In some implementations, the power source 2110 may be used to propel the vehicle and to match the speeds of the input shaft 2100 and the input gear 2300. In some implementations, the power source 2110 may be an auxiliary power source that is used solely for speed synchronization and not for propulsion. In some implementations, the power source 2110 may output a higher torque and adjust the torque more quickly than a conventional synchronizer with a friction element. Thus, the power source 2110 may offset or, in some instances, replace the friction element when speed matching the input shaft 2100 to the input gear 2300.

[0130] It should also be appreciated that the transmission 2000 shown in FIG. 2B is one exemplary implementation. The respective components of the transmission 2000 may be arranged differently in other transmission designs. For example, the transmission 2000 may include multiple input shafts 2100 corresponding to multiple power sources 2110 (e.g., a hybrid vehicle may include two input shafts 2100 for the EM and the ICE). In another example, the transmission 2000 may include multiple output shafts 2200.

[0131] It should also be appreciated the transmission 2000 may include multiple synchronizers 2500 for additional gears in the transmission. In some implementations, each synchronizer 2500 may be actuated by a corresponding actuator 2700. In some implementations, a single actuator 2700 may be configured to actuate multiple synchronizers 2500 (e.g., using multiple shift rods and/or forks). In some implementations, the synchronizer 2500 may be disposed on the output shaft 2200 instead of the input shaft 2100. For such configurations, the input gears 2300 may be rigidly mounted to the input shaft 2100 and the output gears 2400 may be rotatable relative to the output shaft 2200.

[0132] The input shaft 2100, the output shaft 2200, the couplings 2010a and 2010b, and the bearings 2020a, 2020b, 2020c, and 2020d may be similar to components previously used in various transmissions for various types of vehicles including, but not limited to ICE vehicles, electric vehicles, and hybrid vehicles. The first input and output gears 2300a and 2400a and the second input and output gears 2300b and 2400b may also be based on meshed gears previously used in vehicle transmissions. The input gear 2300 may include a body with gear teeth, disposed along the exterior edge (e.g., the circumference), that engages the output gears 2400 of the transmission 2000. The gear teeth of the input gear 2300 is to be distinguished from the dog teeth (or another coupling mechanism) used to engage the input gear 2300 to the sleeve 2520 of the synchronizer 2500. The dog teeth may be a separate component mounted to the body or integrated and formed as part of the body of the input gear 2300.

[0133] In some implementations, the synchronizer 2500 and/or the input gear 2300 may not include a friction element. As described above, a typical friction element includes a conical protrusion, recess, and blocker ring. In conventional synchronizers, a portion of the volume of the sleeve and/or the hub may be used to support the friction element. By eliminating the friction element from the synchronizer 2500, the length of the sleeve 2520 and the hub 2510 may be reduced, thus reducing the weight and size of the transmission 2000. For example, the sleeve 2520 and the hub 2510 may have a length that is reduced by about 30% to about 50% compared to a synchronizer with a friction element. The combination of removing the friction elements and reducing the size of the synchronizer 2500 may reduce the overall weight and size of the transmission 2000.

[0134] In some implementations, the geometry of the dog teeth of the sleeve 2520, the first input gear 2300a, and the second input gear 2300b may also be based on conventional dog teeth design. For example, each dog tooth may comprise a tapered tip section and a base section. However, the inclusion of the position encoders and the resultant speed and position matching functionality of the transmission 2000 may allow for simpler, smaller dog tooth geometries. For example, the use of position matching to prevent unwanted collisions of dog teeth may allow the tip sections of each dog tooth to be substantially reduced or, in some instances, eliminated entirely. Similarly, the base section of each dog tooth may also be reduced provided the dog tooth is able to remain engaged with another corresponding dog tooth during operation.

[0135] The actuator 2700 is used to move the sleeve 2520 of the synchronizer 2500 to disengage and/or engage the first or second input gears 2300a and 2300b to the input shaft 2100. In some implementations, the synchronizer 2500 may also include a neutral position where the sleeve 2520 is substantially disengaged from both the first and second input gears 2300a and 2300b such that no torque is transferred to the output shaft 2200. Thus, during a gear shift, the actuator 2700 may be configured to move the sleeve 2520 between three indexed positions (i.e., a neutral position, a first position to engage the first input gear 2300a, a second position to engage the second input gear 2300b).

[0136] To actuate between said indexed positions, the actuator 2700 may include one or more actuators. For example, FIG. 2C shows the actuator 2700 may include two pneumatic actuators 2710a and 2710b in a stacked arrangement. As shown, the actuator 2700 may actuate between the three positions based on the whether one or both of the actuators 2710a and 2710b are retracted and/or extended. Specifically, the actuator 2700a may have a piston that shifts the sleeve 2520 from the second input gear 2300b to neutral and the actuator 2700b may have a piston that shifts from neutral to the first input gear 2300a. The actuator 2700b may be mounted onto a slider or rail moved by the actuator 2700a.

[0137] When shifting from the second input gear 2300b to the first input gear 2300a, the actuator

2700a may first be extended such that the piston of actuator 2700a contacts a hard stop. As the actuator 2700a is used, air may be supplied to the actuator 2700b to keep the piston of actuator 2700b in a retracted position. Once the sleeve 2520 is moved to the neutral position, the air used to constrain the piston of actuator 2700b may be removed and the piston of actuator 2700a may remain pressurized to maintain the desired sleeve 2520 position. Once the shift is triggered, the actuator 2700b may be activated and the piston of actuator 2700b extended to move the sleeve to engage the first input gear 2300a.

[0138] The stacked configuration of two actuators is a simple, inexpensive approach to provide actuation between three indexed positions. Other actuator 2700 configurations, however, may also be implemented in the transmission 2000. For example, actuators with three-position pistons may be used. Additionally, electronically controllable linear motors may also be used for actuation. Generally, the actuator 2700 may be various types of actuators including, but not limited to a hydraulic actuator, a pneumatic actuator, an electrical actuator, a mechanical actuator, a solenoid actuator, and any combination of the foregoing. The choice of a particular actuator type may depend, in part, on the desired force output and the speed at which such actuation may be applied.

[0139] For example, pneumatic pistons may be preferable in a transmission where the desired force for actuation of the sleeve 2520 is small and/or simpler implementation is desirable. For instance, the exemplary demonstration of the transmission 2000 described below utilizes pneumatic pistons for ease of implementation in a lab environment. In another example, hydraulic actuators may be preferable in vehicles where a high force output and/or rapid speed synchronization is desirable.

[0140] The size and dimensions of the actuator 2700 may also depend, in part, on the size and mass of the components being actuated (e.g., the sleeve 2520 of the synchronizer 2500 and/or the first and second input gears 2300a and 2300b). For example, a shift event (i.e., a shift from one gear to another gear) typically includes four phases: (1) disengagement from a gear (e.g., moving a sleeve of a synchronizer into a neutral position), (2) pre-synchronization (e.g., moving the sleeve such that a friction element is engaged), (3) synchronization (e.g., matching the speeds of the sleeve and input gear), and (4) engagement to another gear (e.g., moving the sleeve into engagement). Of these four phases, the disengagement, pre-synchronization, and engagement phases are typically achieved by the actuator 2700.

[0141] For a given hydraulic pressure (e.g., higher pressure requires larger pumps and more pumping energy), a larger diameter actuator is desirable in order to provide a higher force output, thus resulting in a lower synchronization time. However, given a limited fluid flow rate, which may be determined by the hydraulic valve, a larger piston may move more slowly resulting in slower piston motion and, hence, a longer duration for the disengagement, pre-synchronization, and engagement phases.

[0142] In a conventional transmission, the tradeoff between high force and slow speed should be balanced. However, for the transmission 2000, speed matching between the synchronizer 2500 and the input gears 2300 may be performed by another power source 2110 (e.g., an EM). Thus, the synchronization phase of the shift event may no longer be reliant on the actuator 2700. For this reason, the transmission 2000 may allow for a smaller actuator 2700. For example, the actuator 2700 may have a smaller piston, which results in faster piston motion and thus a faster gear shift. The size of the actuator 2700 may be further reduced so long as the force output is sufficient to move the various components of the transmission (e.g., the sleeve 2520, a shift rod, and/or a shift fork). For instance, the removal of friction elements from the synchronizer 2500 and the input gears may allow the actuator 2700 to be further reduced in size. In some implementations, the transmission 2000 may utilize an actuator 2700 capable of providing a shift velocity ranging between about 0.1 m/s to about 1 m/s.

[0143] The shift velocity of the actuator 2700 may also be varied during a shift event. For example, the controller 2800 may send an actuator signal that changes the velocity of the actuator 2700 based on the particular phase of the shift event. During the disengagement, pre-synchronization, and synchronization phases, the shift velocity may be increased to achieve a faster gear shift. However, during the engagement phase, the shift velocity may be reduced to prevent the synchronizer 2500 and the input gear 2300 from generating acoustic noise, vibration, or shock that exceeds a desired threshold (e.g., the NVH limits of a vehicle).

[0144] The position encoder 2600 is used to monitor the position of the input shaft 2100 and/or the input gears 2300. In implementations where the input shaft 2100 is coupled to the input gear 2300 via respective dog teeth of the sleeve 2520 and the input gear 2300, the position encoder 2600 may have a resolution sufficient to track the position of each dog tooth within the dog teeth as a function of time. Said in another way, the position encoder 2600 may have sufficient angular resolution to resolve the angular position of each dog tooth amongst a plurality of dog teeth. This is in stark contrast to conventional position encoders used in vehicle transmissions (e.g., an optical encoder with a one count per revolution resolution), which are typically only able to monitor the speed of a shaft. In some implementations, the position encoder 2600 may be configured to continuously monitor the position of the dog teeth during operation of the vehicle. In some implementations, the position encoder 2600 may be activated only during a gear shift.

[0145] In some implementations, the position encoder 2600 may include an index ring 2610 and a sensor 2620. As described above, the sensor 2620 may be mounted to a static reference (e.g., the support structure) and the index ring 2610 may be mounted to the rotating component (e.g., the input shaft 2100/sleeve 2520 of the synchronizer, the first input gear 2300a, the second input gear 2300b). The index ring 2610 may be dimensioned to be larger or smaller than the rotating component being monitored. In some implementations, the index ring 2610 may be mounted directly to the rotating component (e.g., the index ring 2610 is mounted to the input gear 2300 and thus rotates with the input gear 2300) or indirectly (e.g., the index ring 2610 is mounted to the output gear 2400 and thus rotates in along a substantially opposite direction to the input gear 2300). For example, the index ring 2610 may be mounted to the first output gear 2400b. The position of the first input gear 2300a may be derived based on the gear ratio between the first input gear 2300a and the first output gear 2400b.

[0146] The index ring 2610 may include multiple spatial features that are correlated with the position of the input shaft 2100 and/or the input gear 2300. These spatial features may be measured by the sensor 2620 and used to determine a relative position between the input shaft 2100 and the input gear 2300. For example, the position encoder 2600 may be an optical encoder. The index ring 2610 may be a rotating disk where the spatial features are radial, periodic patterned sections that transmit or block light (e.g., a chopper wheel of an optical chopper). The sensor 2620 may be a light sensor (e.g., a photodiode, a bolometer, a thermopile) to monitor the amount of light transmitted by the spatial features. In another example, the position encoder 2600 may be a magnetic encoder. In this case, the index ring 2610 may be a rotating disk where the spatial features are periodic, alternating magnetic poles. The sensor 2620 may be a magnetic sensor (e.g., a Hall sensor) configured to detect a portion of a magnetic pole and produce a produce a signal corresponding to the magnetization of said portion.

[0147] In implementations where dog teeth are used as the coupling mechanism, the spatial features of the index ring 2610 may be configured to produce an index pulse response corresponding to each dog tooth. Once an index pulse is detected by the sensor 2620, the sensor 2620 may then record a signal response from the spatial feature(s) corresponding to said dog tooth. For example, the spatial feature may produce a signal response with various waveforms including, but not limited to a square wave, a sine wave, a sawtooth wave, and any combinations of the foregoing. The signal response may be a digitized representation of the spatial feature where each point recorded by the sensor 2620 may correspond to a specific location along said spatial feature. This, in turn, may be correlated to a specific location along said dog tooth (e.g., the edge(s) and center of the dog tooth).

[0148] In some implementations, the sensor 2620 may record said points at substantially equal time intervals. Thus, the cumulative number of points recorded (also referred to as "counts") after detection of an index pulse may be used to derive a position along said spatial feature and/or dog tooth. In some implementations, the number of counts recorded for each dog tooth may instead be kept substantially constant. Thus, the time intervals for recording the points may vary for each dog tooth detected as the index ring 2610 is being accelerated or decelerated.

[0149] The spatial features may be distributed uniformly about the rotation axis of the rotating component. Referring to the case where the synchronizer 2500 and the input gear 2300 engage via respective dog teeth, the number of spatial features on the index ring 2610 may, in practice, differ from the number of dog teeth being monitored. The position of each dog tooth may thus be derived as a fractional number of the spatial features. For such cases, an indexed tooth and/or a reference spatial feature may be used to identify the position of a particular dog tooth based on the rotating component completing at least one full revolution.

[0150] In some implementations, the number of spatial features on the index ring 2610 may correspond to an integer multiple of the dog teeth being monitored. For such cases, the position of a particular dog tooth amongst the dog teeth may be unnecessary. Instead, measuring one (or an integer multiple) number of spatial features may be sufficient to determine a relative position between the input shaft 2100 and the input gear 2300. For example, a single spatial feature may correspond to a single dog tooth. In some implementations, an integer multiple number of spatial features may instead be used for each dog tooth. It should also be appreciated the multiple features of the index ring 2610 do not have to be spatially aligned to each dog tooth.

[0151] The count value recorded by the sensor 2620 may also be periodically reset. For example, the count number may be reset after the dog teeth completes one revolution (i.e., the highest count number corresponds to the product of the number of dog teeth and the total number of counts per dog tooth). In another example, the count value may be reset for each index pulse detected (i.e., the highest count number corresponds to the total number of counts recorded for each dog tooth). This approach may be preferable when comparing the relative positions of two sets of dog teeth and/or to reduce the amount of memory used to store the position of the dog teeth. For example, if the angular resolution and the number of recorded counts per dog tooth are the same for each position encoder 2600, then the relative position of the respective dog teeth may be found by subtracting the respective count values recorded by each sensor 2620. The resulting difference in count value provides the relative position of the respective dog teeth relative to an initial offset. In some implementations, the offset may be defined based on when the respective dog teeth are engaged.

[0152] In some implementations, the position encoder 2600 may be used to monitor multiple shafts and/or gears in the transmission 2000. For example, a single position encoder 2600 may be disposed on the input shaft 2100. The input shaft 2100 may further include multiple synchronizers 2500 that each have multiple sets of dog teeth (e.g., one synchronizer 2500 may have two sets of dog teeth to engage to two input gears 2300). The multiple sets of dog teeth may vary in number, but may each be constrained to rotate with the input shaft 2100. Thus, so long as the ratio of the dog teeth between the input gears 2300 are known, a single position encoder 2600 monitoring one input gear 2300 may be used to determine the position of other input gears 2300 in the transmission 2000. In another example, the multiple output gears 2400 may be rigidly mounted to the output shaft 2200. A position encoder 2600 may be used to monitor the position and/or speed of one output gear 2400 based on the gear teeth instead of the dog teeth. The number of gear teeth between the output gears 2400 may vary based on their respective gear ratios. If said gear ratios are known, it may be possible to monitor the position of each output gear 2400 using a single position encoder 2600.

[0153] The position encoder 2600 may be various types of encoders including, but not limited to a magnetic encoder and an optical encoder. The type of encoder used in the transmission 2000 may depend, in part, on the operating conditions within the transmission. For instance, the position encoder 2600 should be able to withstand the high temperatures, immersion in oil (or other fluids within the transmission 2000), and vibrations during operation.

[0154] In one exemplary implementation, the position encoder 2600 may be a magnetic encoder used to monitor the position of dog teeth on the sleeve 2520 of the synchronizer 2500 and/or the input gear 2300. A magnetic index ring 2610 may be desirable in a vehicle transmission because magnetic materials have been shown to be able to withstand a hot, oil filled, and high vibration environment. Additionally, the signal produced by a magnetic index ring 2610 typically exhibits a more stable amplitude at different rotational speeds unlike a reluctance sensor, which receives a higher voltage signal at higher shaft velocities.

[0155] FIGS. 3A and 3B show an exemplary index ring 2610 used in the position encoder 2600. As shown, the index ring 2610 may include a plurality of magnetic pole pairs 2612 (i.e., the plurality of spatial features) uniformly distributed along the circumference (or edge) of the index ring 2610. The sensor 2620 may thus be positioned proximate to the edge of the index ring 2610. In some implementations, the magnetic pole pairs 2612 may instead be distributed along the face of the index ring 2610 and the sensor 2620 oriented parallel to the rotation axis of the index ring 2610 to monitor the position of the index ring 2610.

[0156] The index ring 2610 may be formed in various ways including, but not limited to a bulk magnetic rotor and a magnetic tape overlaid along the circumference of a non-magnetic rotor. Each pole pair 2612 may include a region with a first magnetic pole 2614 oriented towards the center of the index ring 2610 and a second magnetic pole 2616, adjoining the first magnetic pole 2614, oriented away from the center of the index ring 2610. The plurality of pole pairs 2612 may be arranged such that the first and second magnetic poles 2614 and 2616 alternate along the index ring 2610. In some implementations, the index ring 2610 may be configured such that each pole pair 2612 corresponds to a single dog tooth. Alternatively, multiple pole pairs 2612 may be used to track a single dog tooth.

[0157] FIG. 3B shows the pole pair 2612 may produce a sinusoidal signal response when measured by the sensor 2620. In some implementations, the sensor 2620 used to track the pole pairs 2612 on the index ring 2610 may include two Hall sensors oriented 90 degrees out of phase with respect to one another. As shown in this example, a single pole pair 2612 may correspond to a single period of the sinusoidal signal response. In some implementations, an index pulse may be detected by the sensor 2620 when the amplitude of the sinusoidal signal exceeds a desired threshold. The threshold, for instance, may correspond to a peak of the sinusoidal signal response. In some implementations, the threshold may be recorded by integrating a portion of the sinusoidal signal response. The threshold for triggering an index pulse response may be adjusted based, in part, on the desired precision of the dog tooth position and/or the signal to noise ratio of the signal response.

[0158] The relative position of the respective dog teeth of the two components may thus be determined by comparing the two sinusoidal signals recorded by the corresponding position encoders 2600. FIG. 3B shows an overlay of the two sinusoidal signals. The phase difference between the two sinusoidal signals corresponds to the relative difference in position along respective spatial features and thus, the relative position along the respective dog teeth.

[0159] FIG. 4A shows an exemplary arrangement of two index rings 2610a and 2610b with an angular offset. As shown, each position encoder 2600 resets the count value after detecting an index pulse. The count values recorded by the two position encoders 2600 are different (e.g., count1 = 140 and count2 = 35). If each spatial feature and/or dog tooth is recorded using the same number of counts, then the subtraction of these count values (e.g., count1 – count2 = 105) by the controller 2800 provides the relative alignment and/or position offset of the respective dog teeth. By measuring the count values as a function of time, the controller 2800 may also extract the velocity of the respective dog teeth. FIG. 4B shows an exemplary flow chart and logic flow to measure the position and speed difference between the two index rings 2610a and 2610b. An exemplary GUI to control and monitor the position and speed difference is also shown.

[0160] The phase difference measured between the two position encoders 2600 may thus be used to determine when the respective dog teeth are in sufficient alignment for engagement. FIGS. 5A-5C show exemplary diagrams of the alignment between a top set of dog teeth 2310a and a bottom set of dog teeth 2310b and corresponding sinusoidal signal responses. Generally, the dog teeth 2310a and 2310b may both be rotating at varying speeds during operation. However, for the purposes of illustrating the effects of varying alignment between the dog teeth 2310a and 2310b, the dog teeth 2310b is fixed relative to the dog teeth 2310a.

[0161] As previously mentioned, the dog teeth 2310a and 2310b may not be initially aligned, thus any measurement of the phase difference may include an offset. This offset may not be fixed, but instead may vary depending on the relative speed of the dog teeth 2310a and 2310b when the sinusoidal signal responses are first recorded. In order to calibrate for this offset, the dog teeth 2310a and 2310b should be arranged in a manner that produces a known and repeatable offset.

[0162] In one example, FIG. 5A shows one reference arrangement where the dog teeth 2310a and 2310b are engaged. When engaged, the dog teeth 2310a and 2310b will rotate at the same speed, thus producing an offset that remains substantially unchanged. The offset may be measured based on the phase difference between the two recorded sinusoidal signal responses. By measuring the offset when the dog teeth 2310a and 2310b are engaged, the offset may also be used to determine the appropriate alignment to engage the dog teeth 2310a and 2310b. Said in another way, the dog teeth 2310a and 2310b may be engaged when the alignment of the dog teeth 2310a and 2310b is such that the resultant relative motion produces a final phase difference between the signal responses that matches the offset.

[0163] This calibration may be performed at the factory where the transmission 2000 is manufactured and/or assembled. Furthermore, this calibration may be periodically repeated during operation of the transmission. For instance, the offset may be calibrated each time the dog teeth 2310a and 2310b are engaged.

[0164] FIG. 5B shows that once the offset is calibrated, subsequent engagement of the dog teeth 2310a and 2310b involves determining the appropriate phase difference to initiate a shift. The phase differences to initiate a shift may depend on the relative motion of the dog teeth 2310a and 2310b during actuation (e.g., the shift velocity of the dog teeth 2310a and 2310b moving into engagement), the relative rotational velocity of the dog teeth 2310a and 2310b, and the distance separating the dog teeth 2310a and 2310b. FIG. 5B shows one possible phase difference to initiate a shift that would engage the dog teeth 2310a and 2310b without causing unwanted collisions between the dog teeth 2310a and 2310b. FIG. 5C shows the sinusoidal signal responses of the dog teeth 2310b (again fixed relative to dog teeth 2310a for purposes of illustration) and the dog teeth 2310a when initiating a shift and when fully engaged.

[0165] In this exemplary implementation, the sensor 2620 may be various types of magnetic sensors including, but not limited to a Hall switch, a ratiometric Hall sensor, and a magnetic encoder. The particular type of magnetic sensor may depend on the desired waveform to monitor the index ring 2610 and/or the sampling rate. For example, FIG. 6A shows an exemplary Hall switch that produces a square wave per dog tooth. FIG. 6B shows an exemplary ratiometric Hall sensor that produces a sine wave per dog tooth. FIG. 6C shows an exemplary magnetic encoder that produces a square wave per dog tooth, which may then be sampled at 160 counts per tooth.

[0166] In one exemplary implementation, the magnetic encoder of FIG. 6C may be an AS5304 magnetic Hall effect sensor. The sensor may include two internal Hall effect sensors and data processing to output an ABi encoder signal, with 160 counts per magnetic pole pair 2612, and an index pulse at the edge of every pole pair 2612. The digital output signal may be read with LabView compatible hardware, thus simplifying operation. The inclusion of two Hall effect sensors per index ring 2610 may also compensate for variations in magnetic pole size for each pole pair 2612. The AMS magnetic encoder also comes in several form factors. In one example, the encoder may include a radially magnetized magnet on the end of a shaft and may determine shaft position. In another example, the encoder may include a round disk magnet with multiple poles where the sensor is configured to read the axial face of the disk. The AS5304 is traditionally used for linear motion, but may be wrapped around a disk to allow the sensor to read the radial face of a disk. This allows the AS5304 to be implemented in the desired form factor for the transmission. FIGS. 7A and 7B show an exemplary implementation of the magnetic encoder of FIG. 6C as a position encoder 2600. FIG. 7A shows the number of pole pairs 2612 along the index ring 2610 may be divided according to the number of dog teeth. For each dog tooth, the pole pair 2612 may produce an index pulse response and signal response comprising 160 counts of the resultant sinusoid produced by the pole pair 2612. FIG. 7B shows the sensor 2620 may be disposed proximate to the index ring 2610 during operation.

[0167] FIGS. 8A and 8B show an exemplary apparatus 3000 to manufacture the index ring 2610. As shown in FIGS. 8A and 8B, the apparatus 3000 may receive the index ring 2610 for magnetization. The index ring 2610 may comprise an inner disk (e.g., a 3D printed disk) with a 0.5” wide adhesive-backed magnet wrapped around the outer edge of the disk and glued at the joint between the respective ends of the magnet. A piece of tape may be used to help hold the seam together. The apparatus 3000 includes a US Digital 40,000 count optical encoder 3010 to monitor the rotational position of the index ring 2610 and a 1 RPM gear motor to rotate the index ring 2610. The apparatus 3000 may also include a hobbyist servo to move a neodymium magnet into contact with the index ring 2610. The magnet may have a North and South pole that contacts the index ring 2610, thus magnetizing and producing a pole pair 2612 of a desired size. Once the pole pair 2612 is formed, the magnet may be retracted from the index ring 2610, the index ring 2610 may then be rotated to the next pole pair position, and the magnet may then be brought into contact with the index ring 2610 again. The north and south pole pair magnets were fixtured with a spacer to create a 2 mm pole, with each pair resulting in a 4 mm pole pair. A Labview MyRIO was used to operate the apparatus 3000. FIG. 8C shows a completed index ring 2610 overlaid with magnetic paper showing the pole pairs 2612 formed.

[0168] FIGS. 9A-9C show charts characterizing the index ring 2610 formed using the apparatus 3000 of FIGS. 8A and 8B. In particular, FIG. 9A shows a chart of the voltage response of the index ring 2610 as a function of the 40,000 optical encoder 3010 count. As shown, the North and South pole pairs 2612 of the index ring 2610 are distinguishable and substantially similar along the index ring 2610. FIG. 9B shows a chart comparing the encoder counts of the optical encoder 3010 used in FIG. 9A and the magnetic encoder. As shown, the magnetic encoder supports fewer counts, but is nevertheless substantially linear with the optical encoder. FIG. 9C shows the circumferential distance error between the optical and magnetic encoders along the circumference of the index ring 2610, indicating the position error of the magnetic encoder is small.

[0169] This exemplary implementation where the position encoder 2600 is a magnetic encoder may further be implemented without inclusion of a reference feature and/or alignment pin for proper installation into the transmission, thus simplifying the manufacture and assembly of the transmission 2000. Instead, the position of the index ring 2610 with respect to the dog teeth may be calibrated *in situ*.

[0170] In some implementations, the synchronizer 2500 may engage with one of the input gears 2300 when a vehicle leaves the manufacturing line. Measurements of the offset (i.e., the phase or position difference) between the respective dog teeth provides the alignment of the dog teeth when engaged as well as the backlash. This process may be repeated for each subsequent gear in the transmission 2000. The calibrated offsets may then be stored in memory for later retrieval. In this manner, when the vehicle prepares to shift a gear, a processor may retrieve the calibrated offsets from the memory to determine when to initiate the gear shift based on the RPM difference to ensure proper engagement of the dog teeth occurs.

[0171] An Exemplary Method of Shifting Gears Using the Transmission

[0172] The transmission 2000, as described above, is capable of providing a combination of speed and position matching to facilitate the engagement of the synchronizer 2500 to an input gear 2300 during a gear shift. The following provides additional description of the methods and various benefits of using position matching during a gear shift. To provide context for how speed and position matching may be used to improve the gear shift, a description of a full shift event corresponding to a downshift shall now be presented.

[0173] A full shift event of the synchronizer 2500 from the second input gear 2300b to the first input gear 2300a begins by disengaging the second input gear 2300b from the synchronizer 2500 and ending with the engagement of the first input gear 2300a to the synchronizer 2500. In this example, the first input gear 2300a may have a smaller gear ratio than the second input gear 2300b. Thus, the first input gear 2300a may rotate at a higher speed than the second input gear 2300b for a given input shaft speed. FIGS. 10A and 10B show representative charts of the torque and the speed of a power source 2110 (e.g., an EM), respectively, during the full shift event.

[0174] As shown, the shift event starts with applying a torque to accelerate the vehicle using the power source 2110 (in this case an EM) while engaged to the second input gear 2300b. When initiating a gear shift, the torque produced by the power source 2110 should second be reduced to avoid generating unwanted shocks to the vehicle felt by the driver when disengaging the synchronizer 2500 from the second input gear 2300b. For example, the power source 2110 may reduce the torque output to about 0 N·m. Once the torque output of the power source 2110 is reduced, the sleeve 2520 of the synchronizer 2500 is disengaged from the second input gear 2300b and moved to a neutral position using the actuator 2700. The neutral position corresponds to a position where the synchronizer 2500 applies a substantially reduced or, in some instances, negligible torque to the second input gear 2300b and the first input gear 2300a.

[0175] Once the synchronizer 2500 is in neutral, a speed matching phase begins where the power source 2110 applies a torque to accelerate the synchronizer 2500 (or decelerate in the case of a downshift) until the speed of the synchronizer 2500 is similar to the first input gear 2300a. As the speed of the synchronizer 2500 approaches the speed of the first input gear 2300a, the torque generated by the power source 2110 is reduced in order to prevent the speed of the synchronizer 2500 from overshooting the first input gear 2300a.

[0176] When the speed of the synchronizer 2500 is similar to the first input gear 2300a, a position matching phase may begin by monitoring the positions of the dog teeth of the synchronizer 2500 and the first input gear 2300a, respectively, using the position encoders 2600a and 2600b. In some implementations, the positions of the dog teeth may be monitored throughout the shift event to at least monitor the respective speeds of the synchronizer 2500 and the input gear 2300. Once the difference in speeds between the synchronizer 2500 and the first input gear 2300a are within a desired threshold and the position of the dog teeth are aligned such that the engagement occurs without an unwanted collision, the sleeve 2520 of the synchronizer 2500 is moved to engage the first input gear 2300a using the actuator 2700. After the first input gear 2300a is engaged, the power source 2110 may apply a torque to continue accelerating the vehicle in the first input gear 2300a.

[0177] It should be appreciated the methodology of performing an upshift from the first input gear 2300a to the second input gear 2300b may be substantially similar to the process depicted in FIGS. 10A and 10B. The primary difference during an upshift is that the power source 2110 may be decelerated while in the neutral position since the second input gear 2300b may be at a slower speed compared to the first input gear 2300a.

[0178] The present disclosure is primarily concerned with the speed and position matching phases of the full shift event. As described above, the controller 2800 is used, in part, to receive the position measurements from each position encoder 2600, use the position measurements to perform speed and/or position matching of the input shaft 2100 and the input gear 2300, and to generate an actuator signal to initiate a gear shift when said speed and/or position matching between the input shaft 2100 and the input gear 2300 is achieved. The following disclosure describes one exemplary method where position and speed matching are applied to a transmission 2000 that uses dog teeth to engage the input shaft 2100 to the input gear 2300. However, it should be appreciated that the various processes and steps described with respect to this particular exemplary implementation may be readily adapted and/or otherwise incorporated into other speed and position matching methods and coupling mechanisms.

[0179] The controller 2800 may generally receive as inputs: (1) the geometry of each dog tooth, (2) the geometry of the dog teeth (e.g., radius, pitch), (3) the velocity of the actuator 2700, (4) the distance between the dog teeth, and/or (5) the relative speeds of the dog teeth being engaged. Based on these inputs, the respective positions of the dog teeth associated with the input shaft 2100 and the input gear 2300 may be monitored as a function of time.

[0180] FIG. 11A shows a simplified representation of a rotating element (e.g., a gear or a shaft) of the transmission 2000. The rotating element may rotate about an axis, resulting in a circular motion path. For the purposes of illustrating the relative alignment between respective dog teeth of the input shaft 2100 and the input gear 2300, the circular motion path may be unwrapped into a linear motion path. The dog teeth may thus periodically slide past a reference point to indicate the respective ends of the circular motion path. It should be appreciated that the controller 2800 may account for the curvature of the gear and/or shaft when determining the relative position and speed of the respective dog teeth.

[0181] FIG. 11B shows exemplary dog teeth 2310a and 2310b associated with the input shaft 2100 and the input gear 2300, respectively. In particular, the dog teeth 2310 may be coupled to the synchronizer 2500 and, hence, moved into engagement with the dog teeth 2310 via the actuator 2700. As before, the dog teeth 2310 may be treated as being fixed relative to the motion of the dog teeth 2310. However, both the dog teeth 2310 may generally be moving. Each dog tooth 2311 includes a base portion 2312 and a tip portion 2314. A square envelope may be defined around the base and tip portions 2312 and 2314 to provide a safety buffer for the shift event and to simplify the calculations necessary in real-time.

[0182] FIG. 11B shows that successful engagement may occur when the respective base portions 2312 of the dog teeth 2310a and 2310b contact. If only the tip portions 2314 contact, the dog teeth 2310a and 2310b are considered not engaged. As shown in Inset 2 of FIG. 11B, successful engagement may be based on the dog teeth 2310a and 2310b being at least partially engaged. The process of achieving successful engagement of the dog teeth 2310a and 2310b may also depend on several criteria including, but not limited to the degree of engagement between the respective dog teeth (e.g., the dog teeth may be preferably fully engaged), avoiding undesirable collisions of the dog teeth 2310a and 2310b, and ensuring any collisions (e.g., axial collisions of dog teeth when first making contact) do not generate acoustic noise that exceeds a desired threshold. For example, a gear shift initiated too early or too late may cause the tip portion 2314 of each dog tooth 2311 in the dog teeth 2310a and 2310b to collide and subsequently slide into engagement. As described above, this may lead to unwanted acoustic shock and noise in the vehicle and, hence is preferably avoided.

[0183] It should be appreciated the criteria may be defined and evaluated in various ways while still accomplishing the same desired outcome. For instance, the criteria may be to engage the dog teeth 2310a and 2310b such that undesirable collisions do not occur. In one example, the criteria may be satisfied if the tip portions 2314 of the dog teeth 2310a and 2310b do not contact one another during a gear shift. This may be evaluated by tracking the motion of a reference point on the dog tooth 2311 (e.g., the corners between the base and tip portions 2312 and 2314, the corners of the square envelope, a centroid position of the dog tooth 2311). In another example, the criteria may instead be satisfied if only the base portions 2312 of the dog teeth 2310a and 2310b make contact. This approach may be used in cases where the dog tooth 2311 does not have a tip portion 2314. In yet another example, the criteria may be satisfied if the dog teeth 2310a and 2310b are engaged such that acoustic noise does not exceed a desired threshold (e.g., a NVH limit).

[0184] FIG. 11C shows that the relative position and alignment of the dog teeth 2310a and 2310b may be tracked by measuring a phase difference between the signal responses recorded by the respective position encoders 2600. As shown, the phase difference may be defined, in part, by the angle and/or the circumferential distance between the dog teeth 2310a and 2310b. The phase difference between the dog teeth 2310a and 2310b may include an offset (e.g., defined when the dog teeth 2310a and 2310b are engaged as described previously). FIG. 11C shows three exemplary positions of the dog teeth 2310 relative to the dog teeth 2310 at three different times where the phase difference increases. As shown in FIG. 11C, the phase difference periodically resets to zero as the dog teeth 2310a and 2310b move relative to each other to indicate one dog tooth in the dog teeth 2310 moving past the dog teeth 2310 such that another, identical dog tooth in the dog teeth is now in the same position as the one dog tooth.

[0185] The controller 2800 may derive the phase difference based on position measurements from the respective position encoders 2600 associated with the input shaft 2100 and the input gear 2300. The phase difference may be used, in part, to monitor the alignment between the dog teeth 2310a and 2310b in order to determine when a gear shift should be initiated. For example, the controller

2800 may use the phase difference, the relative speed between the dog teeth 2310a and 2310b, the shift velocity of the actuator 2700, and the distance between the dog teeth 2310a and 2310b to predict a trajectory (e.g., a motion path) of the dog teeth 2310 when a gear shift is initiated. The trajectory may be used, in part, to determine whether the dog teeth 2310a and 2310b can be successfully engaged, whether undesirable collisions occur, and the degree of engagement between the dog teeth 2310a and 2310b.

[0186] The trajectory of the dog teeth 2310a and 2310b may be determined using a combination of vector summation and a system of equations. In some implementations, the corners of each dog tooth (i.e., the corners of the square envelope) may move with a velocity vector V (e.g., V_1 for corner 1 and V_2 for corner 2) as shown in FIG. 11B. The velocity vector V is the sum of the vector motion comprising a vertical component, V_v, representing the shift velocity of the actuator 2700, and a horizontal component, V_H, representing the velocity difference between the two sets of dog teeth 2310a and 2310b. The velocity vector V may thus be broken into two components as follows,

$$\begin{cases} V_H = \Delta V = V_{H1} - V_{H2} \\ V_v = V_{act} \end{cases} \tag{1}$$

where V_{H1} is the speed of dog teeth 2310, and V_{H2} is the speed of dog teeth 2310, and V_{act} is the speed of the actuator 2700.

[0187] If a dog tooth in the dog teeth 2310 starts at a horizontal position C, the system equations for V_1 may be represented as follows,

$$\begin{cases} C_{min} \leq C + \int_t^{t+t_{close}} V_H dt \\ d_{gap} = \int_{t_0}^{t_{close}} V_v dt \end{cases} \tag{2}$$

where C_{min} is the width of a dog tooth, t is the start time, and t_{close} is the time to move the actuator a distance d_{gap}. Solving Eq. (2) and rearranging terms leads to the following result,

$$\begin{cases} C \geq C_{min} - V_H t_{close} \\ t_{close} = \frac{d_{gap}}{V_v} \end{cases} \tag{3}$$

where t_0 is assumed equal to zero.

[0188] The system of equations for V_2 may be solved in a similar manner. In this case, the actuator

2700 should move the dog teeth 2310 a distance of d_{tot} in a time t_{shift} as follows,

$$\begin{cases} C + C_{min} \leq C_{max} - V_H t_{shift} \\ t_{shift} = \frac{d_{tot}}{V_v} \end{cases} \tag{4}$$

where $C + C_{min}$ is the starting position of the bottom right corner of the dog tooth in dog teeth 2310, C_{max} is the beginning of the next tooth, and d_{tot} is the vertical distance to travel before the dog teeth 2310 begin lateral contact.

[0189] FIGS. 12A-12C show exemplary trajectories of the dog teeth 2310 when a gear shift is initiated at three different phase differences. FIG. 12A shows a gear shift initiated at a phase difference that results in an undesirable collision between the dog teeth 2310a and 2310b. In this case, the collision may be determined based on contact between the respective corners of the dog tooth 2311 (where the base portion 2312 joins the tip portion 2314). However, an undesirable collision may also be defined as physical contact between the respective tip portions 2314 of each dog tooth 2311. Although the dog teeth 2310a and 2310b may still engage after the collision, the resulting shock is undesirable and should preferably be avoided.

[0190] FIG. 12B shows a gear shift initiated at a phase difference that results in the full engagement of the dog teeth 2310a and 2310b. The engagement of the dog teeth 2310a and 2310b may be defined based on the extent in which the respective base portions 2312 of the dog teeth 2310a and 2310b are in contact. A full engagement may correspond to contact between the respective base portions 2312 of each dog tooth 2311 being limited by the geometry of each dog tooth 2311. In this case, contact between the base portions 2312 of each dog tooth 2311 is limited by the respective tip portions 2314 touching the opposing dog teeth.

[0191] It should be appreciated that when the respective base portions 2312 contact, an acoustic shock or sound may be produced. This acoustic shock is to be distinguished acoustic shocks produced when the tip portions 2314 of the dog teeth collide. In some implementations, the acoustic shock produced when the respective base portions 2312 contact may be reduced by adjusting the shift velocity of the actuator 2700. For example, the shift velocity may decrease immediately before the respective base portions 2312 to prevent an acoustic shock that exceeds a desired threshold (e.g., the NVH limits of a vehicle). FIG. 12B further shows the dog teeth 2310a and 2310b may be spaced apart by a distance larger than the width of each dog tooth 2311. This additional spacing may allow for a range of phase differences that lead to the full engagement of the dog teeth 2310a and 2310b. The phase difference indicated in FIG. 12B represents one limit of said phase difference range.

[0192] FIG. 12C shows a gear shift initiated at a phase difference that occurs later than the phase difference of FIG. 12B (e.g., a late gear shift) as indicated. As shown, the dog teeth 2310a and 2310b may still engage, but the contact between the respective base portions 2312 of the dog teeth 2310a and 2310b may be substantially reduced. Although a partial engagement of the dog teeth 2310a and 2310b may still be sufficient in transferring torque between the input shaft 2100 and output shaft 2200, there may be cases where this is not preferable. For example, rough road conditions may generate substantial vibration within a transmission that may cause the dog teeth 2310a and 2310b to prematurely disengage.

[0193] Based on FIGS. 12A-12C, the phase difference spanning one period of a dog tooth in the dog teeth 2310 moving past a dog tooth in the dog teeth 2310 may be divided into three ranges: (1) a first phase difference range that results in an undesirable collision, (2) a second phase difference range that results in the full engagement of the dog teeth 2310a and 2310b, and (3) a third phase difference range that results in the partial engagement of the dog teeth 2310a and 2310b. As described above, the phase difference periodically resets to zero when one period is completed, thus the phase difference will continuously transition between these respective phase difference ranges.

[0194] The controller 2800 may define a shift window to represent the phase difference range in which the dog teeth 2310a and 2310b can be engaged (e.g., the second phase range, the second and third phase ranges). During a gear shift, the controller 2800 may thus use the shift window as a way of determining whether the dog teeth 2310a and 2310b are sufficiently position and speed matched to engage the dog teeth 2310a and 2310b. In this manner, the shift window provides an objective metric to ensure the dog teeth 2310a and 2310b are engaged in a desired manner as quickly as possible. Said in another way, the controller 2800 may initiate a gear shift when the dog teeth 2310a and 2310b are sufficiently position and speed matched rather than waiting for the speed difference to reduce to zero. The controller 2800 may also incorporate additional criteria to modify the shift window including, but not limited to the degree of engagement of the dog teeth 2310a and 2310b, the magnitude of any acoustic shock produced when engaging the dog teeth

2310a and 2310b, and limitations of the hardware (e.g., limitations on the shift velocity, lag time in the electronics, mechanical stresses incurred by the dog teeth 2310a and 2310b).

[0195] The criteria and the calculations used to determine the trajectory of the dog teeth may be implemented in real-time. By tracking the relative position of the dog teeth 2310a and 2310b, the initial offset, C, the velocities of the respective dog teeth 2310a and 2310b, V_{H1} and V_{H2}, and any known constants, the controller 2800 may determine when to initiate a gear shift. As the ratio of V_H to V_v increases, the duration of the shift window decreases. Beyond a certain threshold, a successful gear shift may not be possible. FIG. 11D shows an exemplary shift window for a 0.5 m/s shift actuator velocity. Referring to FIG. 11B, a collision between the dog teeth 2310a and 2310b based on the V_1 vector occurs to the left of the V_1 line. Insufficient engagement between the dog teeth 2310a and 2310b based on the V_2 vector occurs to the right of the V_2 line. This defines a region of FIG. 11D in which the dog teeth 2310a and 2310b are considered sufficiently aligned to achieve proper engagement.

[0196] The controller 2800 may also manipulate the duration of the shift window to further reduce the shift time. Generally, the duration of the shift window may depend not only on the geometry of the dog teeth 2310a and 2310b (e.g., the pitch and height), but also the speed difference between the input shaft 2100 and the input gear 2300 and the shift velocity of the actuator 2700. During a gear shift, the controller 2800 may actively adjust the speed difference by adjusting the amount of torque applied by the power source 2110 and/or the shift velocity of the actuator 2700. For example, decreasing the speed difference more quickly and/or increasing the shift actuator velocity, the shift window may appear more quickly and last for a longer period of time. Again, the extent to which the shift window is modified may depend on the criteria described above.

[0197] For example, the speed difference between the dog teeth 2310a and 2310b may be reduced in a monotonic manner as a function of time (e.g., the speed of the input shaft 2100 continuously increases to match the speed of the input gear 2300 during a downshift), the shift window may progressively increase in duration as the speed difference is reduced. Said in another way, the phase difference between the dog teeth 2310a and 2310b may remain in the second and/or third phase difference ranges for a longer period of time when the speeds are more closely matched. However, a lower speed difference may also increase the period of time between shift windows since the phase difference correspondingly remains in the first phase difference range longer. In some cases, the speed difference may be reduced to zero and the dog teeth 2310a and 2310b may be misaligned such that the phase difference remains in the first phase difference range, thus preventing engagement. In the event a gear shift is not initiated before the speed difference reaches zero, the controller 2800 may command the power source 2110 to apply a torque to the input shaft 2100 to alter the alignment of the dog teeth 2310a and 2310b such that the phase difference is moved into the shift window.

[0198] In some implementations, the controller 2800 (also referred to herein as “the transmission control unit (TCU)”) may use a state-machine method. The state-machine method defines a sequence of states and transitions for performing the gear shift. FIG. 13 shows one exemplary state-machine method to perform a downshift. In this example, the actuator 2700 comprises Piston 1 and Piston 2 in a stacked configuration. Three position states (i.e., *Gear 2*, *Neutral*, and *Gear 1*) are used to define the control functions based on the position of the sleeve 2520. Three action states (i.e., *Disengagement*, *Synchronization*, and *Engagement*) are used to determine the control actions during the shift event. The controller 2800 is initialized in one of the three position states based on the position of the sleeve 2520,

[0199] Initially, when in the *Gear 2* position state, power source 2110 (e.g., an EM) may provide a sufficient torque for the vehicle to accelerate or to maintain a constant velocity as commanded by the vehicle or driver. When the shift event begins (e.g., initiated by a user interface (e.g., a LabVIEW interface), or a transmission shift supervisor in the vehicle), the *Disengagement* action state is activated and the upper limit to the torque output from the power source 2110 is progressively reduced (e.g., linearly or exponentially). A reduction to the torque output allows the dog teeth of the sleeve 2520 to disengage from the second input gear 2300b without creating undesirable shocks to the drivetrain. Once the measured torque output falls a desired value, Piston 1 of the actuator 2700 may be turned on, disengaging the sleeve 2520 from the second input gear 2300b.

[0200] Based on the measured position of the sleeve 2520, the controller 2800 transitions to the *Neutral* position state. Here, the upper limit to the torque output is reset to its original value prior to disengagement. The *Synchronization* action state then occurs where the power source 2110 adjusts the speed of the synchronizer 2500 to match the speed of the first input gear 2300a. The manner in which the power source 2110 adjusts its torque output during this speed matching phase may vary. In some implementations, the torque may be ramped to the upper limit as quickly as possible in order to reduce the shift time. In some implementations, the torque may be adjusted such that the speed of the input shaft 2100 is accelerated/decelerated smoothly, thus limiting the torsional stress imparted on the input shaft 2100.

[0201] Once the input shaft 2100 and the first input gear 2300a are sufficiently speed matched, the controller 2800 may then determine the shift window and wait for the phase difference between the respective dog teeth to be within the shift window. Again, the shift window may be defined to avoid undesirable collisions between the respective dog teeth of the sleeve 2520 and the first input gear 2300. Once this condition is met, the *Engagement* action state is activated where Piston 2 of the actuator 2700 is fired to engage the synchronizer 2500 to the first input gear 2300a. Once the synchronizer 2500 and the first input gear 2300a are fully engaged, the controller 2800 transitions to the *Gear 1* position state.

[0202] FIGS. 13A-13F show one exemplary gear shift that incorporates both speed and position matching during a gear shift. FIG. 13A shows the RPM of the power source 2110 (e.g., the EM coupled to the synchronizer 2500) and the RPM of the input gear 2300 to be engaged. This example represents an upshift, hence, the RPM of the power source 2110 is initially slower than the input gear 2300, but increases with time in a monotonic manner. The RPM of the input gear 2300 is sinusoidal to show the speed of the input gear 2300 may oscillate about a nominal value during operation of the vehicle due to various irregularities (e.g., friction effects, engine firing effects).

[0203] FIG. 13B shows a chart of the RPM difference (i.e., the speed difference) between the synchronizer 2500 and the input gear 2300. The oscillatory behavior of the curve is due to the oscillations in RPM of the input gear 2300. As shown, the acceleration of the synchronizer 2500 reduces the speed difference to such an extent that the speed difference crosses zero. A negative speed difference indicates the synchronizer 2500 and the input gear 2300 are experiencing relative motion in a direction opposite to when the speed matching first began.

[0204] FIG. 13C shows the phase difference between the synchronizer 2500 and the input gear 2300 as a function of time. For this example, the dog teeth of the sleeve 2520 and the input gear 2300 are representative of a typical dog tooth design having base and tip portions. The shift velocity is assumed to be 0.5 m/s. As shown, the phase difference periodically resets to zero as the respective dog teeth of the sleeve 2520 on the synchronizer 2500 and the input gear 2300 move relative to one another and completes a period (five periods are shown in FIG. 13C). Each period further contains a shift window defined based on shift window start and end curves (e.g., a lower and upper limit to the second and third phase difference ranges previously described). The location of the shift window within each period is dependent on the initial offset of the dog teeth of the sleeve 2520 and the input gear 2300. As shown, the duration of each shift window (e.g., from shift window A to shift window E) increase as the speed difference decreases. However, the period between each shift window correspondingly increases as well.

[0205] FIG. 13D shows the position and duration of each shift window as a function of time. In some implementations, the controller 2800 may initiate a gear shift at the beginning of the shift window A (i.e., the first shift window that appears) in order to reduce the gear shift time. However, in some implementations, other operating factors may render the other shift windows to be more desirable. For example, a longer shift window indicates a gear shift may be performed at lower shift velocities, which can be beneficial in terms of reducing acoustic shock and/or noise in the vehicle.

[0206] FIG. 13E shows the shift time as a function of the speed difference between the synchronizer 2500 and the input gear 2300. The shift time is based, in part, on the shift velocity of 0.5 m/s and the distance separating the dog teeth of the sleeve 2520 and the input gear 2300 for engagement. Based on the speed matching profile of FIG. 13A, the shift time increases as the speed difference decreases due to the longer period of time to achieve the smaller speed difference.

[0207] FIG. 13F shows the degree of engagement between the dog teeth of the sleeve 2520 of the synchronizer 2500 and the input gear 2300. As shown in the inset, the dog teeth may be fully engaged at the beginning of the shift window (i.e., the second phase difference range) and partially engaged thereafter (i.e., the third phase difference range). FIG. 13F shows that the degree of vertical engagement is dictated by when the gear is initiated within the shift window.

[0208] In some implementations, speed and position matching may also be used to compensate for speed irregularities in the transmission 2000 during a gear shift. A speed irregularity may be defined as a deviation from a nominal (or commanded) speed in the input shaft 2100 and/or the output shaft 2200. Speed irregularities may be caused by systematic or random mechanical noise in the input shaft 2100 and/or the output shaft 2200. For example, a misalignment between a piston and a crankshaft in an ICE may produce systematic vibrations in the drive shaft that are transferred to the input shaft 2100. The period of the vibrational noise may be correlated to the speed or RPM of the ICE. In another example, the vehicle may be driving on a gravel road. The rough road conditions may produce random vibrations through the wheels and/or suspension of the vehicle that are transferred to the output shaft 2200.

[0209] In some implementations, if the position encoder 2600 has a sufficiently fine temporal resolution (e.g., or a large frequency range) to monitor the presence of speed irregularities in the transmission 2000, the controller 2800 may modify the shift window to at least partially compensate for said speed irregularities. For example, the presence of random mechanical noise may increase the uncertainty in position and/or the speed measurements of a shaft or input gear. One approach to compensate for said increase in uncertainty may be to reduce the duration of the shift window. For instance, the controller 2800 may compute a shift window based on nominal measurements of speed and position (e.g., an average or mean). If the random noise is approximated by a Gaussian distribution, the controller 2800 may reduce the duration of the shift window by a factor dependent on the standard deviation of said Gaussian distribution such that the modified shift window provides a probability of a successful gear shift above a desired threshold (e.g., greater than 99% probability of successful engagement).

[0210] In another example, the presence of systematic noise may be such that the respective dog teeth of the input shaft 2100 and the input gear 2300 may move relative to one another in an oscillatory manner (i.e., the dog teeth may transition between clockwise and counterclockwise motion). FIG. 15A shows the RPM difference as a function of time for an exemplary case where the speed irregularity causes the RPM difference to oscillate about zero. If the systematic noise is repeatable over at least a few periods, the controller 2800 may determine a modified trajectory based on (1) the phase difference (e.g., the alignment between the dog teeth) and (2) the expected oscillatory motion between the dog teeth if a gear shift is initiated.

[0211] FIG. 15B shows exemplary trajectories of the dog teeth 2310 computed at various phase differences and various times along the RPM difference curve of FIG. 15A. For instance, the phase differences A and B occur at similar positions along the sinusoidal RPM difference curve of FIG. 15A, resulting in a trajectory that is similar in shape, but offset in alignment. The phase difference

C, however, results in a trajectory that deviates in shape and alignment from the trajectories corresponding to phase differences A and B. As shown, the controller 2800 may use previous measurements of the speed irregularity and the present alignment of the dog teeth to predict when to initiate a gear shift.

[0212] FIG. 16 shows an exemplary approach of predicting the motion of the dog teeth when a speed irregularity is present. In one approach, the controller 2800 may use a curve that is fitted to the measured RPM difference to predict the expected position and speed difference at a future time. This may be followed by an evaluation of whether said future position and speed difference may result in a successful gear shift. For the exemplary RPM difference curve of FIG. 16, the controller 2800 may estimate the amplitude(s) and frequency(s) of the speed irregularity and fit a sine wave to estimate future speed differences. The predicted position and speed differences may then be used to estimate the trajectories of the respective dog teeth. If the controller 2800 determines a successful engagement is possible, the controller 2800 may then initiate a gear shift. In some implementations, the controller 2800 may monitor the position and speed difference for several periods to verify and/or update the sine wave to increase the confidence of a successful gear shift.

[0213] A speed irregularity may also be partially cancelled out by applying a variable torque to the transmission 2000. In some implementations, this may be accomplished using a friction element and/or a damper to reduce the amplitude (e.g., smear out) the effects of the speed irregularity. In some implementations, the power source 2110 may be used to output a time-varying torque to reduce a measured speed irregularity. This may be accomplished by forming a closed feedback loop between the controller 2800, the position encoder 2600, the power source 2110, and the actuator 2700. The closed feedback loop may be used to actively adjust the torque output of the power source 2110 such that a desired speed and/or position difference is maintained.

[0214] The use of a power source 2110 to reduce the effects of speed irregularities, however, may be limited by the temporal response to torque adjustments of said power source 2110. This may depend on several factors including, but not limited to the inertial loads applied to the power source 2110 and system delays. For instance, FIG. 17A and 17B shows on exemplary case where the power source 2110 exhibits no torque ramp limit. Thus, the power source 2110 is able to instantaneously change its torque output such that the speed of the power source 2110 follows the speed irregularity as shown in FIG. 17A. FIG. 17B shows the torque may encounter an upper limit under these conditions. FIG. 18 shows an exemplary case where the power source 2110 has a finite response time longer than the period of the speed irregularity. As a result, the power source 2110 may only output a torque that partially compensate the speed irregularity.

[0215] A First Exemplary Demonstration of Position and Speed Matching in a Transmission

[0216] The following disclosure describes an exemplary implementation of the transmission 2000 performed as a laboratory experiment. FIG. 19A shows a benchtop setup 4000a representing a two-speed transmission built to perform a gear shift using a combination of speed and position matching. A first motor 4010 is used to represent the power source 2110 (e.g., an electric motor) of the transmission 2000. A second motor 4020 is used to represent the load 2210 of a vehicle moving at an approximately constant velocity during the shift event. The setup 4000a includes the input shaft 2100 coupled to the first motor 4010 and the output shaft 2200 coupled to the second motor 4020. The setup 4000a includes the first and second input gears 2300a and 2300b mounted to the input shaft 2100 and the first and second output gears 2400a and 2400b mounted to the output shaft 2200. A synchronizer 2500 comprising a hub 2510 and a sleeve 2520 is mounted to the input shaft 2100. The setup 4000a further includes a first position encoder 2600a comprising the first index ring 2610a mounted to the input shaft 2100 and the first sensor 2620a mounted proximate to the first index ring 2610a. A second position encoder 2600b is included where the second index ring 2610b is mounted to the second input gear 2300b and the second sensor 2620b is mounted proximate to the second index ring 2610b. The setup 4000a also includes the actuator 2700.

[0217] FIGS. 19B-19D show several images of the synchronizer 2500 and the first input gear 2300a during a gear shift. In particular, FIG. 19B shows the sleeve 2520 of the synchronizer 2500 in the neutral position. FIG. 19C shows the dog teeth of the sleeve 2520 being speed and position matched to the dog teeth of the first input gear 2300a during the synchronization phase of the gear shift. FIG. 19D shows the sleeve 2520 being moved by the actuator 2700 into engagement with the first input gear 2300a.

[0218] The first and second index rings 2610a and 2610b were fabricated using the system 3000 of FIGS. 8A and 8B. Specifically, the first and second index rings 2610a and 2610b comprised a

0.5 inch wide adhesive backed refrigerator magnet wrapped around a 3D printed wheel. The seam of the magnetic strip was glued and covered by a piece of tape. The first and second sensors 2620a and 2620b are the AS5304 magnetic Hall effect sensor of FIGS. 7A and 7B. These sensors include two internal Hall effect sensors and advanced data processing to output an ABI encoder signal, with 160 counts per magnetic pole pair, and an index pulse at the edge of every pole pair.

[0219] The first and second motors 4010 and 4020 were each Teknic Clearpath CPM-MCVC-3421S-RLS motors. These motors were selected for their high torque and lower speed. The purpose of the setup 4000a was to perform a gear shift at lower speeds, hence higher speeds were not necessary. Additionally, these motors included integrated motor drivers within the motor body that receive only a control logic signal and a 72 V power supply. One benefit of these Clearpath motor/controller series is the ability to control the torque and speed of the motor simultaneously and independently. For instance, a pulse width modulated (PWM) signal may control the closed loop motor speed and a second signal may be configured to act as a torque limit for the motor. If the torque limit is reduced such that the torque is no longer sufficient to meet the speed set point, the motor will slow down and follow the torque limit command (i.e., the motor operates in a closed loop manner based on the torque). This method of operation is preferable since the motor should rapidly switch between speed and torque modes during a gear shift.

[0220] The actuator 2700 is a pneumatic actuator comprising two pistons arranged in a stacked configuration as previously described and shown in FIG. 2C.

[0221] The controller 2800 was a National Instruments CompactRIO (cRIO) 9042, which was used to control the benchtop setup 4000a by replicating the vehicle's TCU. A field programmable gate array (FPGA) on the cRIO was used to record signals from the position encoders 2600a and 2600b. The FPGA was chosen due to the high data rate provided by the encoders (e.g., sampling frequency is about 40 MHz). Furthermore, the controller 2800 was also used to record the overall piston force, the motor speed, and the motor torque at a lower sampling rate (e.g., 1 kHz). Additionally, the controller 2800 was used to control the actuator 2700 during the shift event. During operation, the controller 2800 adjusts the speed and torque setpoints for the power source 2110 and the shift sleeve position of the synchronizer 2500 via the actuator 2700 in order to meet a desired wheel speed. LabVIEW was used to control the benchtop setup 4000a and to record data. FIGS. 19E and 19F show portions of the LabVIEW code used to control the FPGA and a central processing unit (CPU), e.g., the cRIO, in the setup 4000a. FIGS. 19G and 19H show portions of an exemplary graphical user interface (GUI) used to control and monitor the setup 4000a.

[0222] The controller 2800 further included custom printed circuit boards (PCBs) fabricated by JLCPCB. Each PCB was designed to be plugged into the top of a module in the cRio. For instance, a National Instruments (NI) 9401 module was used to write a PWM signal to control the first and second motors 4010 and 4020. Several 9 pin D-sub connectors were used for connection with the first and second motors 4010 and 4020, which each have 8 connection leads. The first and second motors 4010 and 4020 received inputs including enable, speed, and torque. The output included a reconfigurable signal corresponding to various motor-related parameters, such as the output motor torque. The motor PCB contained two integrated circuits (ICs). A bus converter was used for the signals that drive the first and second motors 4010 and 4020. The bus converter converts the 3.3 V, 1 mA signal from the 9401 module into a 5 V, 7 mA signal to drive opto-isolators within the motor driver of each motor. The motor was designed in this manner to prevent potential damage to the motor driver board, which may otherwise result in the electrical shorting at high voltage of the components in the motor driver used for reading and writing, such as the National Instruments hardware.

[0223] At least one signal from the first and second motors 4010 and 4020 was read by the LabVIEW code. For instance, each motor was configured to output the torque value in the form of a duty cycle within a fixed frequency square wave. A Schmitt trigger was used to clean the motor output signal (e.g., reduce noise). Without the Schmitt trigger, the output signal included noise within the square waveform when viewed with an oscilloscope, which led to errors when evaluating the duty cycle (and therefore the torque value) within LabVIEW. The Schmitt trigger was able to improve the signal to noise ratio and was shown to be able to output a high signal when the voltage exceeded 3 V and a low signal when the signal was below 1 V.

[0224] A physical switch was added in-line with each motor's enable signal. Thus, the first and second motors 4010 and 4020 were able to be disabled either by a corresponding GUI button in the LabVIEW code or by physically flipping the switch.

[0225] A second NI 9401 module was used to read the encoder pulses and motor torque values. After the output signal of each motor was processed by the Schmitt trigger, the signal was passed to the second NI 9401 PCB. The position encoders 2600a and 2600b generates three digital signals: A, B, and Index. These signals pass over RJ45 cables, which were designed for high data rates, and directly into LabVIEW for processing. Custom boards for the AMS5304 encoder IC were also designed. These boards allowed for proper positioning of the sensor 2620 within each position encoder 2600 to within 1 mm of the index rings 2610.

[0226] A 9381 module was used to control the solenoid valves and read analog input of the shift actuator position sensor and force gauge in the actuator 2700. A string potentiometer was used to measure the position. The potentiometer was selected for its ease of implementation within the setup 4000a. An Omega load cell and a Tacuna Systems amplifier were used to measure the shift force.

[0227] FIG. 20A shows an exemplary method of controlling the setup 4000a during a full shift event. As shown, the first motor 4010 may be used to accelerate the vehicle (e.g., a driver is pressing on the accelerator pedal to increase the vehicle's speed). Just before a gear shift is initiated, the speed of the vehicle may be approximated as being constant. In order to emulate these conditions in the setup 4000a, the second motor 4020 (represented by "wheel speed") may be commanded to run at a constant speed with a large gain to maintain said speed setpoint. A large gain term is used to emulate the large inertia of a vehicle moving forward at an approximately constant velocity. In this manner, the speed setpoint of the first motor 4010 (represented by "motor") may be set higher than the speed setpoint of the second motor 4020 in order for a torque to be transferred through the setup 4000a while maintaining a substantially constant speed in the second motor 4020.

[0228] The first motor 4010 may transfer a torque to the output shaft 2200 via the synchronizer 2500 (i.e., when the first motor 4010 is accelerating the vehicle) by commanding the speed of the first motor 4010 to be greater than the wheel speed. This is intended to emulate a driver pressing the accelerator pedal, which sets a higher desired motor speed setpoint that the vehicle may eventually reach under acceleration. During this phase, the setup 4000a is operating in a torque mode.

[0229] When a shift event is initiated, the first motor 4010 should first decrease its torque output so the synchronizer 2500 may disengage from the first input gear 2300a. On the benchtop setup

4000a, this process may be implemented by decreasing the torque limit on the first motor 4010. The velocity of the first motor 4010 is chosen based on the gear ratio and the velocity of the second motor 4020 (e.g., representing the vehicle) such that a reduction in the torque output of the first motor 4010 does not change the speed of the input shaft 2100. Once the desired torque value is reached, a shift to neutral is triggered resulting in the sleeve 2520 moving to a neutral position using the actuator 2700. Additionally, the desired speed of the first motor 4010 is set to the wheel speed (as opposed to the previously higher speed setpoint) and the torque limit is restored to its previous value as the lower value is no longer necessary.

[0230] During the speed matching phase of the gearshift, the speed of the first motor 4010 is reduced to a value sufficient to engage the second input gear 2300. The change in speed of the first motor 4010 may occur as quickly as desired with the deceleration rate being limited by the available motor torque and the inertia of the transmission shaft that is changing speed (e.g., the input shaft 2100). As the speed of the first motor 4010 decreases and approaches the speed of the second input gear 2300, the controller 2800 may continuously monitor the two criteria to initiate a shift event: (1) the speed difference should be less than a set value and (2) the dog teeth of the sleeve 2520 and the second input gear 2300b should be aligned. The set value for the speed difference may be changed as desired for each shift event. For example, the speed difference may vary based on road conditions, the gear ratio, and speed irregularities in the input shaft 2100 or the output shaft 2200. When the controller 2800 determines that both conditions are true, a shift to engage the sleeve 2520 to the second input gear 2300b is triggered.

[0231] FIG. 20B shows exemplary data of the measured speed and the speed setpoint of the first motor 4010 collected by the setup 4000a during a shift event. As shown, the measured speed and the speed setpoint initially differ in accordance to the shift method of FIG. 20A. FIG. 21A shows the measured speed difference between the input shaft 2100 and the second input gear 2300b as a function of time. For this example, the shift event was configured to have a speed difference threshold of 4 RPM. FIG. 21B shows the position difference between the respective dog teeth of the input shaft 2100 and the second input gear 2300b as a function of time. As shown, once the speed difference fell below the desired threshold, the dog teeth moved approximately by one dog tooth relative to each other before returning to its original alignment. For this example, the desired position difference corresponding to the shift window was determined to be between 20 and 30 counts.

[0232] FIGS. 22A-22C shows the position and force of the actuator 2700 during the shift event. In particular, FIG. 22A shows the position of the actuator 2700 as a fucntion of time when shifting the setup 4000a from the first input gear 2300ato neutral and from neutral to the second input gear 2300. FIG. 22B shows a magnified view of FIG. 22A where the sleeve 2520 is engaged with the second input gear 2300. As shown, the actuator 2700 was configured to move the sleeve 2520 by approximately 23 mm during the gear shift. FIG. 22C shows the magnitude of the force of the actuator 2700 as a function of time when disengaging the sleeve 2520 from the first input gear 2300a. As shown, the force output undergoes two regimes based on the acceleration and subsequent deceleration of the shift fork.

[0233] As shown and described herein, the method of shifting gears in the transmission 2000 is based on knowledge of the position and speed of the input shaft 2100 and the input gear 2300. In the setup 4000a, the position of the respective dog teeth of the sleeve 2520 (coupled to the input shaft 2100) and the input gear 2300 is measured with a fine resolution encoder sampled at high rate, resulting in an accurate, direct measurement. However, the speed is indirectly estimated with a backward Euler approach. This approach may be subject to large periodic disturbances as shown by the periodic noise in the speed of the first motor 4010 in FIG. 20B, which can affect the precision and/or accuracy of the speed measurement. The performance of the transmission 2000 may be hindered by the inclusion of such noise and should be preferably reduced during operation.

[0234] One approach to reduce this noise may be to filter the estimated speed using a small bandwidth low-pass filter. However, this filter may result in poor performance due to a substantial phase lag. FIG. 23A also shows spectral data of the speed measured by the first position encoder 2600 as a function of frequency. As shown, the noise at steady state is periodic at frequencies proportional to the rotational speed. In other words, peaks in noise are multiples of the main harmonic corresponding to the rotational speed of the first motor 4010. FIG. 23A further shows additional noise distributed across a broad range of frequencies, which may be attributed to instrument noise.

[0235] Another approach is to use an adaptive notch filtering strategy. FIG. 23B shows an illustration summarizing the adaptive notch filter. As shown, the notch filter may include two first-order, low-pass filters configured to attenuate high-frequency components. The first filter has a bandwidth of 200 Hz to remove instrumental noise at high frequency (higher than the fastest dynamics needed to be tracked), thus smoothing out the signal. The second filter is used to estimate the low-varying components of the rotational speed. The second filter thus has a cut-off frequency of 5 Hz corresponding to the main harmonic of the signal. A set of notch filters is then used to smooth out the resonances at frequencies corresponding to multiples of the main harmonic. FIG. 23C shows one exemplary notch filter applied to the frequency spectra of FIG. 23A. A notch filter is preferable for resonances at specific frequencies. The low-varying rotational speeds allow the notch filter harmonics to be readily tuned such that the overall filter may adapt to different operating frequencies. This filter may thus be used for online, real time applications.

[0236] FIG. 23D shows a comparison of the raw and filtered speed measured by the first and second position encoders 2600a and 2600b as a function of time. As shown, the adaptive notch filter qualitatively reduces the amplitude of the noise for both signals. FIG. 23E shows frequency spectra corresponding to the temporal data of FIG. 23D. As shown, high frequency noise is substantially attenuated and noisy peaks corresponding to integer multiples of the main harmonic are also reduced.

[0237] As shown, the adaptive notch filtering approach allows the filtered signal to retain most of the original content without substantial phase lag. Additionally, a signal can be obtained without significant manual tuning of filter parameters due to the adaptive structure of the notch filter. In the current configuration, the only tuning parameter for the notch filters is the damping factor ξ, which determines the smoothness of the signals and the phase-lag. In the current configuration, ξ is set to 0.2.

[0238] It should be appreciated the setup 4000a may be used to simulate various types of gear shifts including, but not limited to an upshift, a downshift, a launch, a smooth torque fill upshift, and a sudden torque fill upshift.

[0239] A Second Exemplary Demonstration of Position and Speed Matching in a Transmission

[0240] FIG. 24A shows another exemplary benchtop setup 4000b used to test and analyze gear shifts with position and speed matching. As shown, the setup 4000b may include two electric motors to drive the setup 4000b. Flywheels may be added to vary the rotational inertia of the setup 4000b. The positions of the dog teeth in the synchronizer 2500 and the input gear 2300 may be determined using position encoders 2600 with variable reluctance sensors. The sleeve and fork of the synchronizer 2500 may be actuated with the actuator 2700 having a pneumatic piston.

[0241] The setup 4000b comprised two electric motors with speed feedback, an overdrive gearset with a single synchronizer from a 1984 Ford F-150 transmission a pneumatic actuator for actuating the shifter, variable reluctance sensors for sensing dog tooth position in the transmission, and the ability to mount flywheels of different inertias. FIG. 24B shows an image of the setup 4000b with the F-150 transmission installed. The setup 4000b was designed to achieve speeds in excess of 3000 RPM, which is greater than the speed difference encountered in a production transmission. Therefore, the setup 4000b is sufficient to test the energy dissipation limits of the synchronizer 2500. Actuation of the synchronizer 2500 may be performed with 1500 N of force for the large inertia shifting mode.

[0242] The setup 4000b was designed to be flexible in its configuration. For example, the setup 4000b was designed such that flywheels of varying inertia may be installed to simulate different vehicle loads on the transmission. Additionally, the synchronizer 2500 may be replaced with other types of synchronizers to test the effects synchronizers with and without friction elements and a varying number of sleeves 2520. Furthermore, the open architecture allows for easy modification of sensor quantity and placement. The sensors 2620 may be mounted to rails that align and hold system components in place. This modular architecture allows for components to be readily replaced with ease.

[0243] FIG. 24C shows the overdrive gearset from a 1984 Ford F-150 transmission. The setup 4000b is designed to be modular, thus the gearset may be readily replaced with a gearset from another vehicle.

[0244] As described above, two motors are used in the setup 4000b to load the transmission in a variety of configurations, such as testing and data collection at both high and low energy conditions. For example, a first test may be performed where data is collected with the input shaft rotating at 100 RPM and the output shaft at 0 RPM. A second test may be conducted to collect data with the input shaft rotating at 3100 RPM and the output shaft at 3000 RPM. Both tests investigate a speed differential of 100 RPM, however the latter test should theoretically have the synchronizer 2500 dissipate approximately 60 times the amount of energy in the former test. Both of these cases may be applied to real world situations, hence the reason for the setup 4000b having two independent motors.

[0245] When sizing the motors and speed controllers, several criteria were taken into consideration: (1) power and torque capability, (2) RPM range, and (3) regenerative braking. For power and torque capability, the setup 4000b should be capable of integrating a small internal combustion engine, therefore the power should be in the range of 3-8 kW (4-10 HP). For the RPM range, the setup 4000b should be capable of at least 3000 RPM in order to test high energy shifting configurations. Brushless motors provide the best performance with lowest anticipated maintenance. For regenerative braking, speed controllers with regenerative braking functionality are preferable to remove power from the system and to rapidly decelerate the motors.

[0246] Based on the desired specifications, the motors used in the setup 4000b were a Mars M907 EM paired with a Kelly Controller KBL72301X rated for 72 V with peak current of 300 A and regenerative functionality. FIG. 24D shows an image of the electric motor used in the setup 4000b. This motor may be operated up to 5000 RPM and may be assembled as a kit with a throttle and preprogrammed controller by Kelly Controls. The setup 4000b includes six 12-Volt batteries wired in series for a total of 72 volts DC to supply electrical power to the motors.

[0247] The transmission portion of the setup 4000b includes the housing, shafts, and gears. For this test setup, a manual 1984 Ford F-150 transmission was used. The Ford F-150 transmission was readily available in the market and reduces the need to build a new housing enclosure. Modeling the synchronizer and comparing analytical results to our experimental results is expected to allow predictions of synchronizer behavior based on geometry.

[0248] The Ford transmission includes several desirable features: (1) the pre-existing housing mitigates the risk of misalignment between the two transmission shafts, allowing the accuracy of the alignment to meet production specifications, (2) the housing provides an enclosure for lubrication, which prevents overheating and reduces frictional effects during synchronization, (3) the housing provides a safe enclosure to prevent pinching and eliminates the possibility of broken or loose components from injuring an operator, and (4) the housing has a shifting mechanism already in place, thus the actuator 2700 man be attached to this linkage and used to engage the synchronizer 2500.

[0249] The actuator 2700 is used to physically move the sleeve 2520 in the transmission to engage and disengage the synchronizer 2500. The actuator 2700 may include a shift fork and a pneumatic system. The shift fork may be mounted on a shaft that is parallel to the power transmitting shafts and may be moved linearly to move the sleeve 2520. Pneumatics were chosen due to the following benefits: (1) high pressure air is readily available and more manageable than high pressure hydraulics used for actuation in vehicles, (2) high power densities may be readily achieved, (3) low cost compared to similar power electromagnetic or ball screw actuators, and (4) force may be readily modulated between different experiments.

[0250] The actuator 2700 may include a pneumatic cylinder with integrated position feedback, two solenoid valves, two pressure accumulators, and a pressure regulator. FIG. 24E shows a schematic of the actuator 2700 used in the setup 4000b. The pressure of the actuator 2700 may be varied from 0 to 100 psi to control the speed and force with which the actuator 2700 moves. The output of the piston is coupled to the shift fork shaft controlling the movement of the sleeve 2520 in the transmission.

[0251] Two pneumatic actuators were selected for the setup 4000b. A Bimba PFC-092-bf with a 1.0625" bore is used for dog tooth shifting and a PFC-502-BF with a 2.5" bore for large inertia shifting. FIG. 24F shows an image of these pneumatic actuators. The 2.5" bore piston can generate up to about 2180 N of force. A 500 RPM differential should have about 1500 N of force to engage the synchronizer 2500, which may be obtained with an air pressure of 70 psi (482 kPa).

[0252] Generally, a large piston may be used for high force tests and a smaller piston may be used for high speed tests. However, for initial tests, a smaller piston was selected for dog tooth shifting since the sleeve 2520 may be moved using a lower force is required and a faster velocity was desired. Compared to a larger piston, a smaller piston gives lower force at a given air pressure due to the smaller piston area and may move faster due to the smaller volume of air to fill the cylinder. For a given displacement, the volume of the smaller piston is 18% the volume of the large piston, thus the smaller piston may move substantially faster at a set airflow rate.

[0253] The alignment of the linear actuator, shift fork, and load bearing shaft was another aspect considered in the setup 4000b. Failure to align these components may lead to the shift fork binding on the gears or sleeve rather than pushing the sleeve over the synchronizer to engage with the intended gear. To address this issue, many of the existing features in the housing were utilized including the original shaft supports.

[0254] The design of the actuator 2700 was validated by performing tests with each actuator to characterize the response time. Fast actuation is desirable for both large inertia shifting and for dog tooth shifting since a successful gear shift depends on engaging the respective sets of dog teeth faster than the relative speed difference between the dog teeth. A series of tests were performed to validate the use of a pneumatic actuator for the setup 4000b. During the test, the piston was set to a fully retracted position and the solenoid valve was triggered to open. Air then flowed into the piston and the motion of the piston is recorded.

[0255] FIG. 25A shows the position of the small piston as a function of time from 20 to 60 psi. FIG. 25B shows the position of the small piston as a function of time for several tests performed at 60 psi. As the pressure increases, the actuation time for the piston decreases. Most of the delay in actuation may be attributed to triggering of the solenoid and pressurizing the inside of the cylinder to initiate motion.

[0256] For the setup 4000b, the sleeve 2520 of the synchronizer 2500 engaged the input gear 2300 using respective sets of dog teeth. The angle of tapered tip portion of each dog tooth was 54 degrees. The outer diameter of the dog teeth on the gear was 82 mm. Based on this geometry, a speed difference of 336 RPM between the sleeve 2520 and the input gear 2300 results in a dog tooth velocity of 1.45 mm/ms. This value represents a limiting case with an actuation velocity of 2 mm/ms measured from the pneumatic actuator at 60 psi.

[0257] The sensor 2620 in the position encoder 2600 comprised a variable reluctance sensor, which is similar to the wheel speed sensors used in a vehicle's anti-lock braking system (ABS). When a dog tooth passes by the sensor, a voltage spike is produced. This signal may be converted to a square wave indicating the position of the dog tooth relative to the sensor 2620, as shown in FIG. 26. FIG. 26 shows a reluctance wheel and the dog teeth passing by a fixed position. As a dog tooth passes the sensor 2620, a voltage peak is recorded (i.e., the sine wave).

[0258] The sensor 2620 may be calibrated such that the peak in the sine wave is aligned to the wave signal representing the physical dog tooth position in time. The location of the sensor 2620 relative to the dog teeth of the input gear 2300 and the sleeve 2520 may be calibrated using a high speed camera. The high speed camera may be triggered to take an image at a desired time and the actual position of the dog teeth may then be compared to the reluctance sensor signal. This calibration is used only to determine the absolute position between the sleeve 2520 and the input gear 2300, not the relative position.

[0259] During a gear shift, the setup 4000b also accounts for delays within the system. For a speed difference of 50 RPM, the period of the signal is 33 ms peak to peak for a synchronizer with 36 teeth. This time scale is on the same order as the 20 ms delay and 5 ms actuation time for the pneumatic system. Although reductions in the delay is desirable, these delays may also be accounted for by the control systems during operation.

[0260] The controller 2800 in the setup 4000b is a National Instruments myRIO. LabVIEW was used for control and data logging. The recorded signals include the piston position, reluctance sensor signal, motor velocity from the speed controllers, motor current, and battery voltage. The piston position and dog tooth position are used for the controls logic. Outputs from the myRIO include triggering of the pneumatic solenoids, speed control of the motors, and enabling the motor drivers to turn on a high power relay.

[0261] An exemplary shift event may begin with each motor being enabled and spinning at a commanded speed (e.g., motor 1 slowly and motor 2 at a higher fixed speed). The reluctance sensor data is used to determine when the dog teeth are aligned. Motor 1 may be commanded to accelerate to a speed higher than motor 2. When the motor speeds are within a set threshold and the dog teeth are aligned, the piston is actuated and motor 1 is set to torque mode.

[0262] Exemplary Drivetrains Using Position and Speed Matching

[0263] The position and speed matching methodology described herein may be incorporated into various types of transmissions and/or drivetrains. This may be accomplished, in part, by the installation of the position encoder 2600 to monitor the respective shafts and gears in the transmission. As described above, a single position encoder 2600 may monitor the position and speed of a single shaft and/or gear or multiple shafts and/or gears.

[0264] For example, FIG. 27A shows an exemplary transmission 5000a coupled to an internal combustion engine 5010 and a clutch 5030. As shown, the internal combustion engine 5010 may deliver torque to an input shaft 2100 that is coupled to an output shaft 2200 via multiple input gears 2300 with varying gear ratios (including a reverse gear). In this example, the synchronizers 2500 are disposed on the output shaft 2200. Position encoders (not shown) may be integrated into the transmission 5000a to monitor the position and speed of each set of dog teeth on the respective sleeves 2520 of each synchronizer 2500 and/or input gear 2300.

[0265] In some implementations, the engine 5010 may be used adjust the speed of the input shaft 2100 relative to the output shaft 2200 during the synchronization phase of the gear shift. For example, the clutch 5030 may be engaged and the engine 5010 may accelerate the input shaft 2100 by outputting a higher torque or decelerate the input shaft 2100 by providing an additional inertial load on the input shaft 2100. In some implementations, the synchronizers 2500 may include friction elements. In some implementations, the transmission 5000a may include a damper (not shown) to reduce the speed of the input shaft 2100 or the output shaft 2200.

[0266] FIG. 27B shows another exemplary transmission 5000b coupled to an engine 5010 for propulsion, a first electric motor 5020a to adjust the speed of the engine 5010, and a second electric motor 5020b for propulsion and/or speed matching. The engine 5010 and the second electric motor 5020b transfer torque to an output shaft 2200 via respective input shafts 2100a and 2100b, respectively. The second electric motor 5020b may be coupled to the output shaft 2200 via a single input gear 2300. The transmission 5000b does not include a clutch; thus, the engine 5010 may be configured to directly engage the synchronizer 2500 on the input shaft 2100a. The first electric motor 5020a may be used to adjust the speed of the engine 5010 prior to engagement (e.g., the engine 5010 may be at idle).

[0267] For high performance hybrid applications, automated manual transmissions (AMT) are typically used. The design of these transmissions is quite different from traditional automatic transmissions found on previous vehicles with little crossover between the two transmission types. A dual clutch transmission (DCT), which is a type of AMT, allows for nearly seamless torque delivery during a gear shift. A typical DCT operates by having two clutches coupled to an engine with a common input, but two separate outputs. For example, a first shaft may be concentrically disposed inside a cavity of a second shaft.

[0268] When the vehicle is being driven in one gear (e.g., first gear) disposed on a first input shaft, another gear (e.g., second gear) located on a second input shaft may already be selected and engaged to a synchronizer on the second input shaft. A gear shift from first gear to second gear comprises disengaging the clutch to the first input shaft and engaging the clutch to the second input shaft. In this manner, the speed matching portion of the gear shift is performed beforehand, thus reducing the shift time when the gear shift is initiated. Once the gear shift is complete, a synchronizer on the first input shaft may be shifted to neutral and preselect a third gear.

[0269] FIG. 27C shows an exemplary transmission 5000c that operates in a substantially similar manner to a DCT with the difference being the transmission 5000c does not include clutches. As shown, the transmission 5000c may be coupled to an electric motor 5020 for propulsion and/or speed matching and an engine 5010 for propulsion. The electric motor 5020 and the engine 5010 may be coupled to none, one or both input shafts 2100a and 2100b based on the dog clutches 5040a and 5040b. The synchronizers 2500b disposed on the input shafts 2100a and 2100b may be used to preselect a particular input gear 2300. For example, the synchronizers 2500 may be arranged such that input shafts 2100a and 2100b continuously receive torque from the engine 5010 and/or the electric motor 5020. The synchronizers 2500 may be arranged such that only one of the input shafts 2100a and 2100b are transfer torque to the output shaft 2200.

[0270] Vehicle transmissions that include an electric motor for propulsion (e.g., a hybrid electric vehicle, an electric vehicle) are typically unable to achieve a large power band (e.g., a high power output across a large array of speeds) and high efficiency simultaneously. This may be attributed, in part, to the torque from the electric motor typically being transferred to the output shaft using only a single gear ratio. However, in several applications (e.g., performance vehicles), it may be beneficial to tailor the power band and efficiency of the electric motor by incorporating multiple input gears at various gear ratios into the transmission. For example, one gear ratio may couple the electric motor to the engine output shaft and a second gear ratio may couple the electric motor to the output shaft.

[0271] FIG. 27D shows an exemplary transmission 5000d coupled to a single clutch, 4-speed transmission coupled to an engine 5010 hybridized with a two-speed electric motor 5020. As shown, the engine 5010 and the electric motor 5020 may each have an input shaft 2100 (i.e., input shafts 2100a and 2100b) that couples to a single output shaft 2200. The engine 5010 may couple to its respective input shaft 2100a via a clutch 5030. The synchronizers 2500a and 2500b are used to couple the engine 5010 to the output shaft 2200 and function in a substantially similar manner to a traditional transmission. The dog clutch 5040 couples the electric motor 5020 to the output shaft 2200b. Compared to the synchronizers 2500a and 2500b, the dog clutch 5040 may be subjected to a higher motor inertia particularly since the electric motor 5020 is not coupled to the input shaft 2100b using a clutch.

[0272] The input shaft 2100b may include a first input gear 2300a and a second input gear 2300b corresponding to a first and second gear ratio. The first gear ratio may be used for high torque delivery at low speeds to improve vehicle launch and/or acceleration and to help increase vehicle performance at the friction limit of the tires. The second gear ratio may be used for power assistance at higher vehicle speeds (e.g., up to the vehicle's full speed). Thus, the second gear ratio may allow the electric motor to remain engaged at high speed. The second gear ratio may additionally be used for energy recovery under braking since the electric motor is able to operate closer to its peak power level.

[0273] A position encoder (not shown) may be integrated to monitor the position and speed of the input shaft 2100b and the first and second input gears 2300a and 2300b. In some implementations, a position encoder may also be installed for the input shaft 2100a and its respective input gears.

[0274] FIG. 27E shows another exemplary transmission 5000e coupled to an engine 5010 for propulsion, a first electric motor 5020a for speed matching, and a second electric motor 5020b for propulsion. The transmission 5000e operates in a similar manner to the transmission 5000b of FIG. 27B. In this case, the second electric motor 5020 has a first input gear 2300a and a second input gear 2300b, which may be selected using a synchronizer 2500.

[0275] **Conclusion**

[0276] All parameters, dimensions, materials, and configurations described herein are meant to be exemplary and the actual parameters, dimensions, materials, and/or configurations will depend upon the specific application or applications for which the inventive teachings is/are used. It is to be understood that the foregoing embodiments are presented primarily by way of example and that, within the scope of the appended claims and equivalents thereto, inventive embodiments may be practiced otherwise than as specifically described and claimed. Inventive embodiments of the present disclosure are directed to each individual feature, system, article, material, kit, and/or method described herein.

[0277] In addition, any combination of two or more such features, systems, articles, materials, kits, and/or methods, if such features, systems, articles, materials, kits, and/or methods are not mutually inconsistent, is included within the inventive scope of the present disclosure. Other substitutions, modifications, changes, and omissions may be made in the design, operating conditions and arrangement of respective elements of the exemplary implementations without departing from the scope of the present disclosure. The use of a numerical range does not preclude equivalents that fall outside the range that fulfill the same function, in the same way, to produce the same result.

[0278] The above-described embodiments can be implemented in multiple ways. For example, embodiments may be implemented using hardware, software or a combination thereof. When implemented in software, the software code can be executed on a suitable processor or collection of processors, whether provided in a single computer or distributed among multiple computers.

[0279] Further, it should be appreciated that a computer may be embodied in any of a number of forms, such as a rack-mounted computer, a desktop computer, a laptop computer, or a tablet computer. Additionally, a computer may be embedded in a device not generally regarded as a computer but with suitable processing capabilities, including a Personal Digital Assistant (PDA), a smart phone or any other suitable portable or fixed electronic device.

[0280] Also, a computer may have one or more input and output devices. These devices can be used, among other things, to present a user interface. Examples of output devices that can be used to provide a user interface include printers or display screens for visual presentation of output and speakers or other sound generating devices for audible presentation of output. Examples of input devices that can be used for a user interface include keyboards, and pointing devices, such as mice, touch pads, and digitizing tablets. As another example, a computer may receive input information through speech recognition or in other audible format.

[0281] Such computers may be interconnected by one or more networks in a suitable form, including a local area network or a wide area network, such as an enterprise network, an intelligent network (IN) or the Internet. Such networks may be based on a suitable technology, may operate according to a suitable protocol, and may include wireless networks, wired networks or fiber optic networks.

[0282] The various methods or processes outlined herein may be coded as software that is executable on one or more processors that employ any one of a variety of operating systems or platforms. Additionally, such software may be written using any of a number of suitable programming languages and/or programming or scripting tools, and also may be compiled as executable machine language code or intermediate code that is executed on a framework or virtual machine. Some implementations may specifically employ one or more of a particular operating system or platform and a particular programming language and/or scripting tool to facilitate execution.

[0283] Also, various inventive concepts may be embodied as one or more methods, of which at least one example has been provided. The acts performed as part of the method may in some instances be ordered in different ways. Accordingly, in some inventive implementations, respective acts of a given method may be performed in an order different than specifically illustrated, which may include performing some acts simultaneously (even if such acts are shown as sequential acts in illustrative embodiments).

[0284] All publications, patent applications, patents, and other references mentioned herein are incorporated by reference in their entirety.

[0285] All definitions, as defined and used herein, should be understood to control over dictionary definitions, definitions in documents incorporated by reference, and/or ordinary meanings of the defined terms.

[0286] The indefinite articles “a” and “an,” as used herein in the specification and in the claims, unless clearly indicated to the contrary, should be understood to mean “at least one.”

[0287] The phrase “and/or,” as used herein in the specification and in the claims, should be understood to mean “either or both” of the elements so conjoined, *i.e.*, elements that are conjunctively present in some cases and disjunctively present in other cases. Multiple elements listed with “and/or” should be construed in the same fashion, *i.e.*, “one or more” of the elements so conjoined. Other elements may optionally be present other than the elements specifically identified by the “and/or” clause, whether related or unrelated to those elements specifically identified. Thus, as a non-limiting example, a reference to “A and/or B”, when used in conjunction with open-ended language such as “comprising” can refer, in one embodiment, to A only (optionally including elements other than B); in another embodiment, to B only (optionally including elements other than A); in yet another embodiment, to both A and B (optionally including other elements); etc.

[0288] As used herein in the specification and in the claims, “or” should be understood to have the same meaning as “and/or” as defined above. For example, when separating items in a list, “or” or “and/or” shall be interpreted as being inclusive, *i.e.*, the inclusion of at least one, but also including more than one, of a number or list of elements, and, optionally, additional unlisted items. Only terms clearly indicated to the contrary, such as “only one of” or “exactly one of,” or, when used in the claims, “consisting of,” will refer to the inclusion of exactly one element of a number or list of elements. In general, the term “or” as used herein shall only be interpreted as indicating exclusive alternatives (*i.e.* “one or the other but not both”) when preceded by terms of exclusivity, such as “either,” “one of,” “only one of,” or “exactly one of.” “Consisting essentially of,” when used in the claims, shall have its ordinary meaning as used in the field of patent law.

[0289] As used herein in the specification and in the claims, the phrase “at least one,” in reference to a list of one or more elements, should be understood to mean at least one element selected from any one or more of the elements in the list of elements, but not necessarily including at least one of each and every element specifically listed within the list of elements and not excluding any combinations of elements in the list of elements. This definition also allows that elements may optionally be present other than the elements specifically identified within the list of elements to which the phrase “at least one” refers, whether related or unrelated to those elements specifically identified. Thus, as a non-limiting example, “at least one of A and B” (or, equivalently, “at least one of A or B,” or, equivalently “at least one of A and/or B”) can refer, in one embodiment, to at least one, optionally including more than one, A, with no B present (and optionally including elements other than B); in another embodiment, to at least one, optionally including more than one, B, with no A present (and optionally including elements other than A); in yet another embodiment, to at least one, optionally including more than one, A, and at least one, optionally including more than one, B (and optionally including other elements); etc.

[0290] In the claims, as well as in the specification above, all transitional phrases such as "comprising," "including," "carrying," "having," "containing," "involving," "holding," "composed of," and the like are to be understood to be open-ended, *i.e.*, to mean including but not limited to. Only the transitional phrases "consisting of" and "consisting essentially of" shall be closed or semi-closed transitional phrases, respectively, as set forth in the United States Patent Office Manual of Patent Examining Procedures, Section 2111.03.

CLAIMS

1. A mechanical power transmission, comprising:

an input shaft that rotates about a first rotation axis;

a rotating element that rotates about the first rotation axis relative to the input shaft;

a first position encoder, comprising:

a first index ring that rotates with the input shaft; and

a first sensor, disposed proximate to the first index ring, to measure a first position of the first index ring; and

a second position encoder, comprising:

a second index ring that rotates with the rotating element; and

a second sensor, disposed proximate to the second index ring, to measure a second position of the second index ring.

2. The mechanical power transmission of claim 1, wherein the rotating element is at least one of a shaft or a gear.

3. The mechanical power transmission of claim 1, wherein the first and second positions each correspond to an angular position measured at increments less than 2π radians.

4. The mechanical power transmission of claim 1, further comprising:

a coupling mechanism disposed between the input shaft and the rotating element; and

an actuator, coupled to the coupling mechanism, to at least one of engage or disengage the rotating element to the input shaft by actuating the coupling mechanism.

5. The mechanical power transmission of claim 4, wherein the coupling mechanism does not include a friction element.

6. The mechanical power transmission of claim 4, further comprising:

a controller, coupled to the first and second position encoders and the actuator, that (1) receives the first and second positions as input, (2) determines at least one of a relative position or a relative speed between the input shaft and the rotating element based on the first and second positions, and (3) outputs an actuator signal that activates the actuator so as to engage the rotating element to the input shaft when the at least one of a relative position or a relative speed satisfies a criterion.

7. The mechanical power transmission of claim 6, wherein the coupling mechanism comprises:

a sleeve, coupled to the actuator, that (1) rotates with the input shaft and (2) is slidably adjustable along the input shaft via the actuator;

a first plurality of dog teeth disposed on the sleeve; and

a second plurality of dog teeth, disposed on the rotating element, that is engageable with the first plurality of dog teeth,

the sleeve causing the rotating element to rotate with the input shaft when the sleeve is moved such that the first and second pluralities of dog teeth are engaged.

8. The mechanical power transmission of claim 7, wherein:

each dog tooth in the first and second pluralities of dog teeth comprises a base portion and a tip portion disposed on the base portion; and

the criterion comprises engaging the first and second pluralities of dog teeth without the respective tip portions of the first and second pluralities of dog teeth colliding.

9. The mechanical power transmission of claim 7, wherein:

each dog tooth in the first and second pluralities of dog teeth comprises a base portion, the base portion having a front and a side; and

the criterion comprises engaging the first and second pluralities of dog teeth such that only the side of the base portion of the first and second pluralities of dog teeth contact.

10. The mechanical power transmission of claim 7, wherein the criterion comprises engaging the first and second pluralities of dog teeth such that clashing between the first and second pluralities of dog teeth is substantially mitigated.

11. The mechanical power transmission of claim 7, wherein:

the first index ring includes a plurality of magnetic pole pairs disposed along the periphery of the first index ring, each magnetic pole pair corresponding to each dog tooth in the first plurality of dog teeth; and

the first sensor comprises a Hall effect sensor to a measure a waveform for each magnetic pole pair.

12. The mechanical power transmission of claim 11, wherein each magnetic pole pair in the plurality of magnetic pole pairs corresponds to one dog tooth in the first plurality of dog teeth.

13. The mechanical power transmission of claim 7, wherein the sleeve does not include a friction element.

14. The mechanical power transmission of claim 7, wherein each dog tooth in the first and second pluralities of dog teeth does not include a tip portion.

15. The mechanical power transmission of claim 6, wherein the input shaft receives a torque input from at least one of an internal combustion engine or an electric motor.

16. The mechanical power transmission of claim 15, wherein the torque input is used, in part, to reduce the relative speed between the input shaft and the rotating element prior to engaging the rotating element to the input shaft.

17. A method of shifting a mechanical power transmission, comprising:

measuring a first position of an input shaft;

while measuring the first position, measuring a second position of a rotating element that is rotatably coupled to the input shaft;

determining a relative position and a relative speed between the input shaft and the rotating element using the first and second positions; and

in response to the relative position and the relative speed satisfying a criterion, actuating a coupling mechanism so as to engage the input shaft to the rotating element thereby causing the rotating element to rotate with the input shaft.

18. The method of claim 17, wherein the rotating element is at least one of a shaft or a rotating element.

19. The method of claim 17, wherein actuating a coupling mechanism so as to engage the input shaft to the rotating element comprises:

sliding a sleeve along the input shaft via an actuator so as to engage a first plurality of dog teeth disposed on the sleeve to a second plurality of dog teeth disposed on the rotating element.

20. The method of claim 19, wherein:

each dog tooth in the first and second pluralities of dog teeth comprises a base portion and a tip portion disposed on the base portion; and

the criterion comprises engaging the first and second pluralities of dog teeth without the respective tip portions of the first and second pluralities of dog teeth colliding.

21. The method of claim 19, wherein:

each dog tooth in the first and second pluralities of dog teeth comprises a base portion, the base portion having a front and a side; and

the criterion comprises engaging the first and second pluralities of dog teeth such that only the side of the base portion of the first and second pluralities of dog teeth contact.

22. The method of claim 19, wherein the criterion comprises engaging the first and second pluralities of dog teeth such that clashing between the first and second pluralities of dog teeth is substantially mitigated.

23. The method of claim 17, wherein the coupling mechanism does not include a friction element.

24. The method of claim 17, further comprising:

while measuring the first and second positions, adjusting the speed of the input shaft by applying a torque from a power source coupled to the input shaft so as to decrease the relative speed between the input shaft and the rotating element.

25. The method of claim 24, wherein adjusting the speed of the input shaft does not include use of a friction element.

26. The method of claim 24, wherein the power source is at least one of an internal combustion engine or an electric motor.

27. The method of claim 24, wherein adjusting the speed of the input shaft comprises:

adjusting the torque from the power source to compensate a speed irregularity in at least one of the input shaft or the rotating element.

28. The method of claim 17, wherein measuring the first position comprises:

detecting an index pulse from an index ring that rotates with the input shaft;

in response to detecting the index pulse, measuring a waveform of a magnetic pole pair using a Hall effect sensor, the magnetic pole pair corresponding to a dog tooth in the first plurality of dog teeth.

29. A mechanical power transmission, comprising:

an input shaft;

a gear, rotatably coupled to the input shaft, having a first plurality of dog teeth;

a synchronizer, coupled to the input shaft, comprising:

a sleeve that rotates with the input shaft, the sleeve having a second plurality of dog teeth disposed at a first end of the sleeve;

an actuator, coupled to the sleeve, to slide the sleeve along the input shaft to at least one of engage or disengage the second plurality of dog teeth to the first plurality of dog teeth, the sleeve causing the gear to rotate with the input shaft when the first and second pluralities of dog teeth are engaged;

a first position encoder comprising:

a first index ring that rotates with the input shaft; and

a first sensor, disposed proximate to the first index ring, to measure a first position of the first plurality of dog teeth;

a second position encoder comprising:

a second index ring that rotates with the sleeve; and

a second sensor, disposed proximate to the second index ring, to measure a second position of the second plurality of dog teeth; and

a controller, coupled to the first and second position encoders and the actuator, that (1) receives the first and second positions as input, (2) determines at least one of a relative position or a relative speed between the input shaft and the gear based on the first and second positions, and (3) outputs an actuator signal that activates the actuator so as to engage the gear to the input shaft when the at least one of a relative position or a relative speed satisfies a criterion.

30. The mechanical power transmission of claim 29, wherein the sleeve has a third plurality of dog teeth disposed at a second end of the sleeve, further comprising:

a second gear, rotatably coupled to the input shaft, having a fourth plurality of dog teeth;

a third position encoder comprising:

a third index ring that rotates with the input shaft; and

a third sensor, disposed proximate to the third index ring, to measure a third position of the third plurality of dog teeth; and

a fourth position encoder comprising:

a fourth index ring that rotates with the second gear; and

a fourth sensor, disposed proximate to the fourth index ring, to measure a fourth position of the fourth plurality of dog teeth.

31. The mechanical power transmission of claim 30, wherein the controller is further coupled to the third and fourth position encoders such that the controller (1) receives the third and fourth positions as input, (2) determines at least one of a second relative position or a second relative speed between the input shaft and the second gear based on the third and fourth positions, and (3) outputs the actuator signal to activate the actuator so as to engage the second gear to the input

shaft when the at least one of a second relative position or a second relative speed satisfies the criterion.

32. The mechanical power transmission of claim 29, wherein:

each dog tooth in the first and second pluralities of dog teeth comprises a base portion and a tip portion disposed on the base portion; and

the criterion comprises engaging the first and second pluralities of dog teeth without the respective tip portions of the first and second pluralities of dog teeth colliding.

33. The mechanical power transmission of claim 29, wherein:

each dog tooth in the first and second pluralities of dog teeth comprises a base portion, the base portion having a front and a side; and

the criterion comprises engaging the first and second pluralities of dog teeth such that only the side of the base portion of the first and second pluralities of dog teeth contact.

34. The mechanical power transmission of claim 29, wherein the criterion comprises engaging the first and second pluralities of dog teeth such that clashing between the first and second pluralities of dog teeth is substantially mitigated.

35. A mechanical power transmission system, comprising:

a first rotating element to rotate at a first speed about a first axis passing through the first rotating element;

a first position encoder coupled to the first rotating element to measure a first angular position of the first rotating element about the first axis as a function of time and the first speed of the first rotating element;

a second rotating element to rotate at a second speed about the first axis, wherein the first axis passes through the second rotating element;

a second position encoder coupled to the second rotating element to measure a second angular position of the second rotating element about the first axis as a function of time and the second speed of the second rotating element;

a coupling element to mechanically couple the first rotating element to the second rotating element to transfer mechanical power from the first rotating element to the second rotating element; and

a controller, coupled to the first position encoder, the second position encoder, and the coupling element, to actuate the coupling element to mechanically couple the first rotating element to the second rotating element based at least in part on:

the first angular position and the first speed of the first rotating element measured by the first position encoder; and

the second angular position and the second speed of the second rotating element measured by the second position encoder.

36. The mechanical power transmission system of claim 35, wherein the first rotating element is an input shaft coupled to a motor.

37. The mechanical power transmission system of claim 35, wherein the second rotating element is a gear.

38. The mechanical power transmission system of claim 35, wherein the first position encoder and the second position encoder respectively are magnetic position encoders.

39. The mechanical power transmission system of claim 35, wherein the coupling element is a frictionless synchronizer.

40. The mechanical power transmission system of claim 35, further comprising a motor, wherein:

the first rotating element is an input shaft coupled to the motor;

the second rotating element is a gear;

the first position encoder and the second position encoder respectively are magnetic position encoders; and

the coupling element is a frictionless synchronizer.

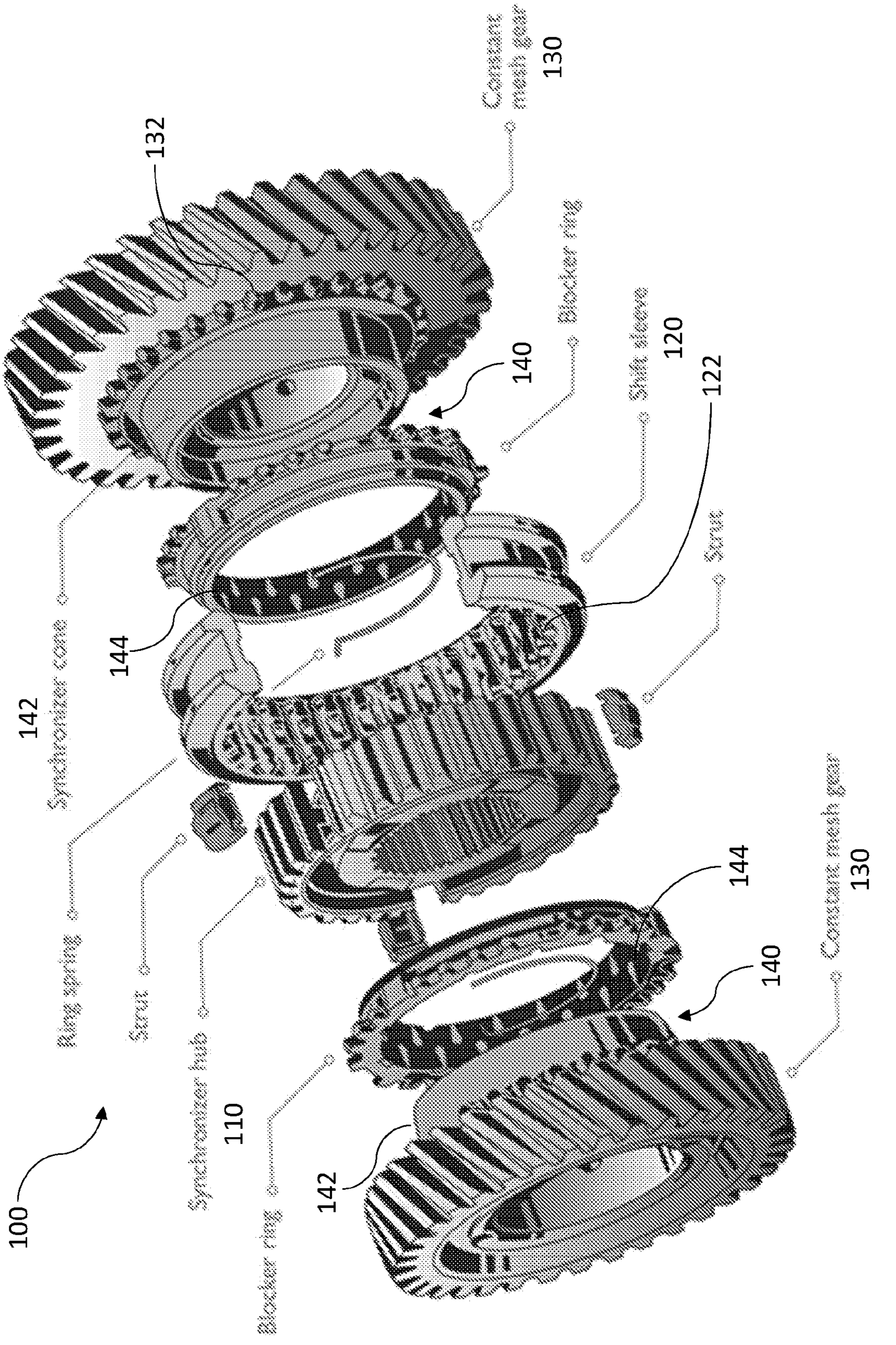

Constant mesh gear
130
132
Blocker ring
140
Shift sleeve
120
122
Strut
Synchronizer cone
142
144
Ring spring
Strut
Synchronizer hub
110
Blocker ring
100
142
144
140
Constant mesh gear
130

FIG. 1A

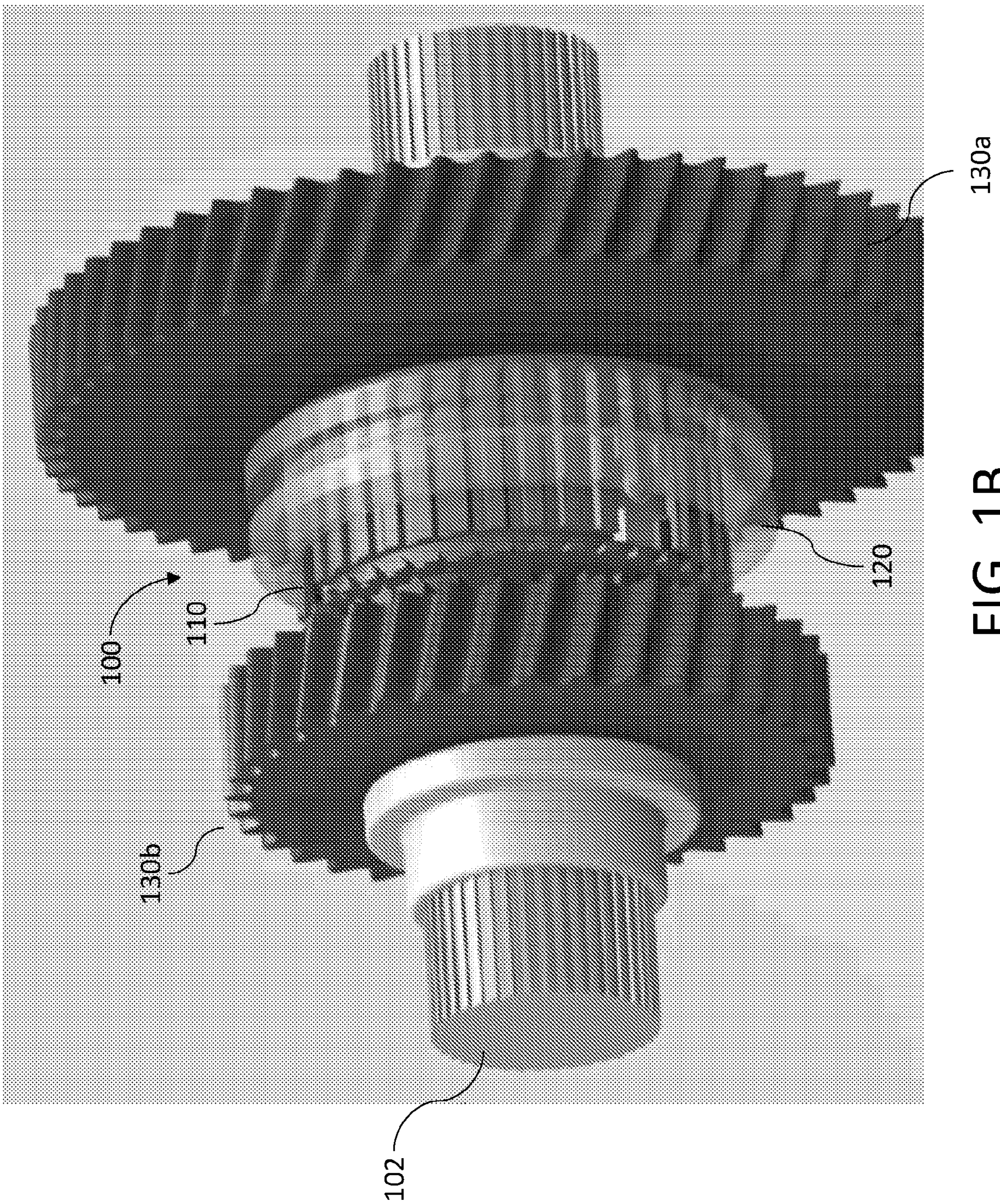


FIG. 1B

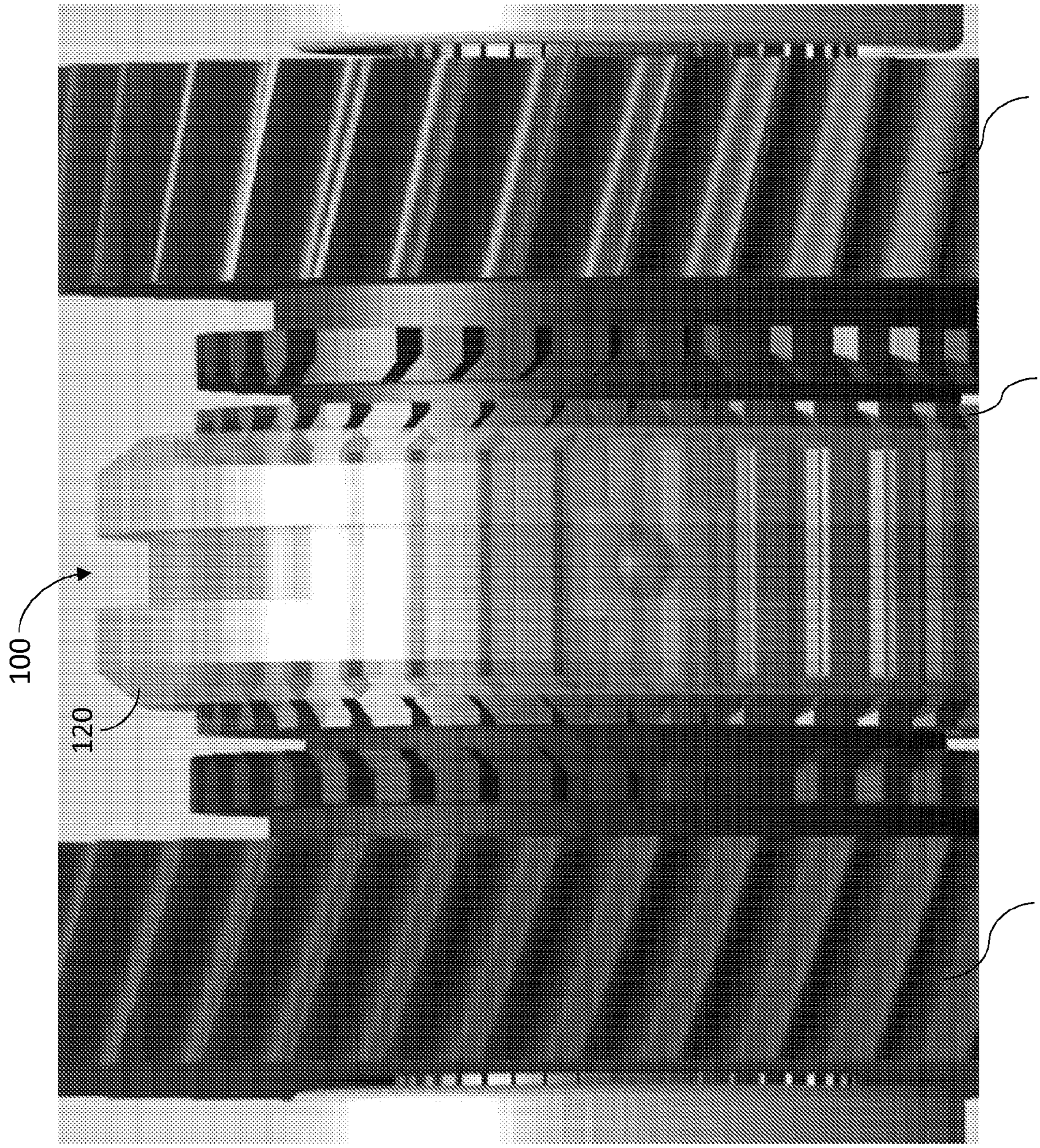


FIG. 1C

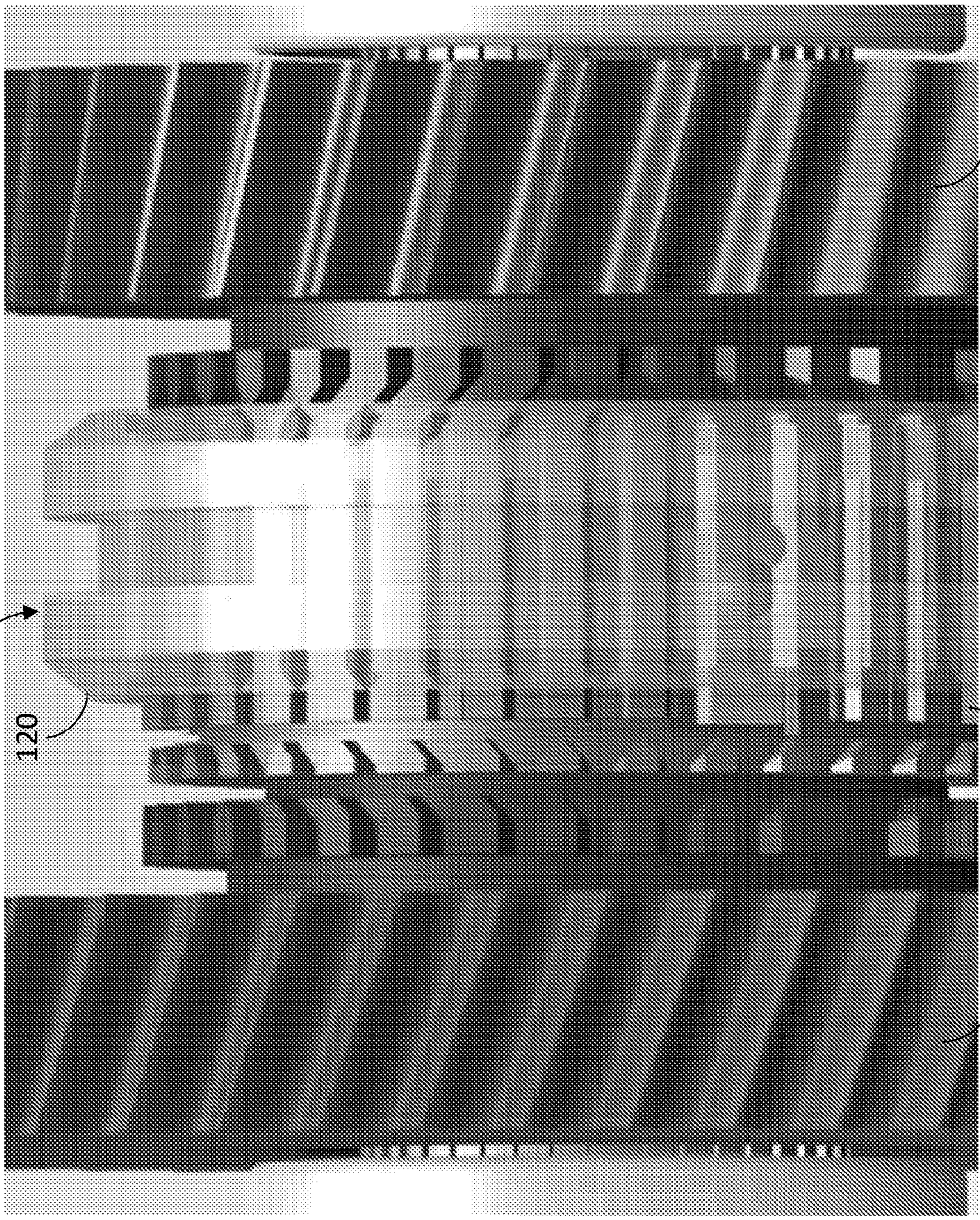


FIG. 1D

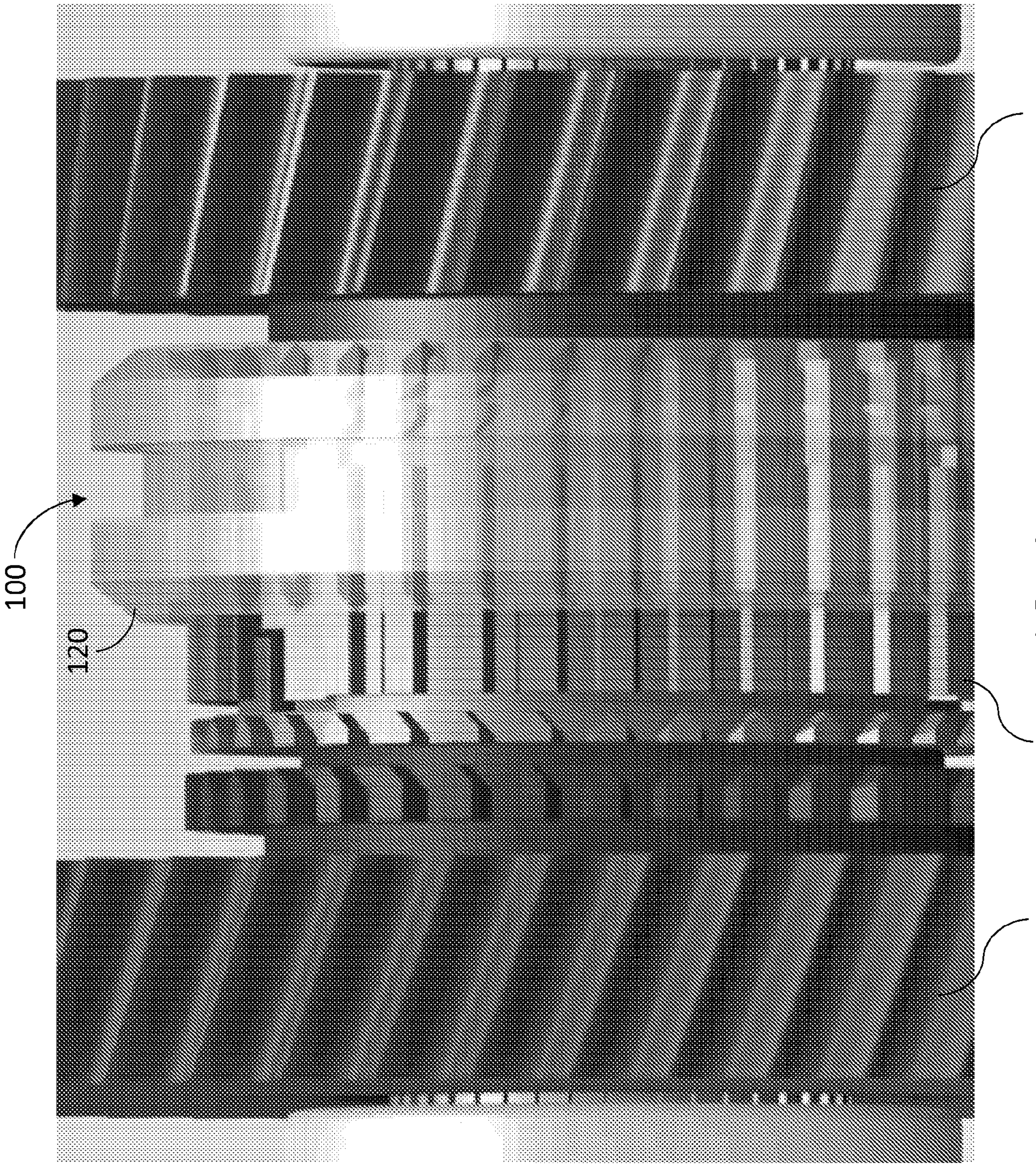


FIG. 1E

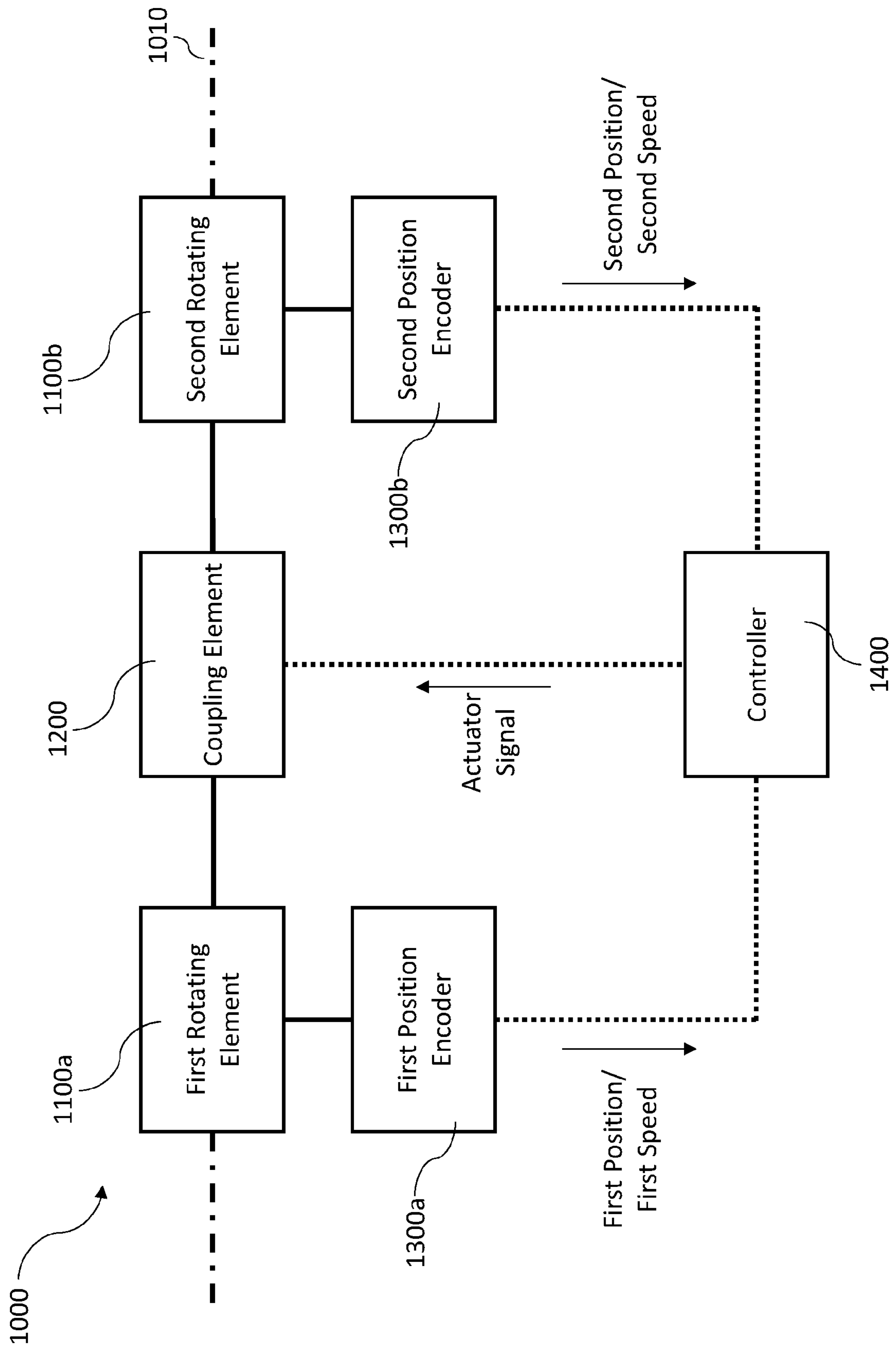

1000
1010
1100a
1100b
1200
1300a
1300b
1400
First Rotating Element
Second Rotating Element
Coupling Element
First Position Encoder
Second Position Encoder
Controller
Actuator Signal
First Position/ First Speed
Second Position/ Second Speed

FIG. 2A

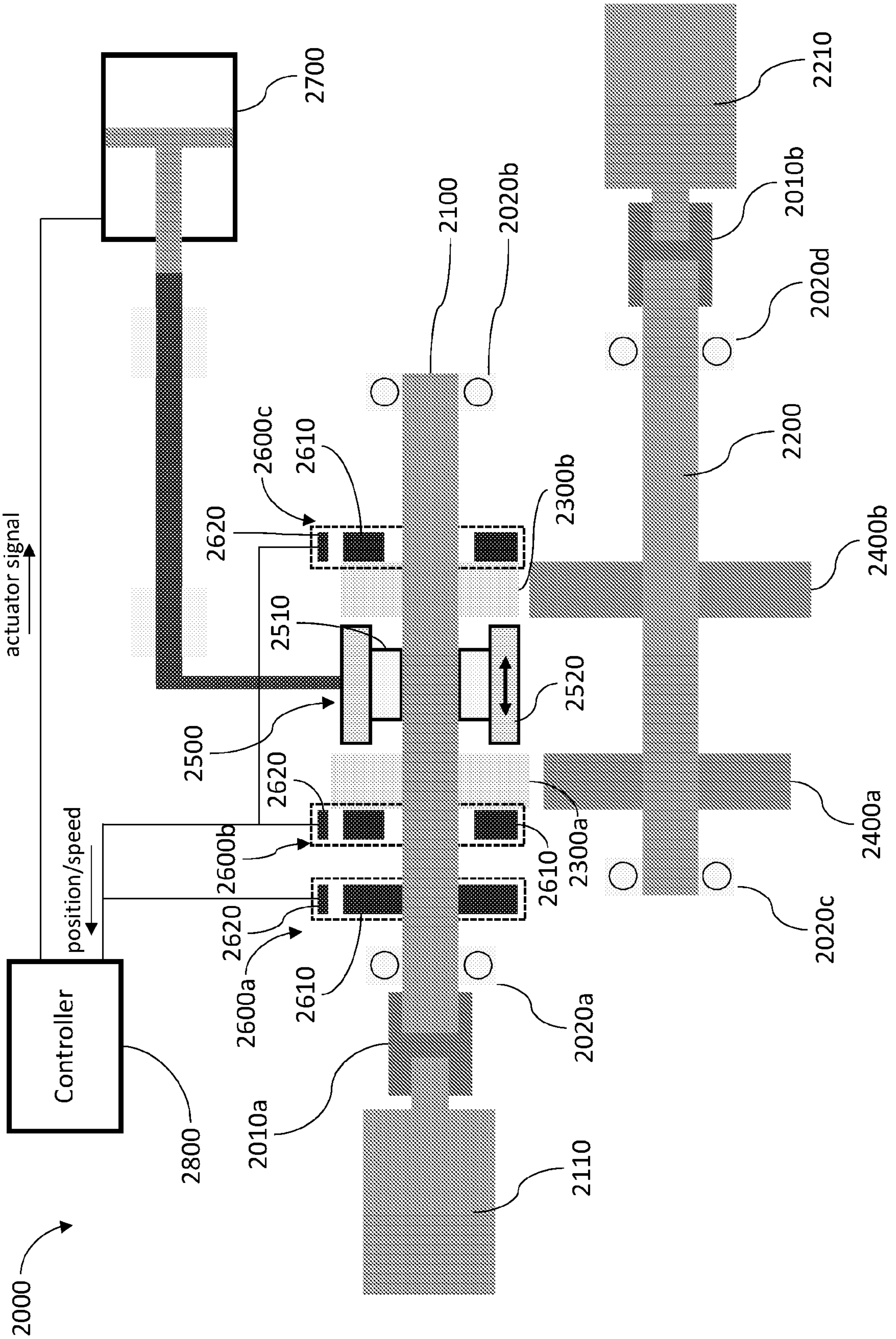

2000
Controller
2800
actuator signal
position/speed
2700
2100
2020b
2210
2010b
2020d
2200
2600c
2610
2620
2300b
2400b
2510
2500
2520
2620
2600b
2610
2300a
2400a
2620
2600a
2610
2020a
2020c
2010a
2110

FIG. 2B

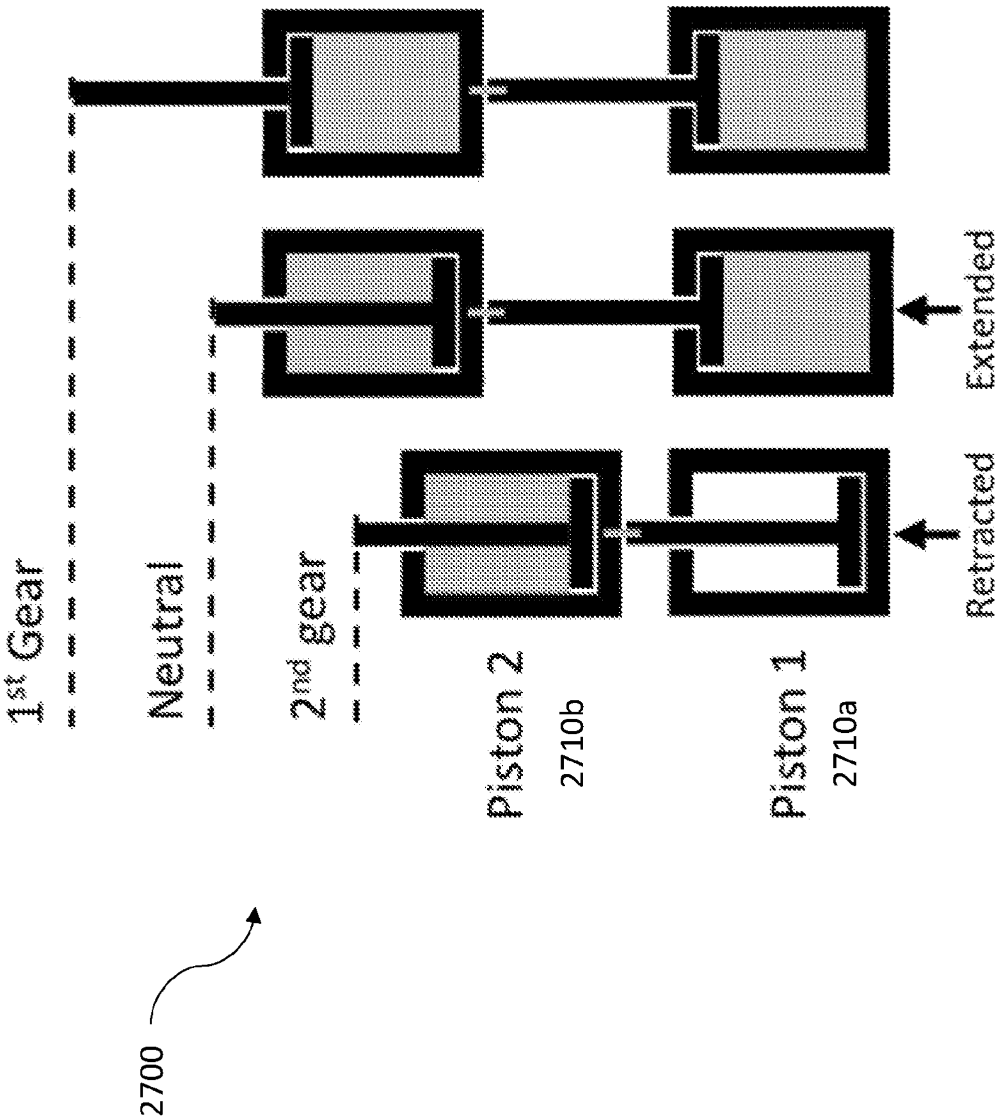

1st Gear
Neutral
2nd gear
Piston 2
2710b
Piston 1
2710a
Retracted
Extended
2700

FIG. 2C

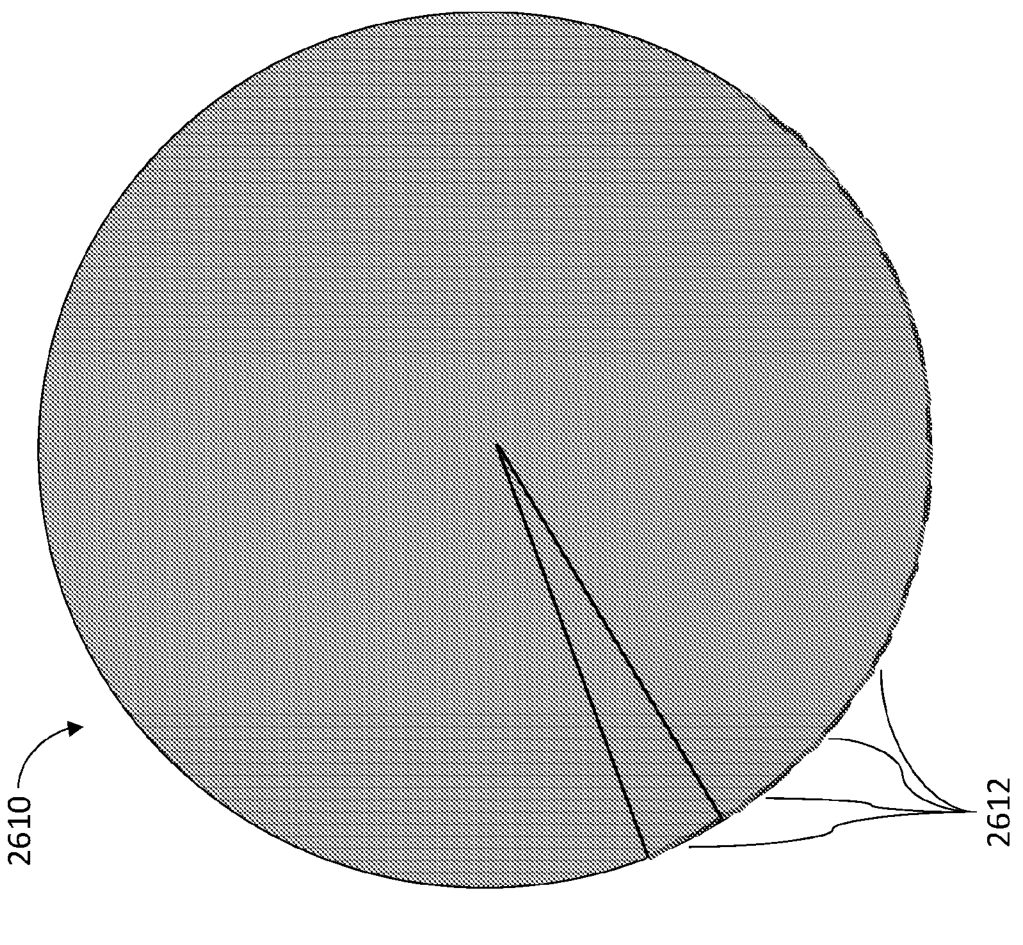


FIG. 3A

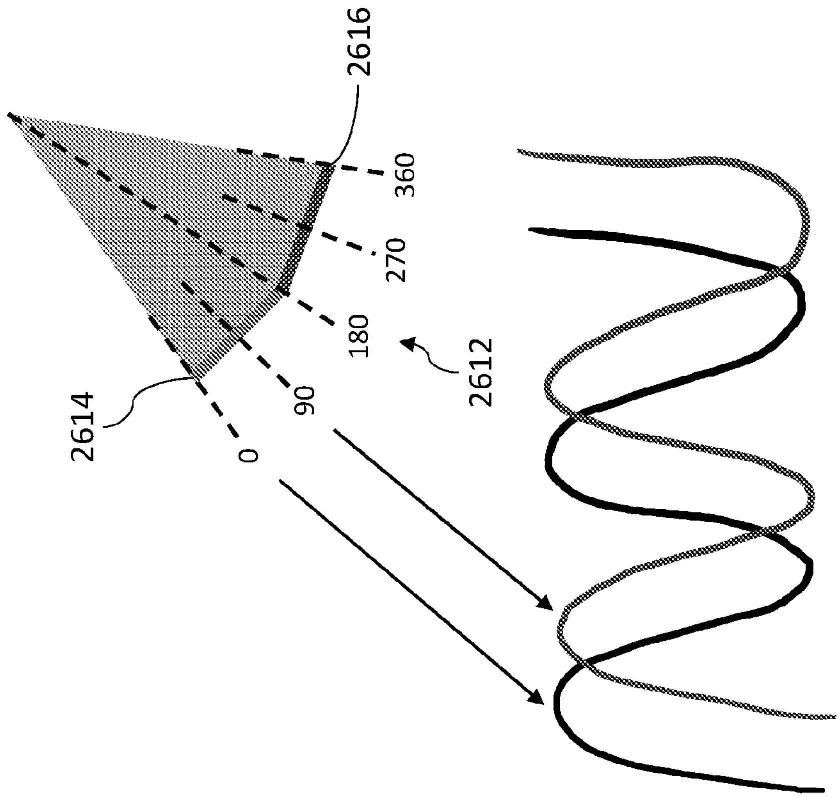


FIG. 3B

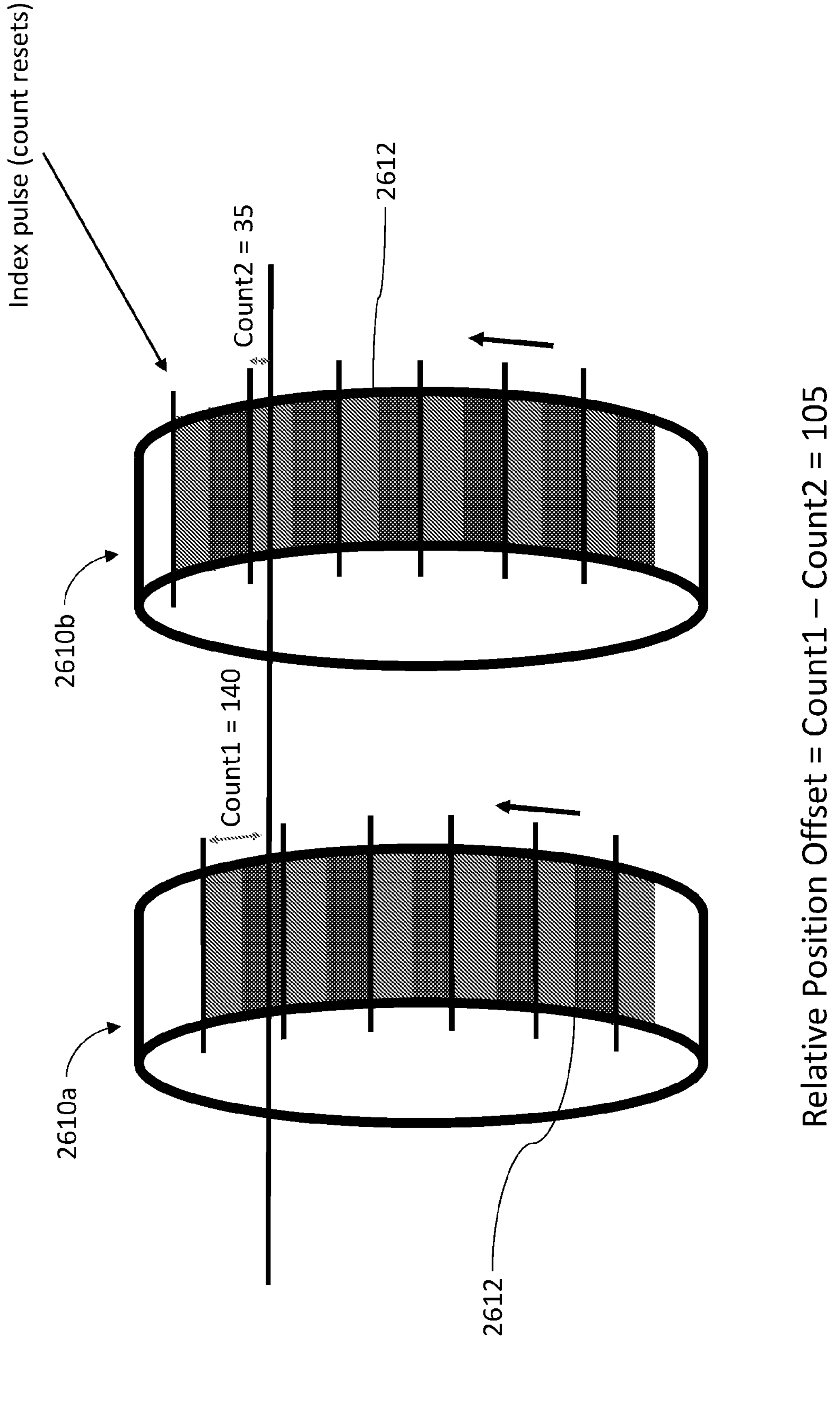

Index pulse (count resets)
Count2 = 35
2612
2610b
Count1 = 140
2610a
2612
Relative Position Offset = Count1 – Count2 = 105

FIG. 4A

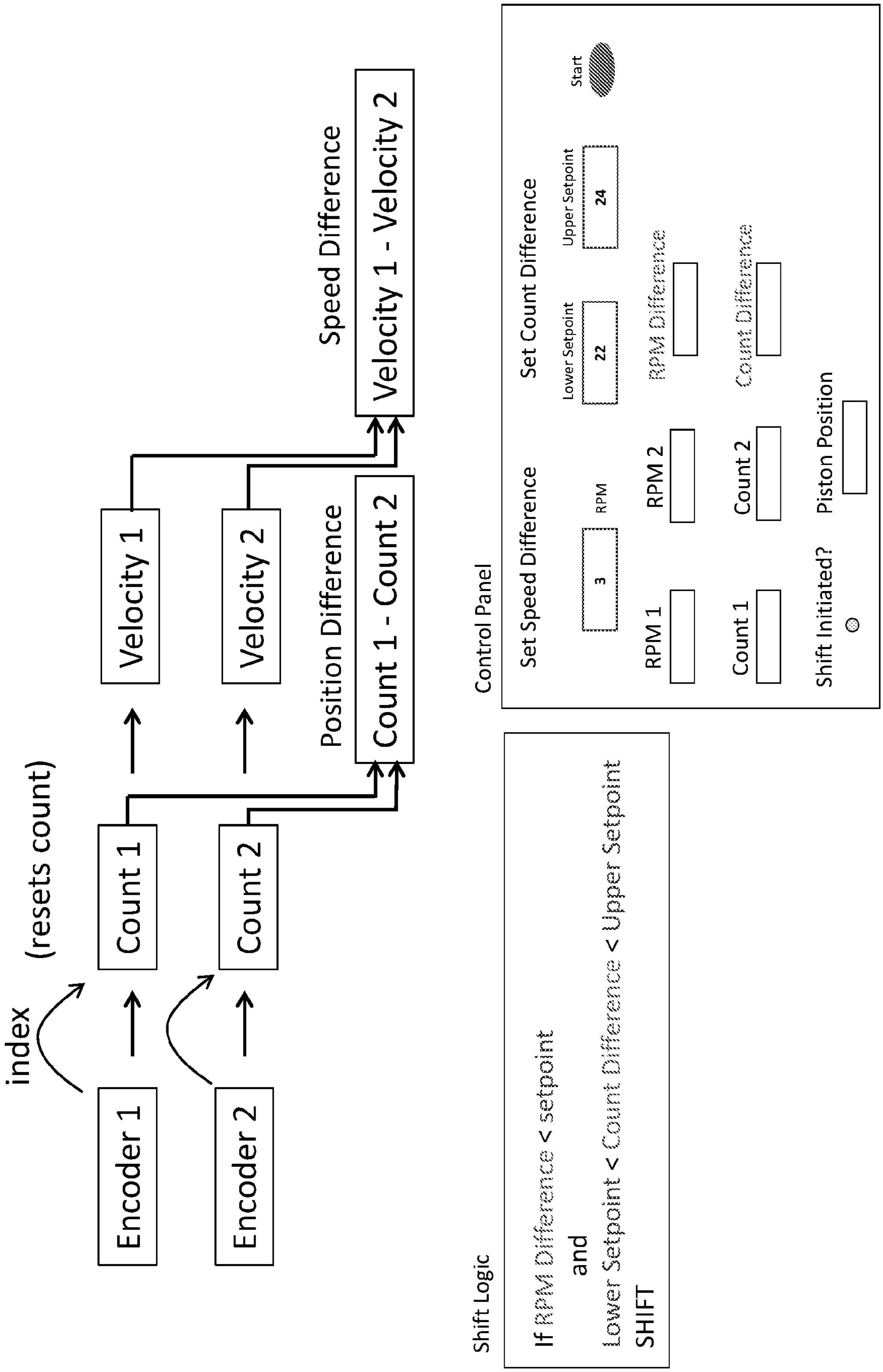

Encoder 1
Encoder 2
index
(resets count)
Count 1
Count 2
Velocity 1
Velocity 2
Position Difference
Count 1 - Count 2
Speed Difference
Velocity 1 - Velocity 2
Shift Logic
If RPM Difference < setpoint
and
Lower Setpoint < Count Difference < Upper Setpoint
SHIFT
Control Panel
Set Speed Difference
3
RPM
Set Count Difference
Lower Setpoint
22
Upper Setpoint
24
Start
RPM 1
RPM 2
RPM Difference
Count 1
Count 2
Count Difference
Shift Initiated?
Piston Position

FIG. 4B

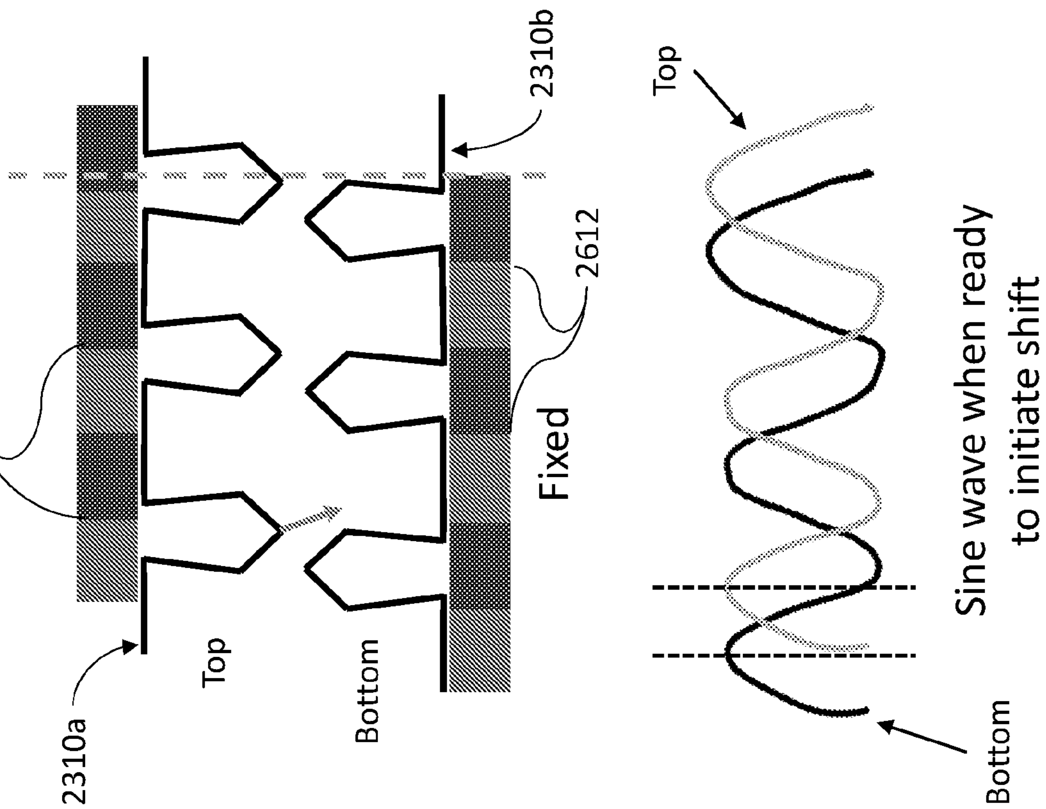

2310a
Top
Bottom
2612
2612
Fixed
2310b
Top
Bottom
Sine wave when ready
to initiate shift

FIG. 5B

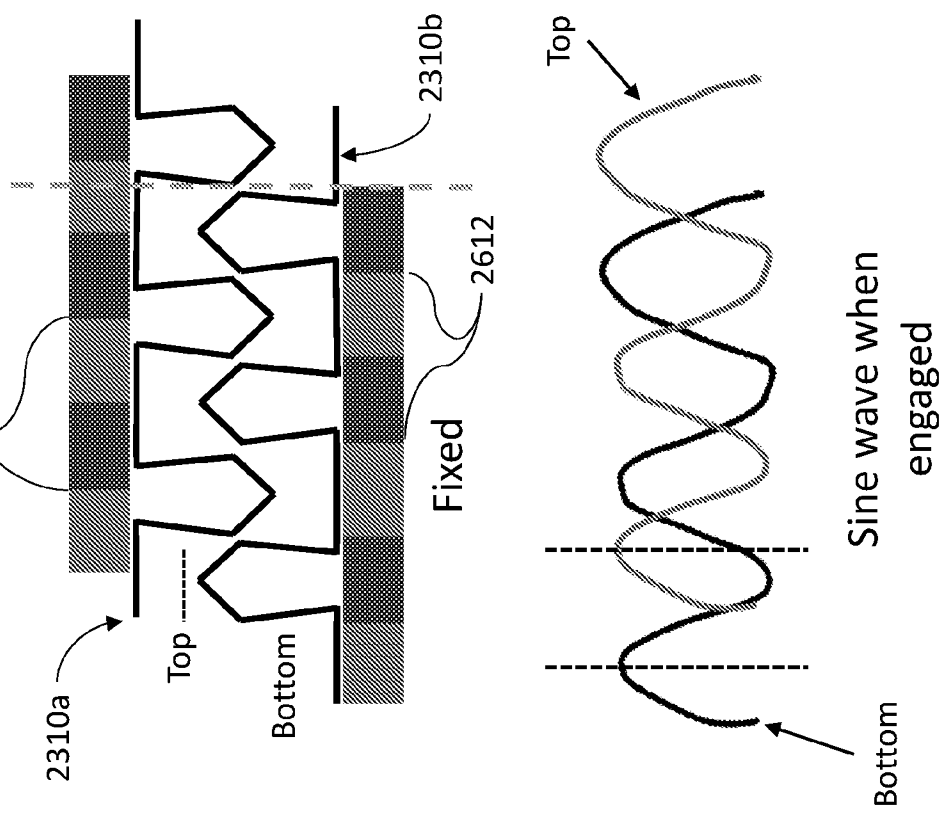

2310a
Top
Bottom
2612
2612
Fixed
2310b
Top
Bottom
Sine wave when
engaged

FIG. 5A

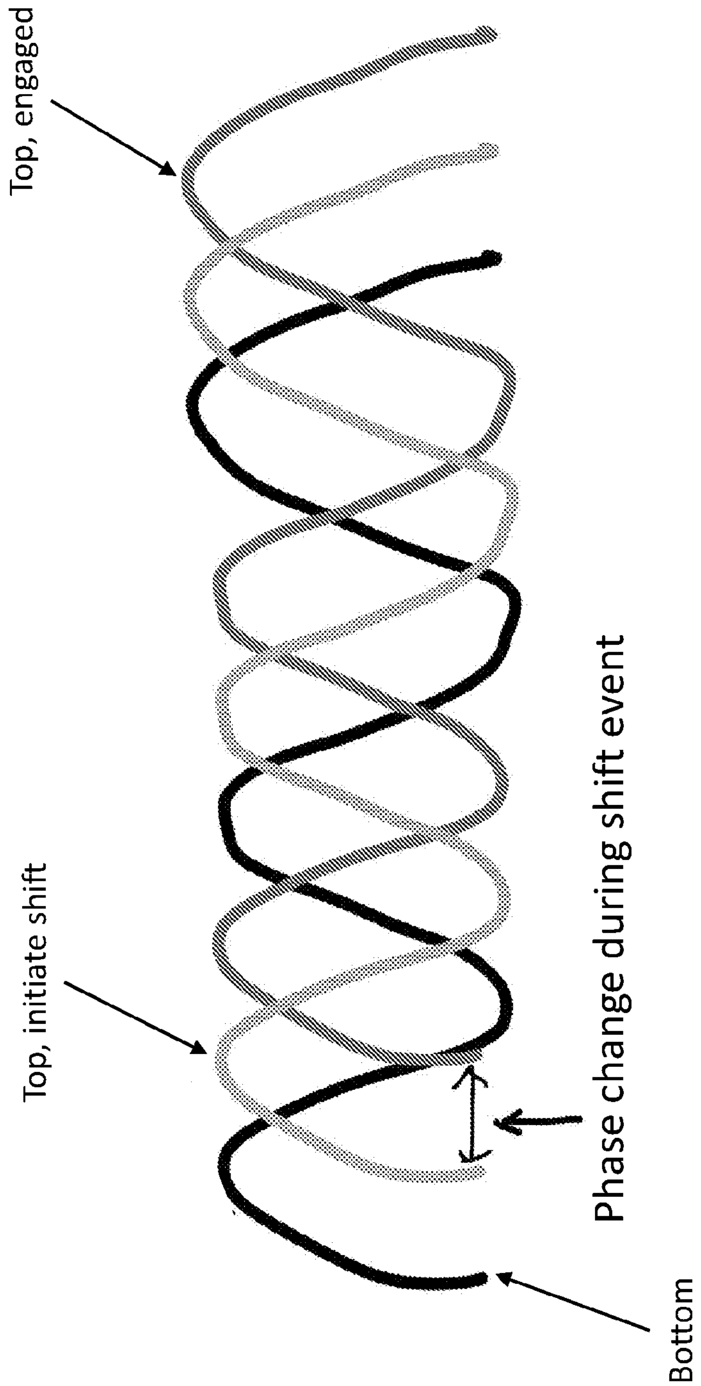
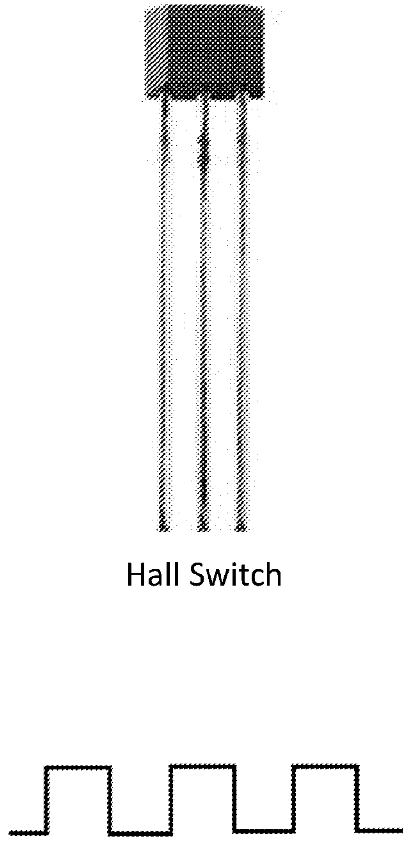
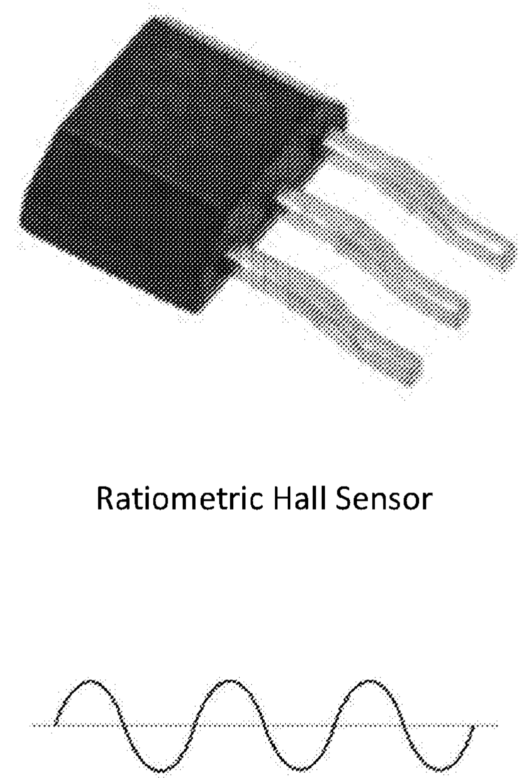

Top, engaged
Top, initiate shift
Phase change during shift event
Bottom

FIG. 5C



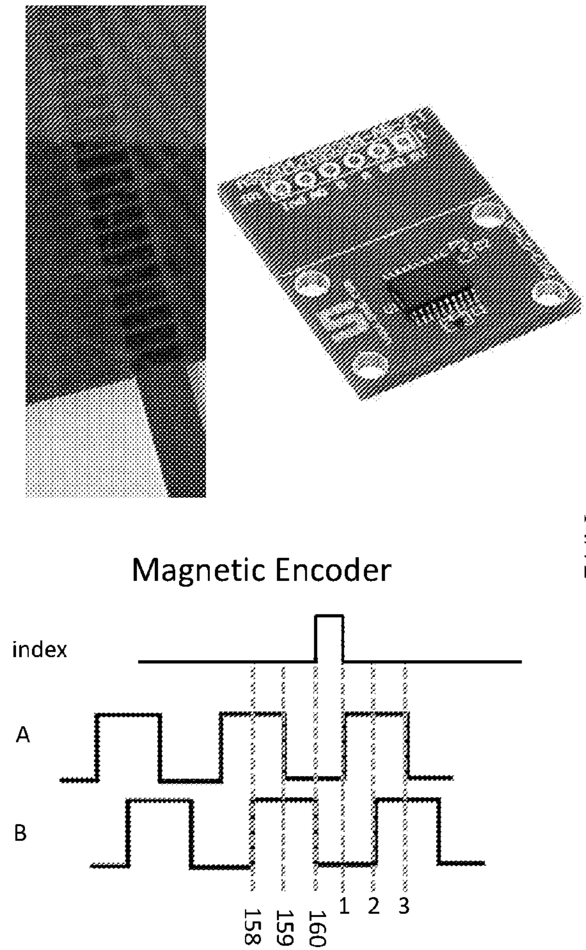
One square per tooth
0V or 5V with magnetic field

FIG. 6A



One sin wave per tooth
Varying voltage with magnetic field

FIG. 6B



160 counts per tooth

FIG. 6C

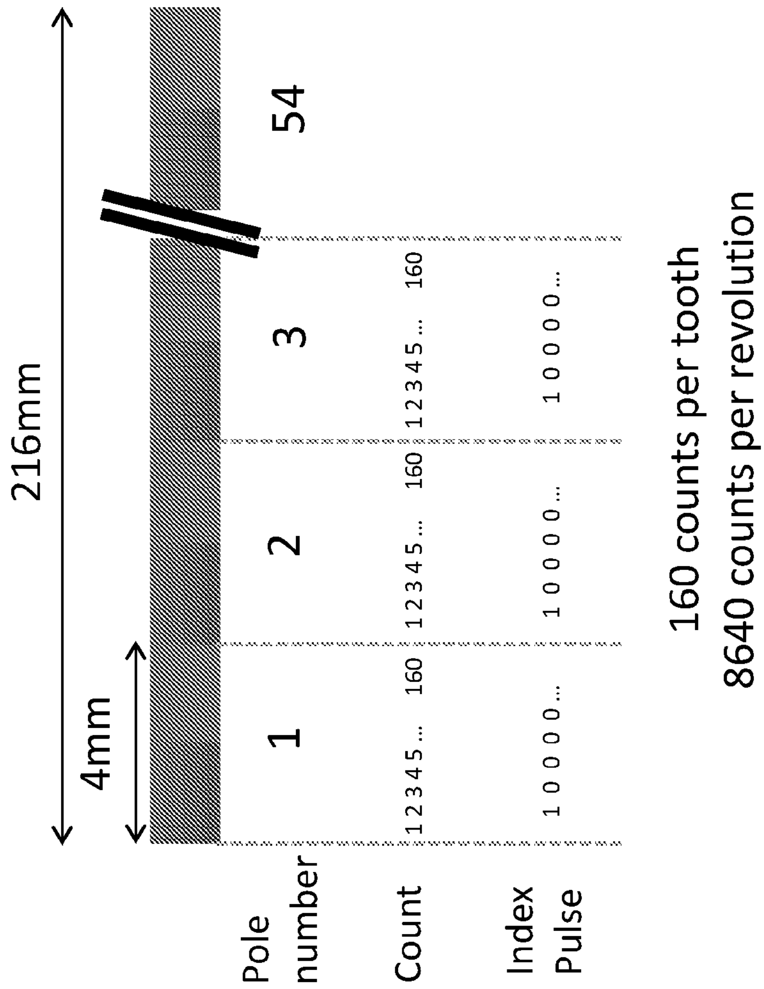

216mm
4mm
Pole number
Count
Index Pulse
1
2
3
54
1 2 3 4 5 ... 160
1 0 0 0 0 ...
160 counts per tooth
8640 counts per revolution

FIG. 7A

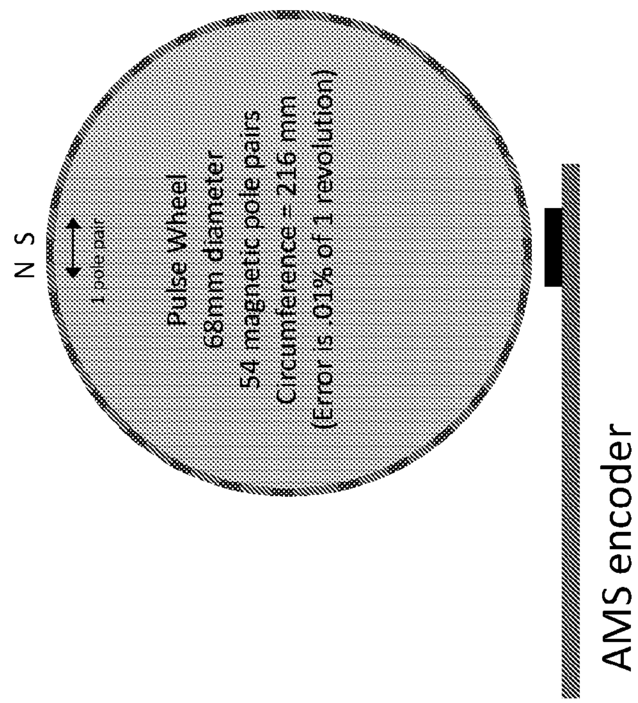

N S
1 pole pair
Pulse Wheel
68mm diameter
54 magnetic pole pairs
Circumference = 216 mm
(Error is .01% of 1 revolution)
AMS encoder

FIG. 7B

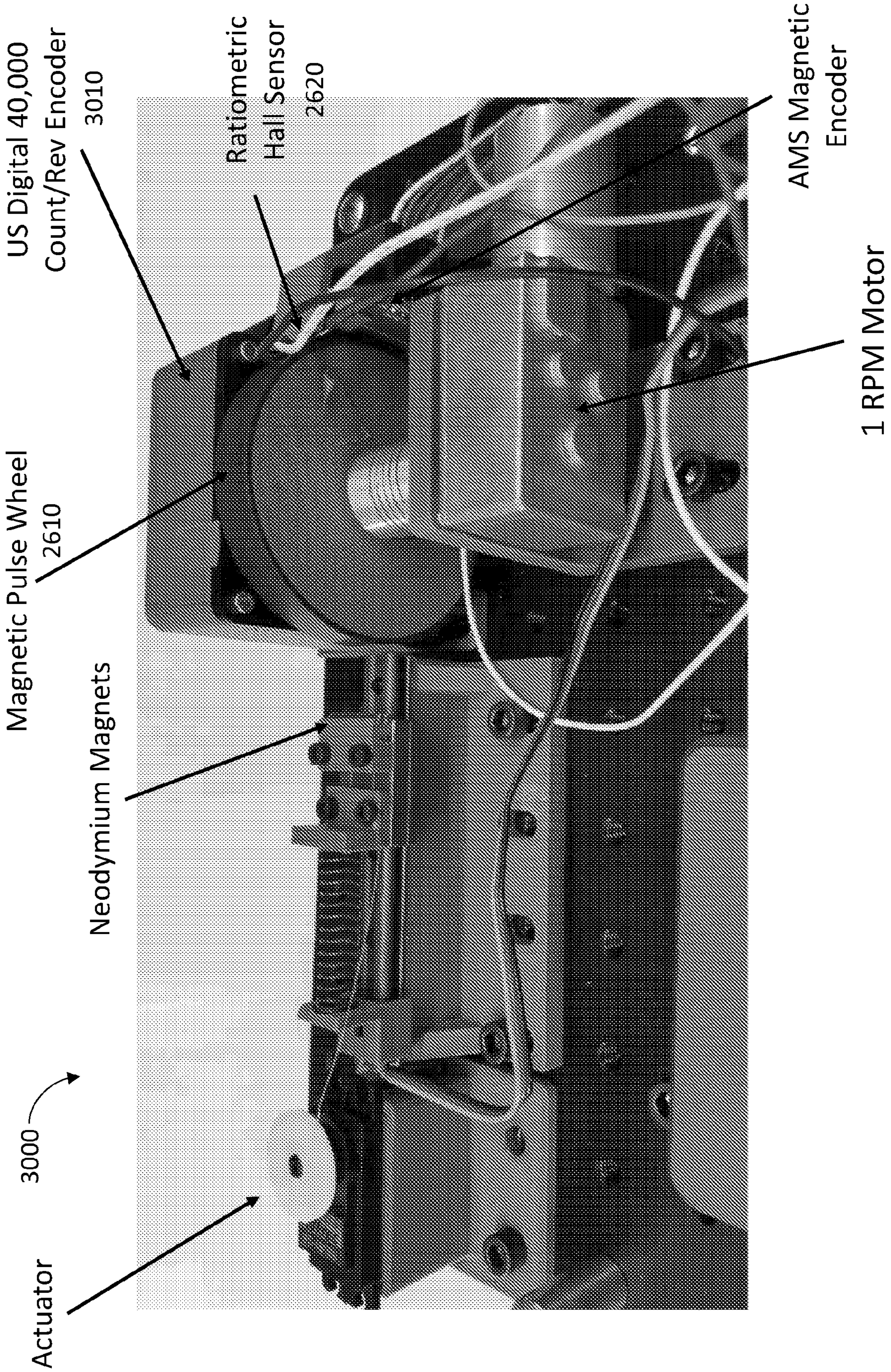

US Digital 40,000 Count/Rev Encoder
3010
Ratiometric Hall Sensor
2620
AMS Magnetic Encoder
Magnetic Pulse Wheel
2610
1 RPM Motor
Neodymium Magnets
3000
Actuator

FIG. 8A

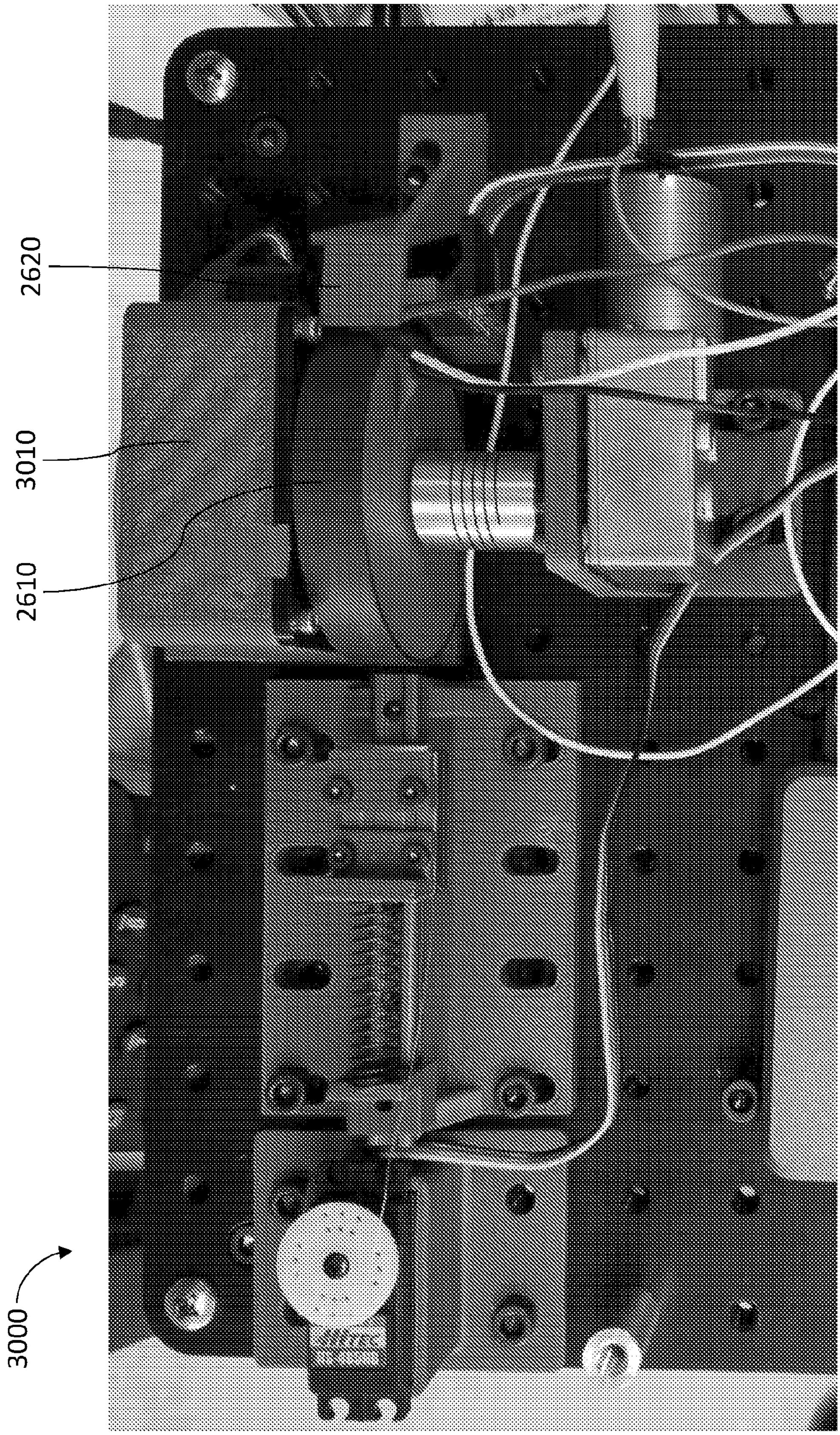


FIG. 8B

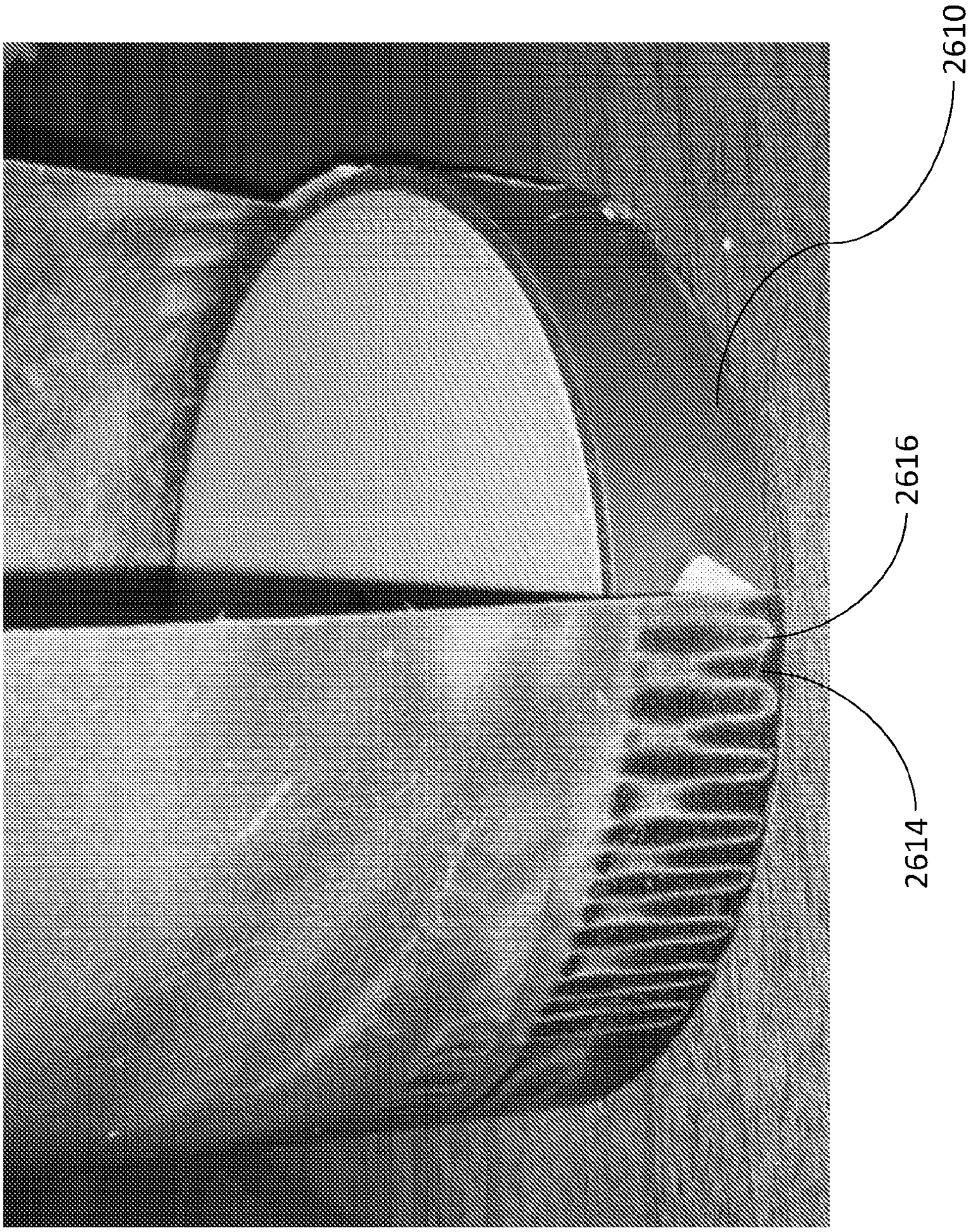


FIG. 8C

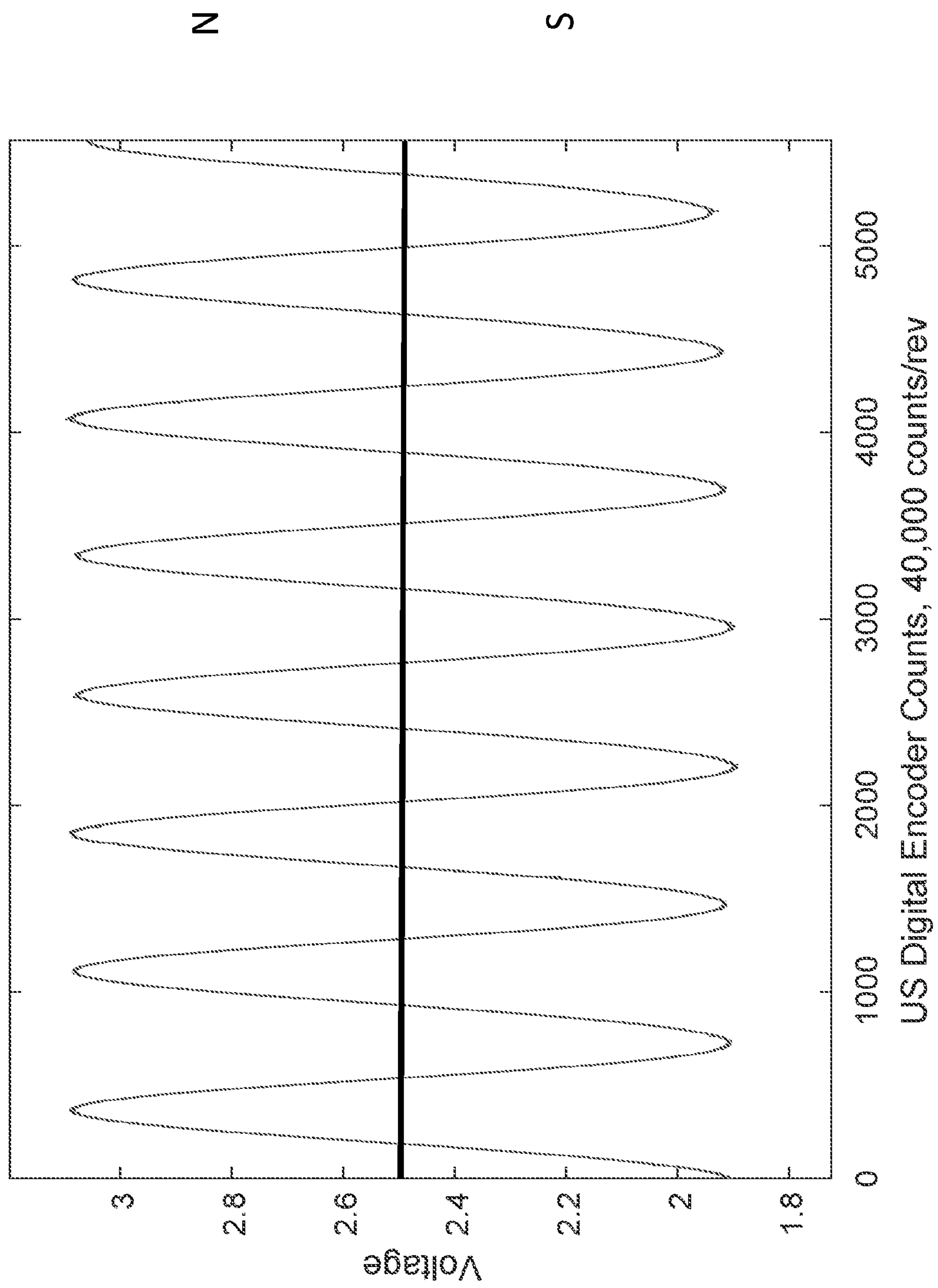

N
S
Voltage
3
2.8
2.6
2.4
2.2
2
1.8
0
1000
2000
3000
4000
5000
US Digital Encoder Counts, 40,000 counts/rev

FIG. 9A

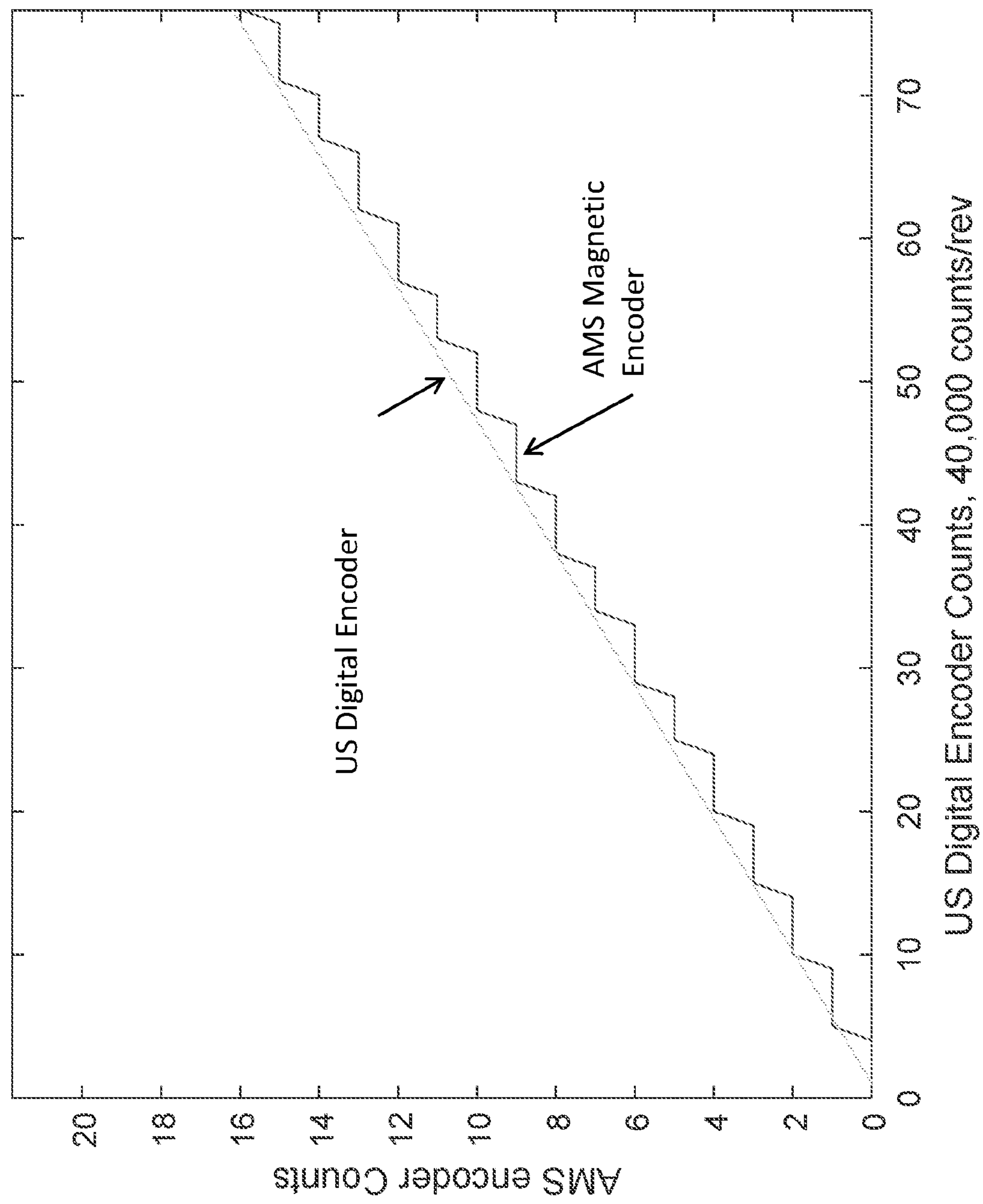

AMS encoder Counts
US Digital Encoder Counts, 40,000 counts/rev
US Digital Encoder
AMS Magnetic Encoder
0
2
4
6
8
10
12
14
16
18
20
0
10
20
30
40
50
60
70

FIG. 9B

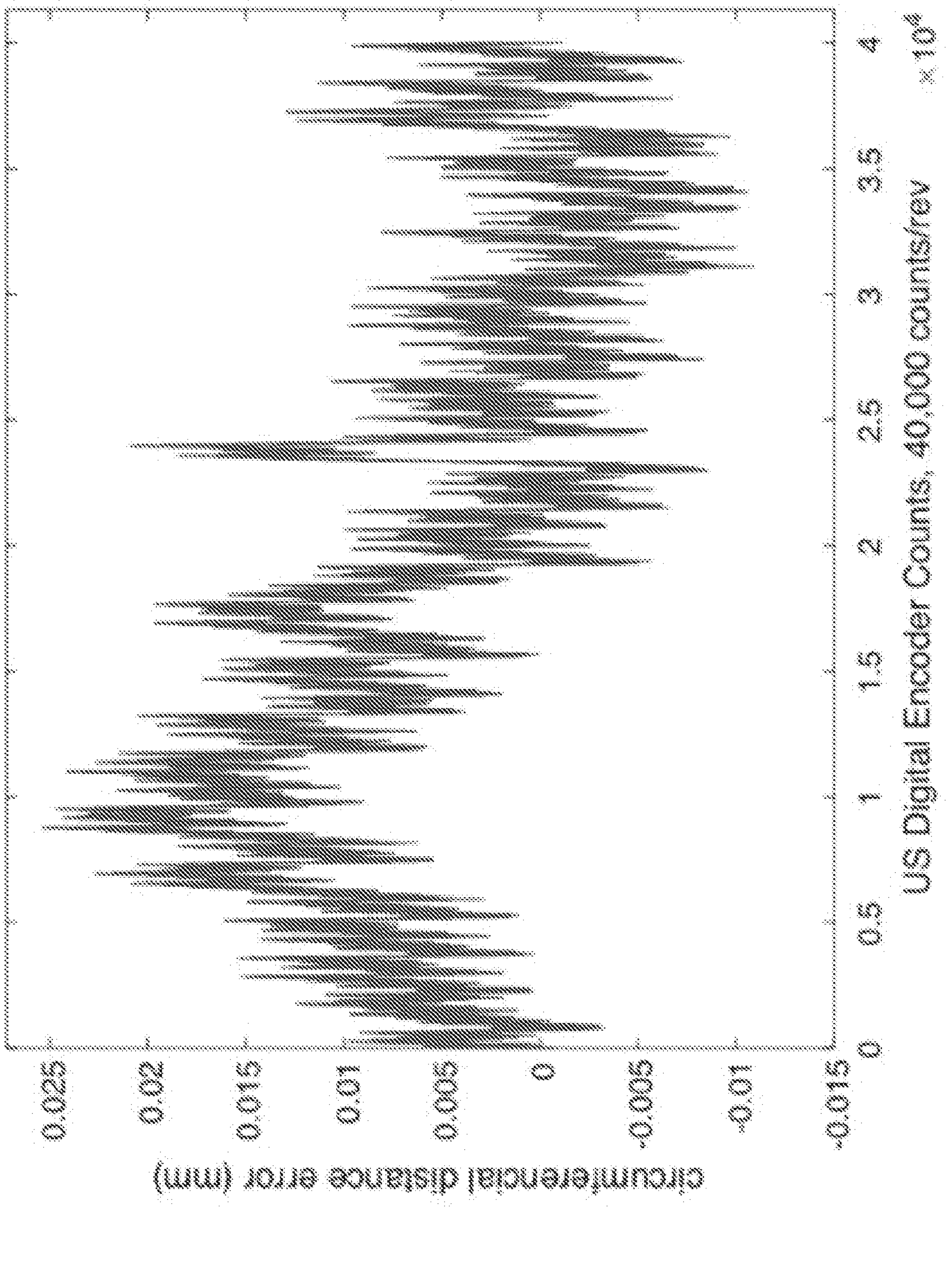

circumferencial distance error (mm)
0.025
0.02
0.015
0.01
0.005
0
-0.005
-0.01
-0.015
0
0.5
1
1.5
2
2.5
3
3.5
4
x 10^4
US Digital Encoder Counts, 40,000 counts/rev

FIG. 9C

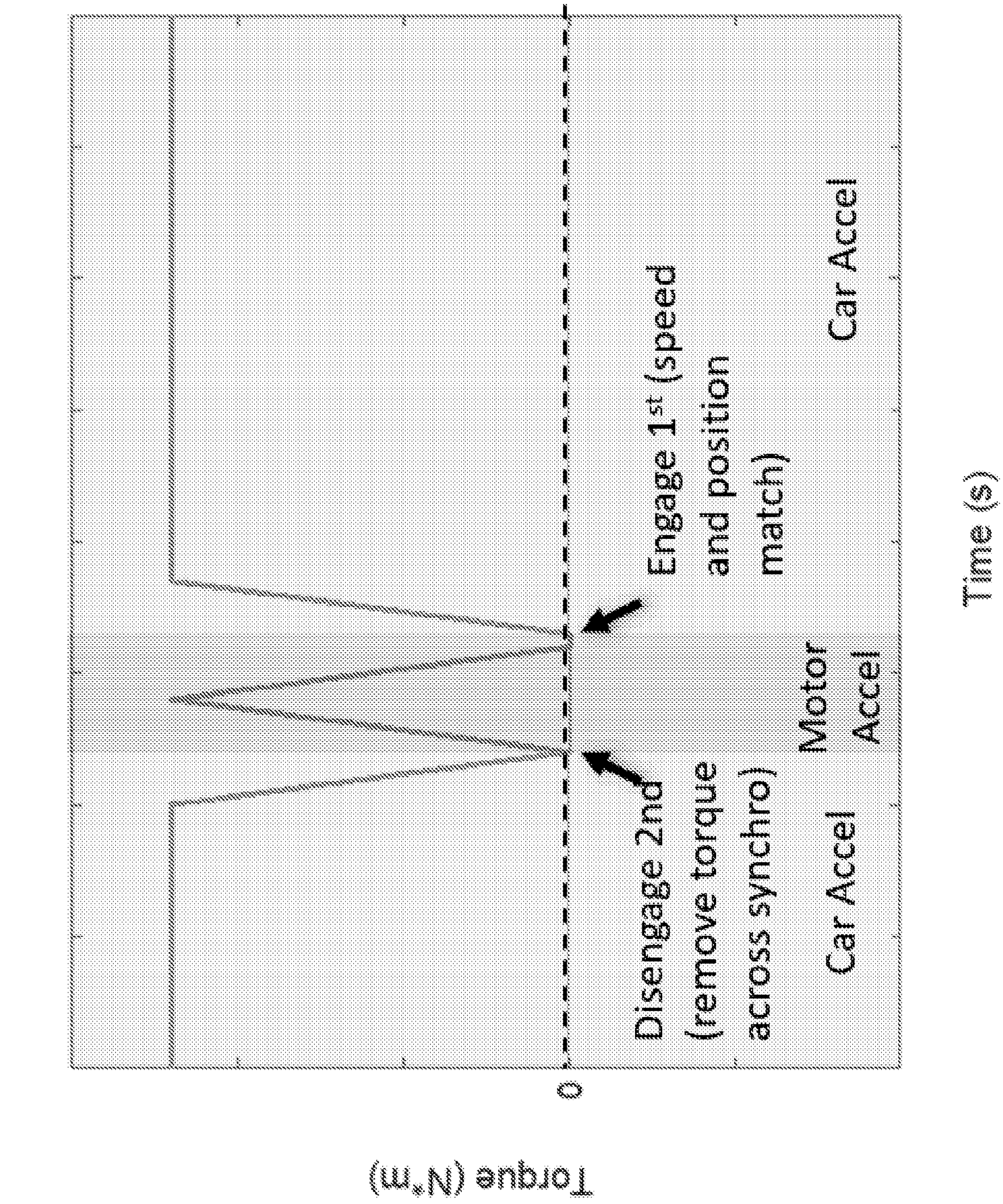

Torque (N*m)
0
Disengage 2nd (remove torque across synchro)
Car Accel
Motor Accel
Engage 1st (speed and position match)
Car Accel
Time (s)

FIG. 10A

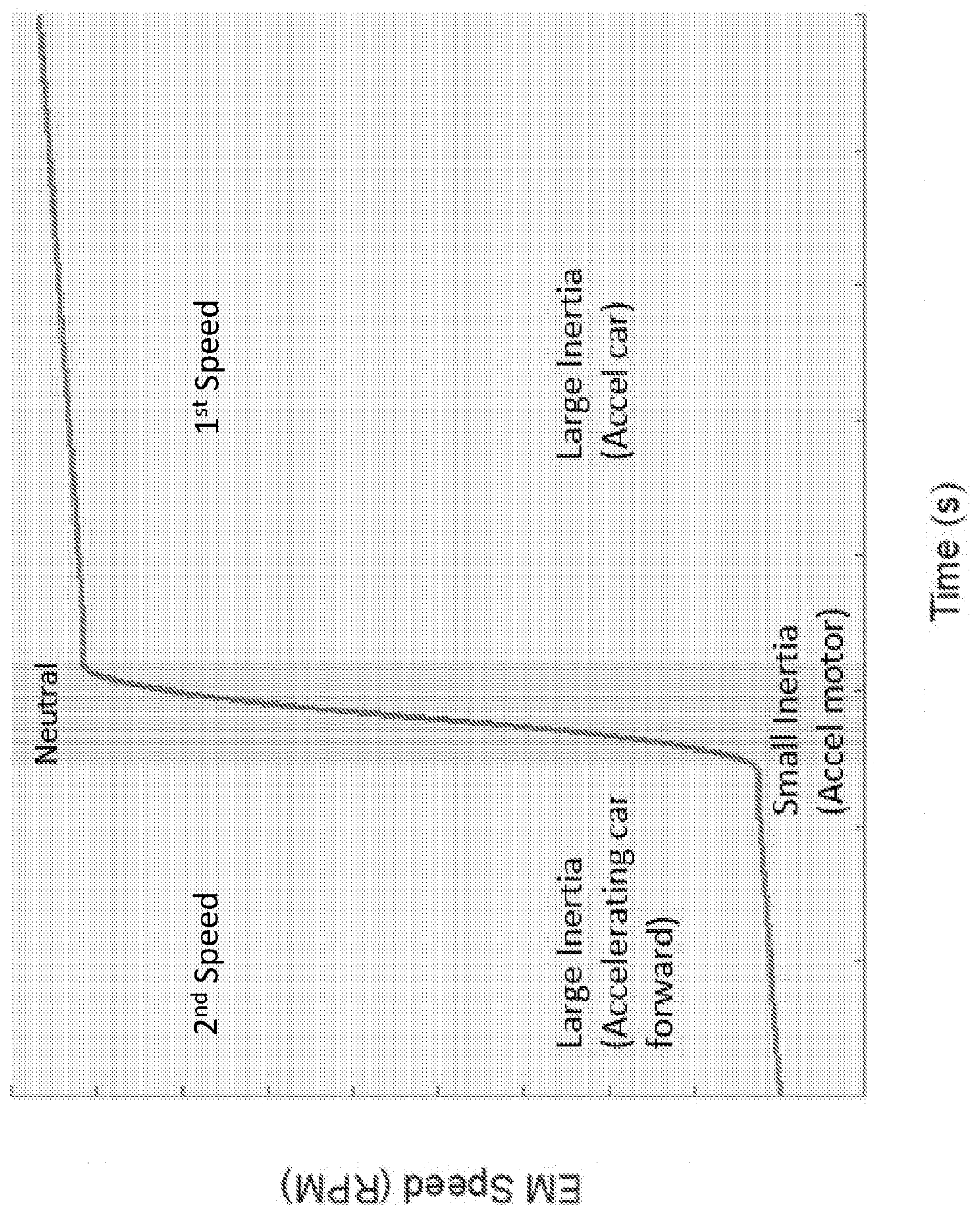

2nd Speed
Large Inertia
(Accelerating car
forward)
Neutral
Small Inertia
(Accel motor)
1st Speed
Large Inertia
(Accel car)
Time (s)
EM Speed (RPM)

FIG. 10B

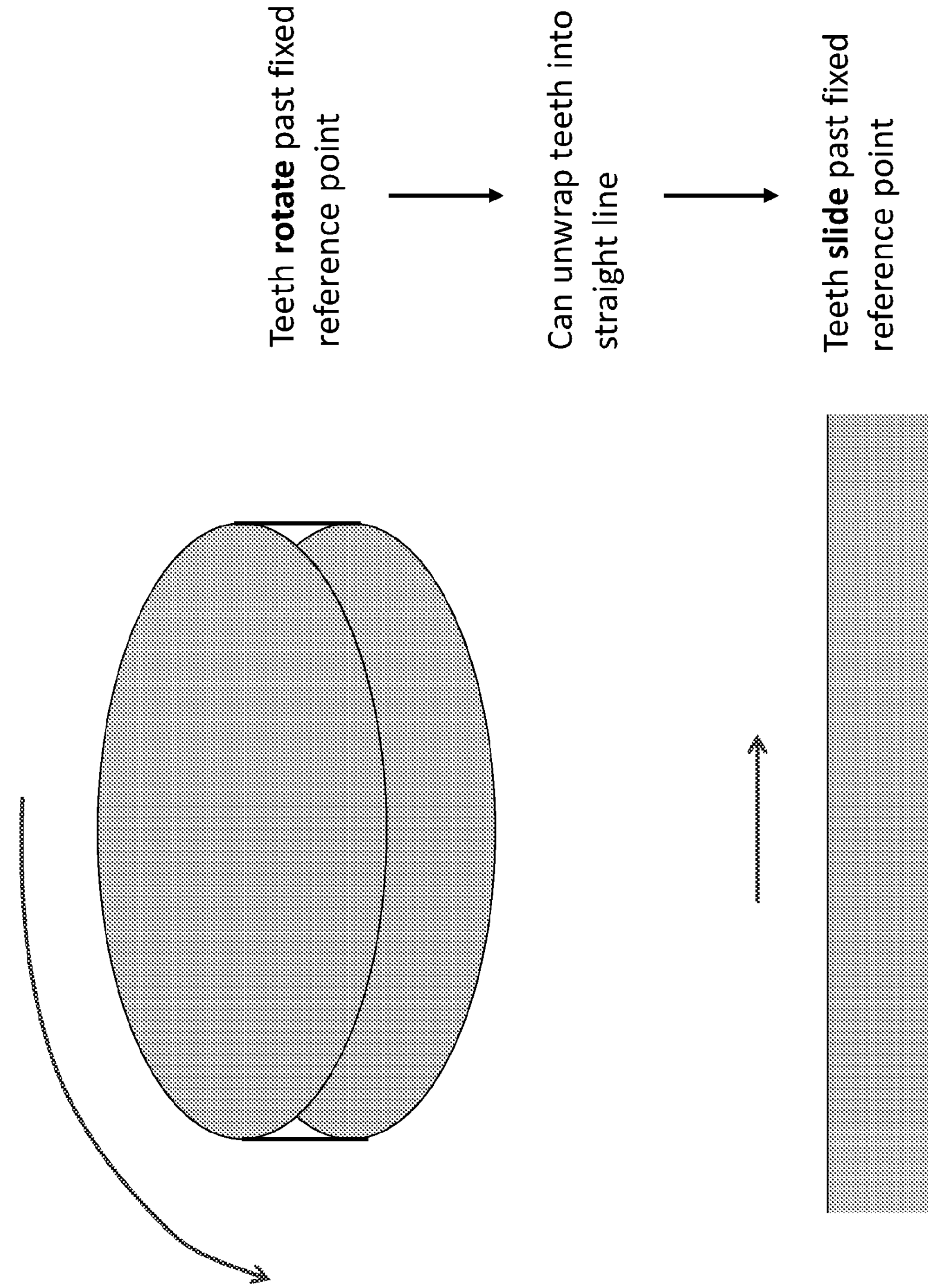

Teeth **rotate** past fixed reference point
Can unwrap teeth into straight line
Teeth **slide** past fixed reference point

FIG. 11A

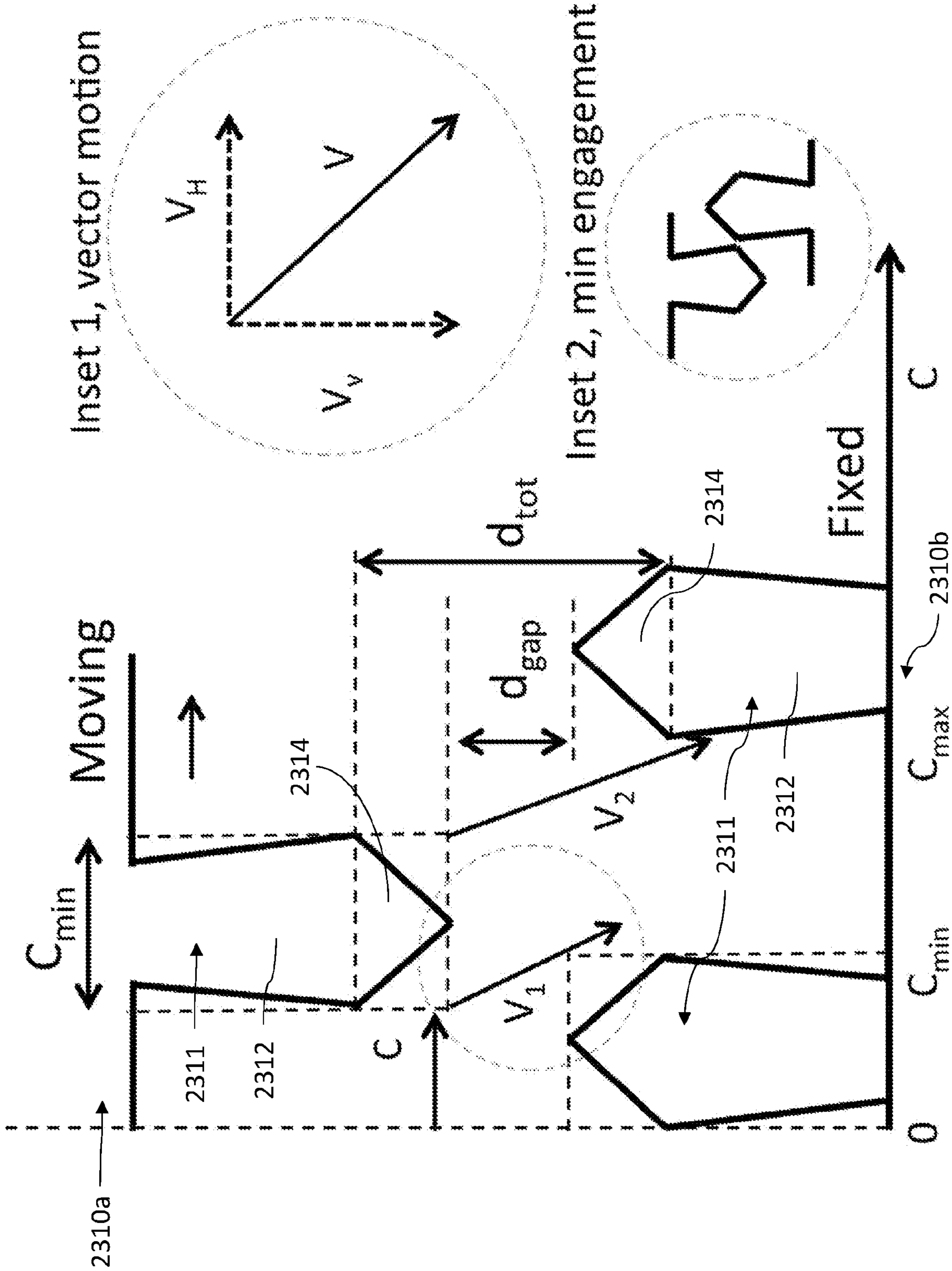

Inset 1, vector motion
V_H
V
V_V
Inset 2, min engagement
Moving
C_min
d_tot
d_gap
2314
2311
2312
C
V_1
V_2
2314
2311
2312
Fixed
2310a
2310b
0
C_min
C_max
C

FIG. 11B

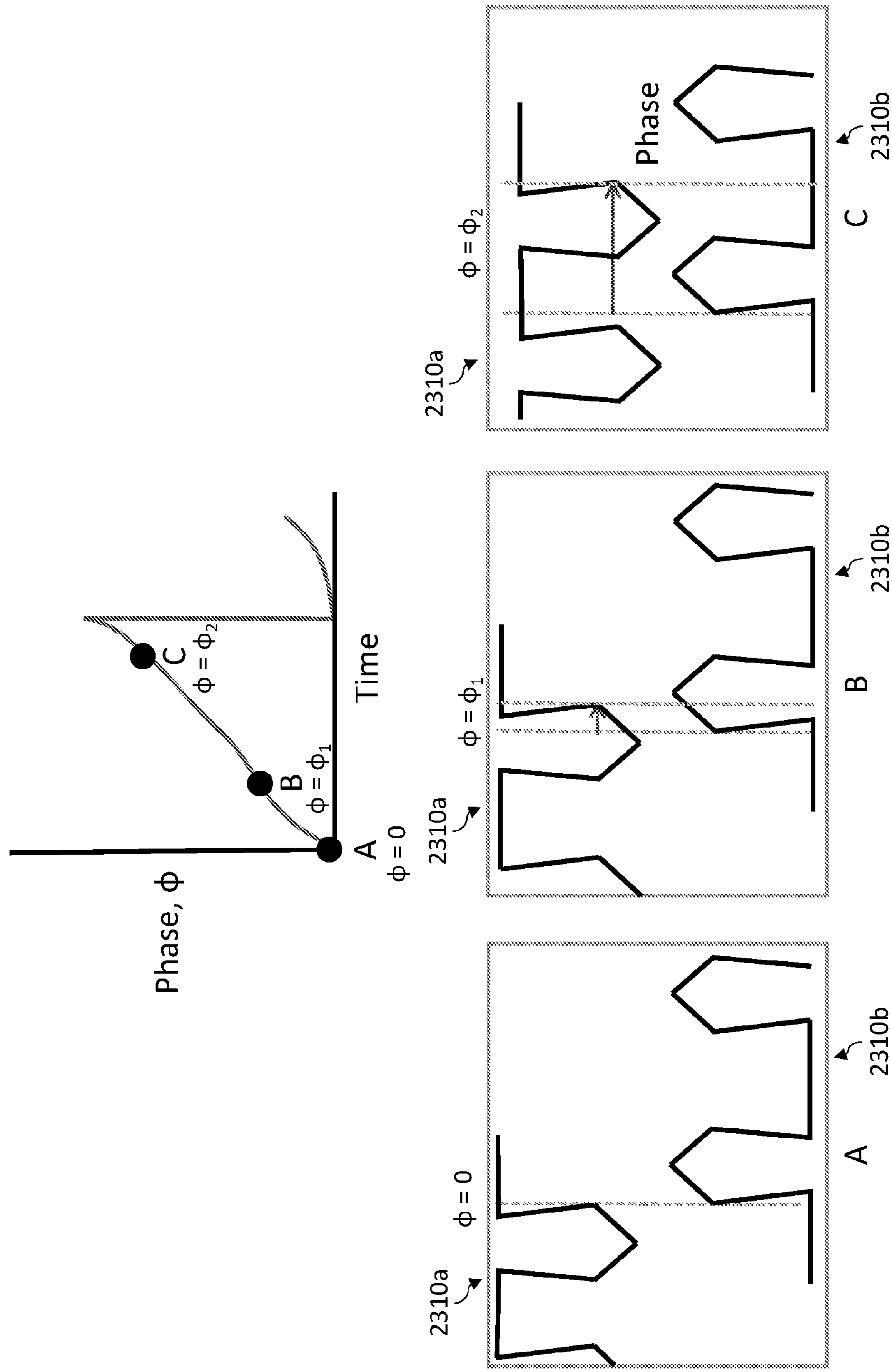

Phase, ϕ
Time
A
ϕ = 0
B
ϕ = ϕ1
C
ϕ = ϕ2
2310a
2310b
ϕ = 0
A
2310a
2310b
ϕ = ϕ1
B
2310a
2310b
ϕ = ϕ2
Phase
C

FIG. 11C

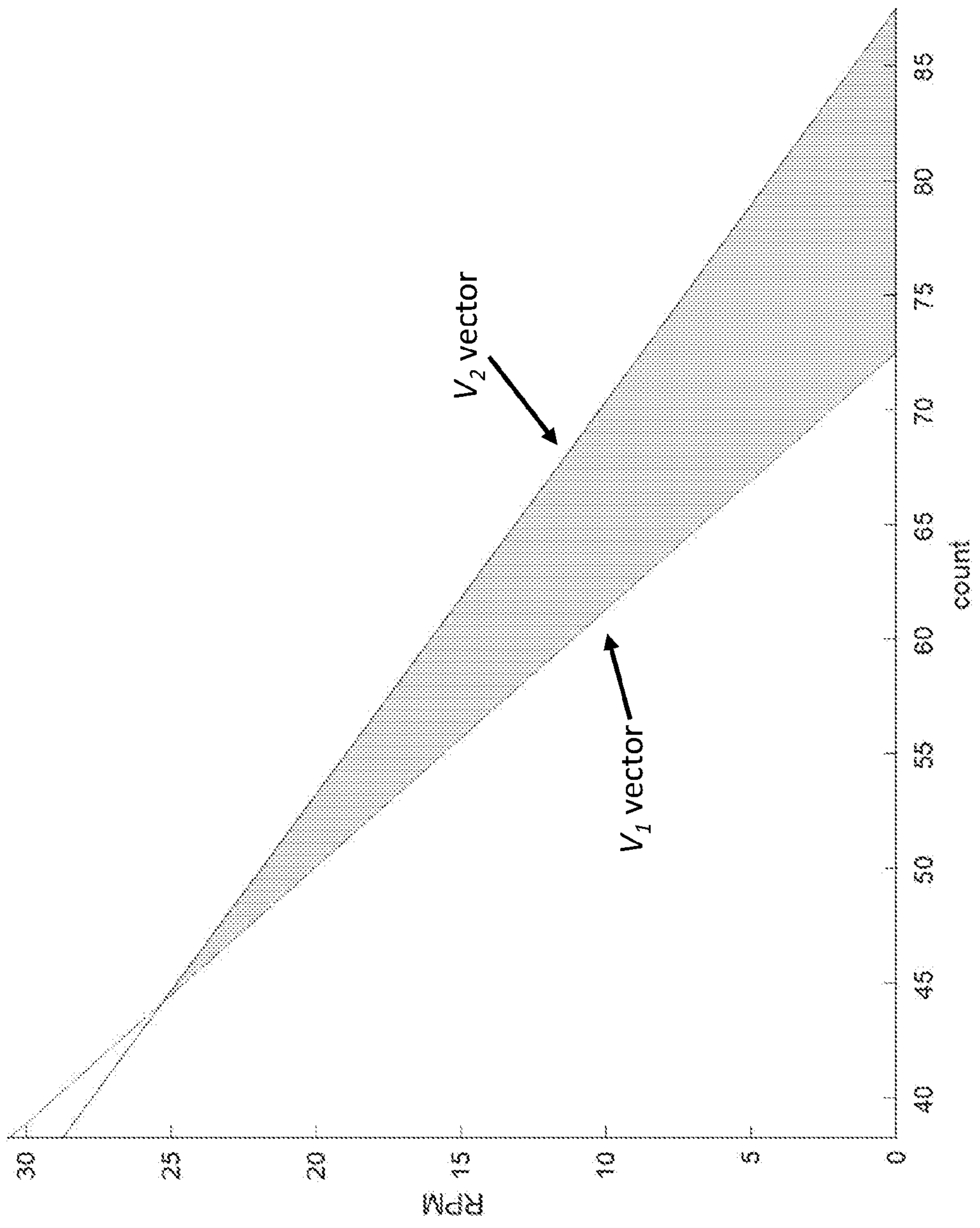

V_2 vector
V_1 vector
RPM
count
0
5
10
15
20
25
30
40
45
50
55
60
65
70
75
80
85

FIG. 11D

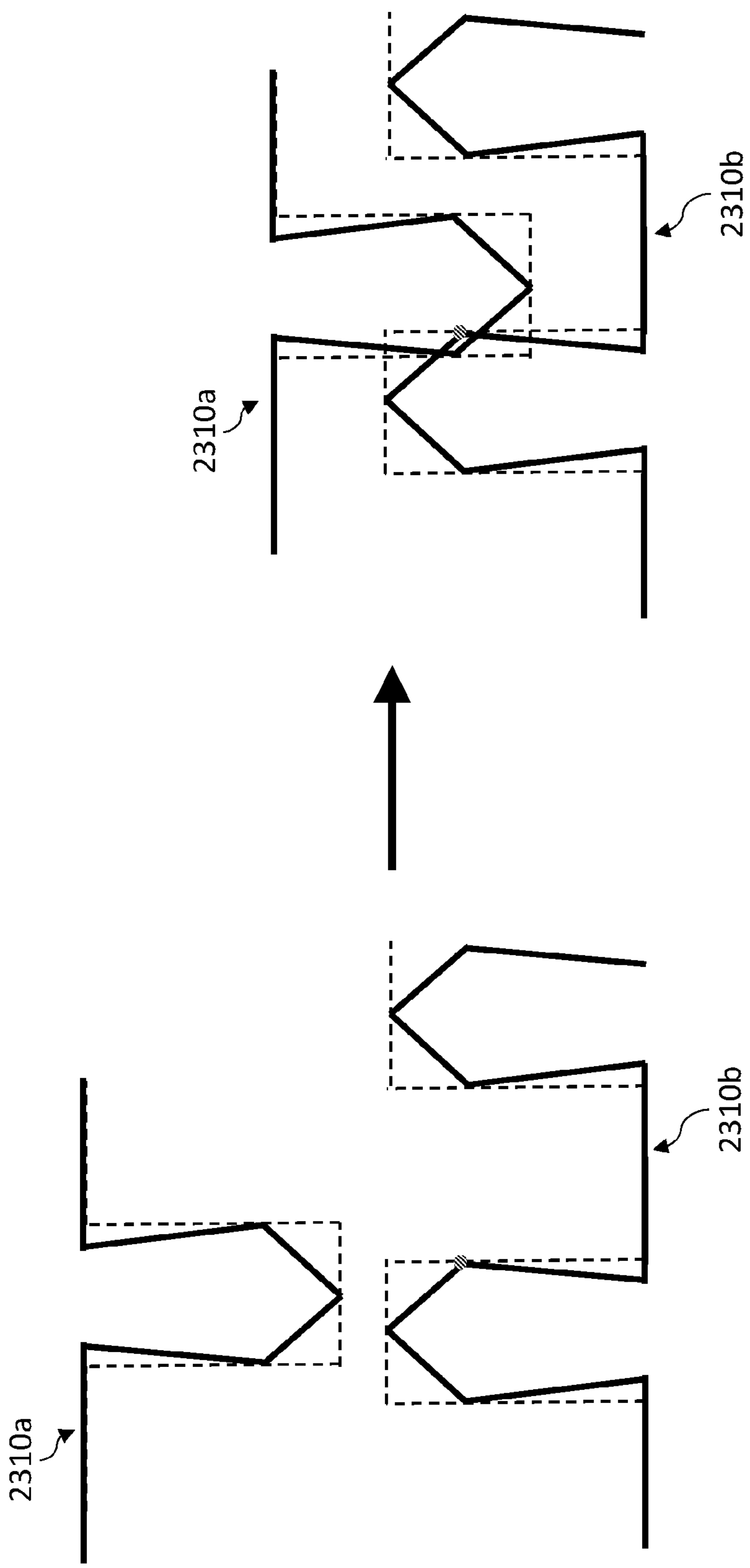


FIG. 12A

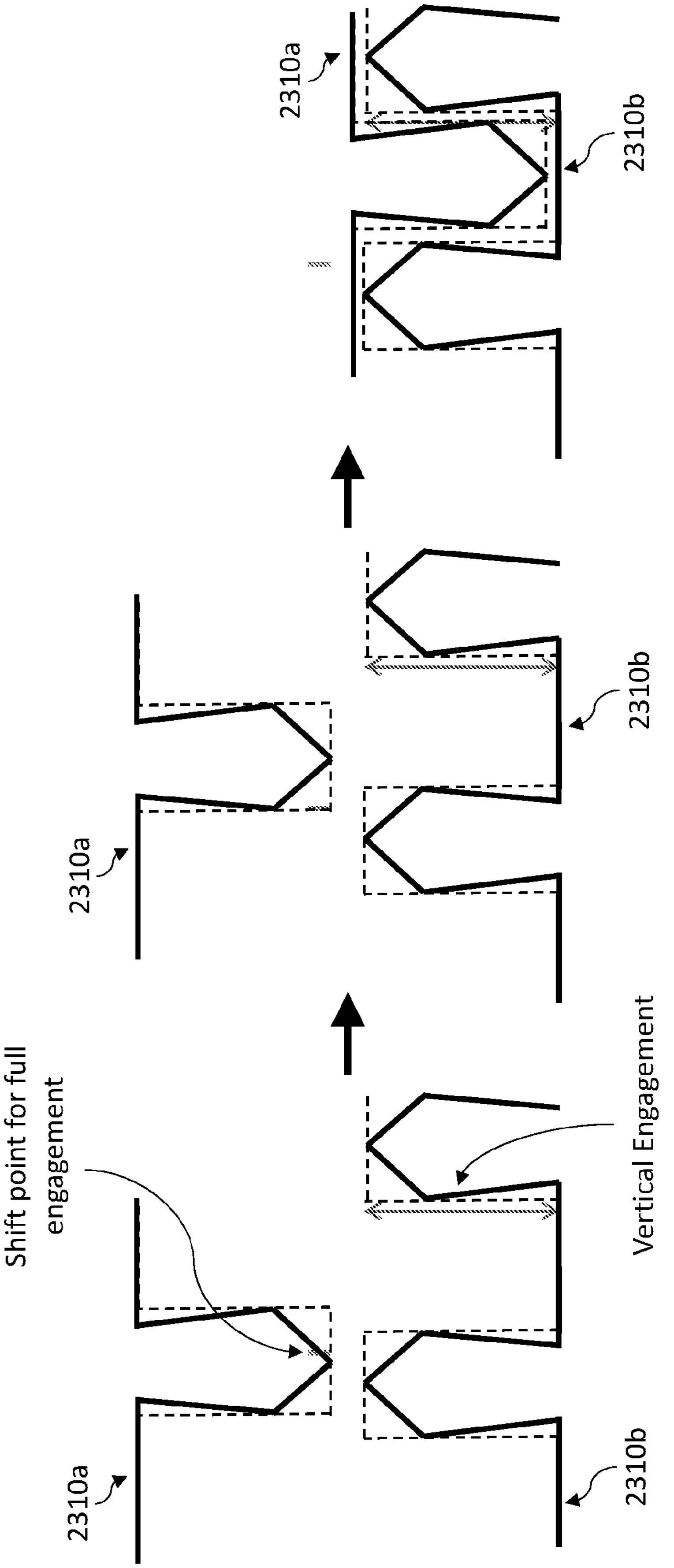

2310a
2310b
2310a
2310b
Shift point for full engagement
Vertical Engagement
2310a
2310b

FIG. 12B

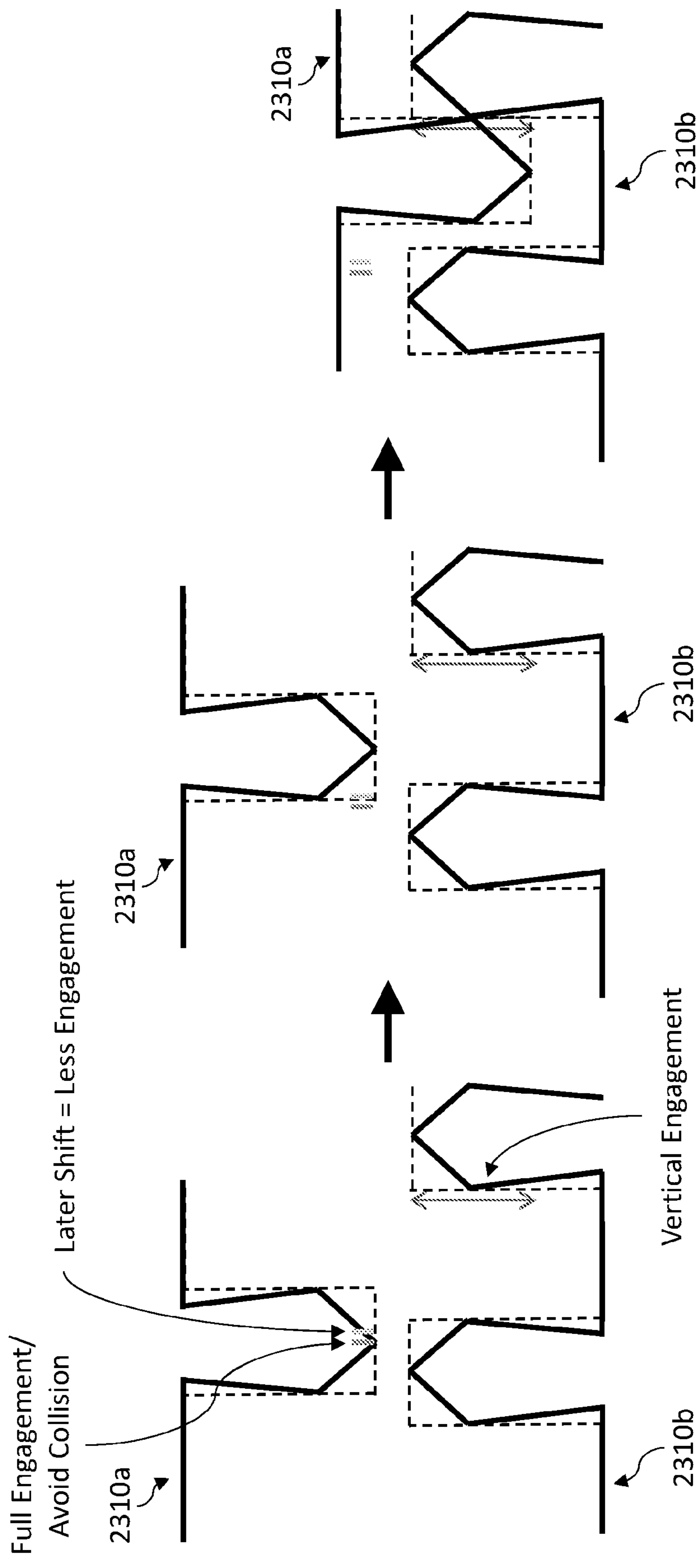

Full Engagement/
Avoid Collision
Later Shift = Less Engagement
2310a
2310b
Vertical Engagement
2310a
2310b
2310a
2310b

FIG. 12C

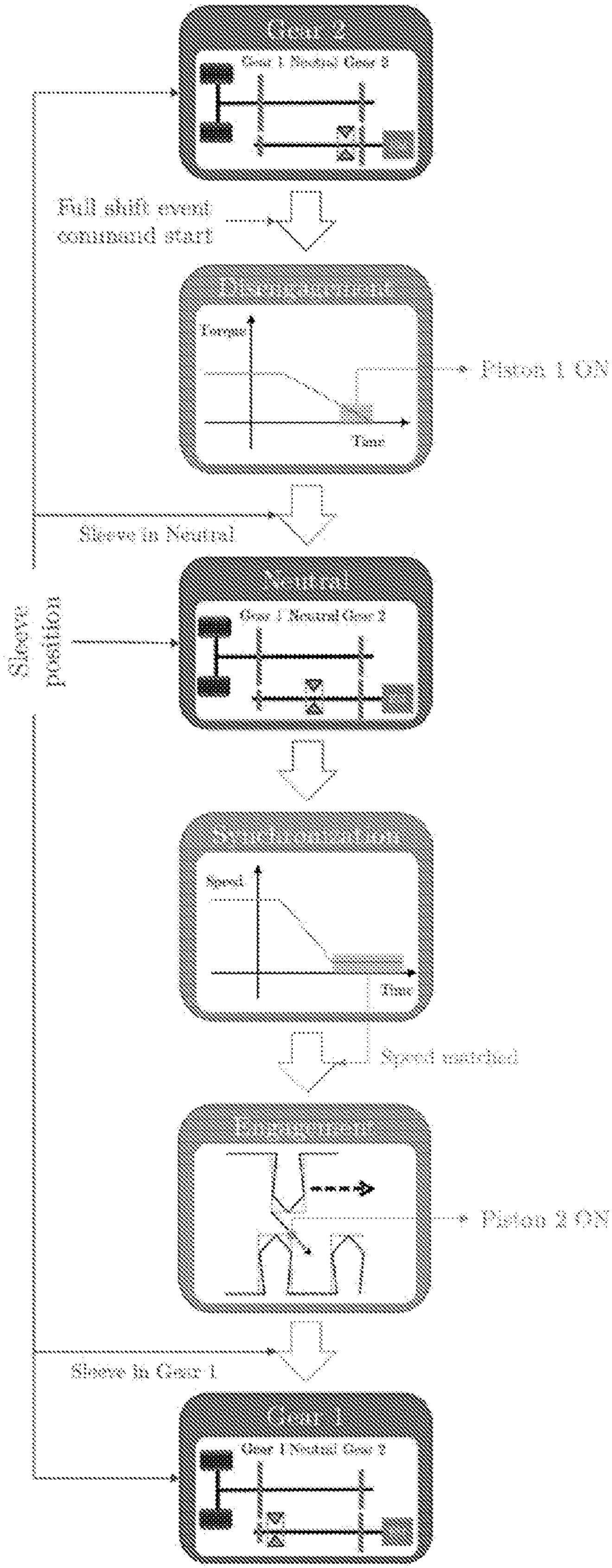

Gear 2
Gear 1 Neutral Gear 2
Full shift event command start
Disengagement
Torque
Time
Piston 1 ON
Sleeve in Neutral
Sleeve position
Neutral
Gear 1 Neutral Gear 2
Synchronization
Speed
Time
Speed matched
Engagement
Piston 2 ON
Sleeve in Gear 1
Gear 1
Gear 1 Neutral Gear 2

FIG. 13

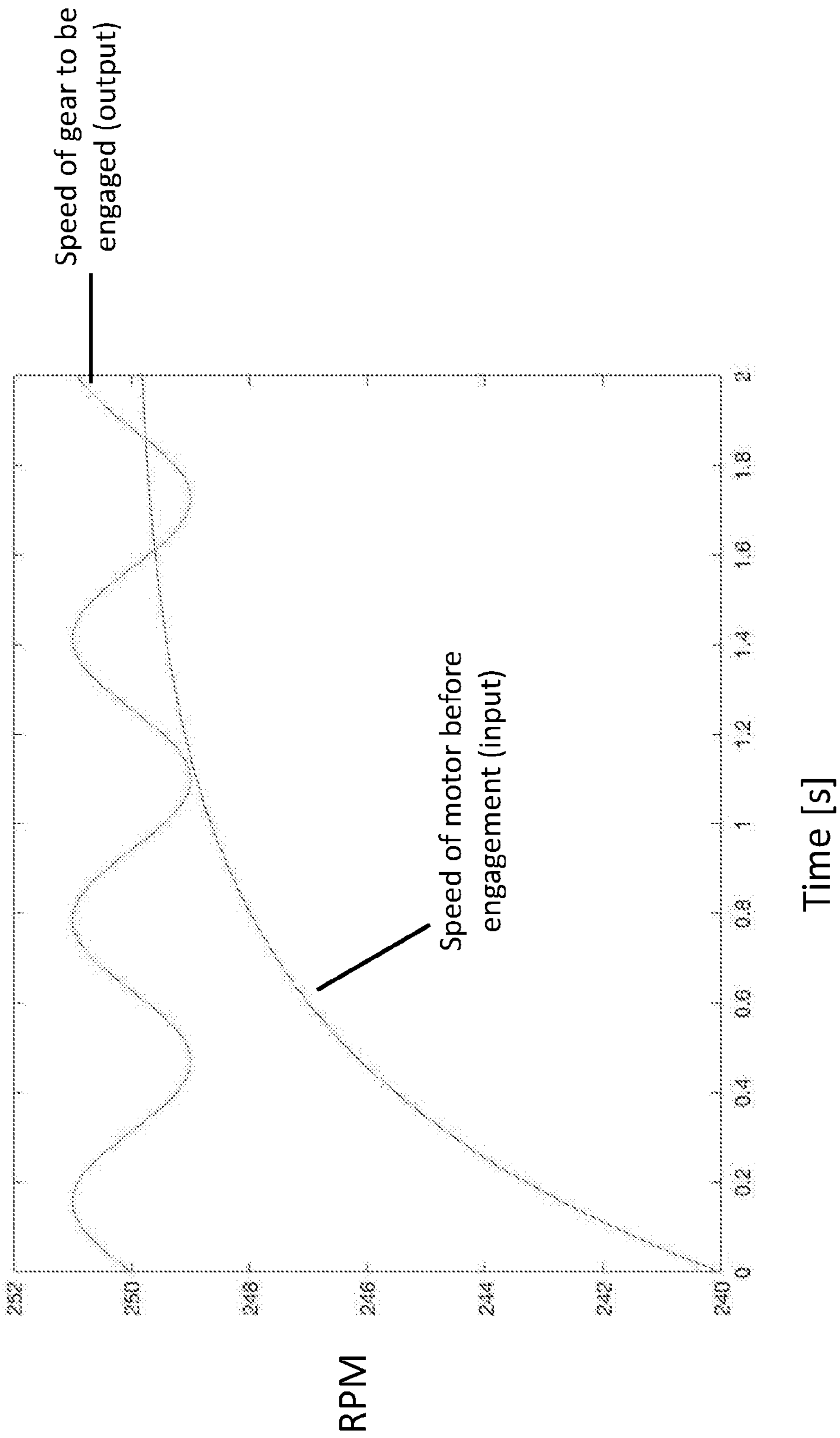

Speed of gear to be engaged (output)
Speed of motor before engagement (input)
RPM
252
250
248
246
244
242
240
0
0.2
0.4
0.6
0.8
1
1.2
1.4
1.6
1.8
2
Time [s]

FIG. 14A

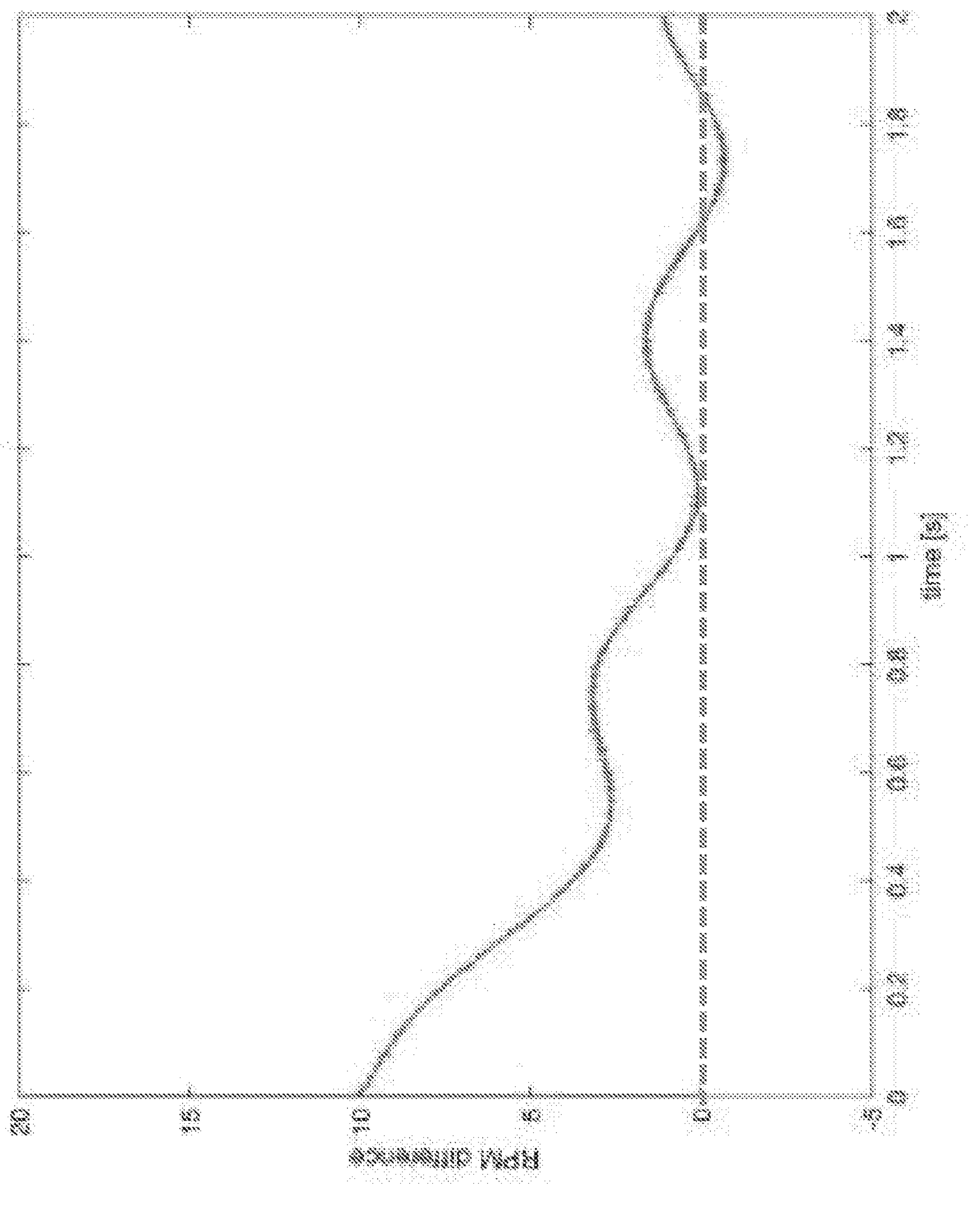


FIG. 14B

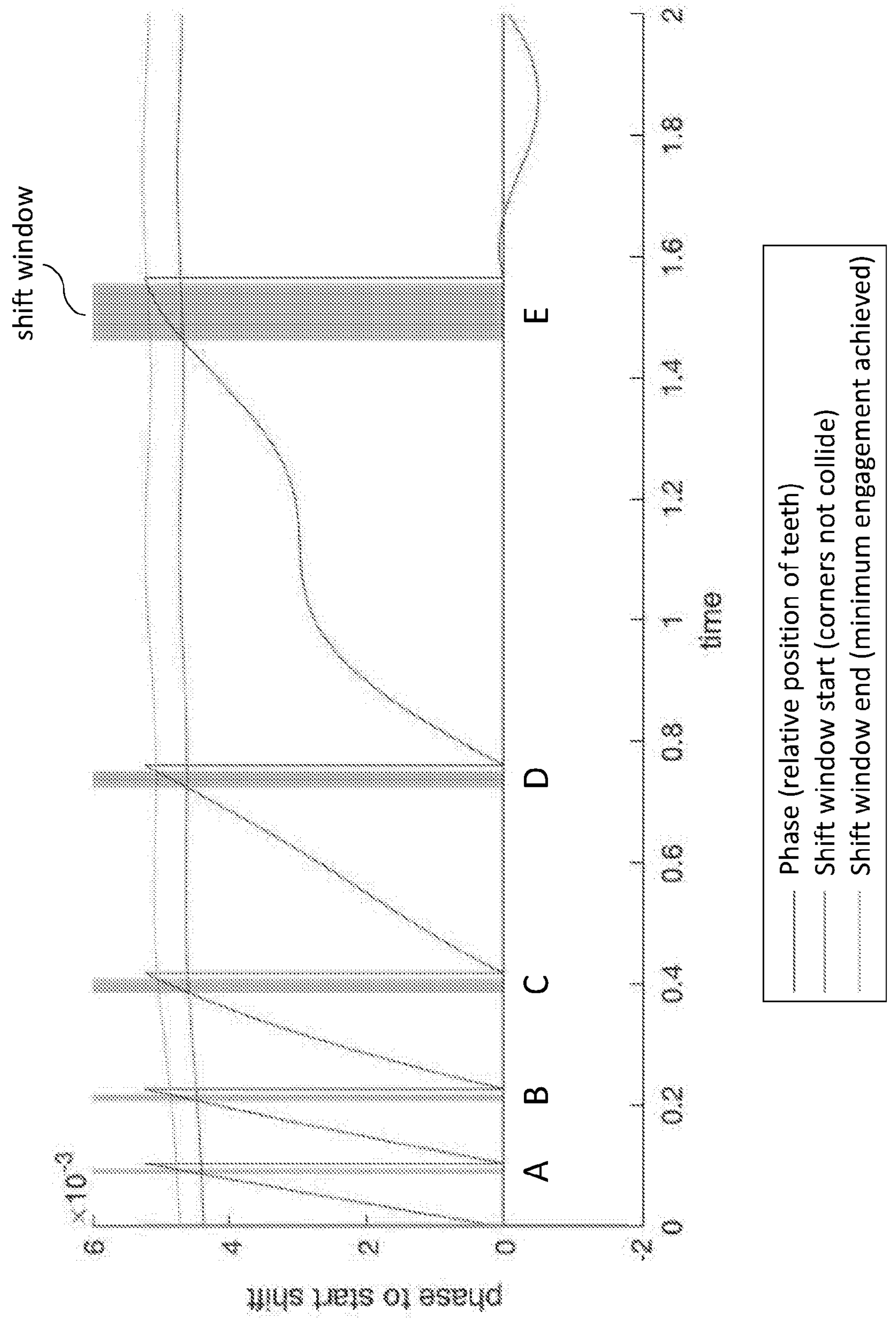

shift window
A
B
C
D
E
phase to start shift
time
x10^-3
Phase (relative position of teeth)
Shift window start (corners not collide)
Shift window end (minimum engagement achieved)

FIG. 14C

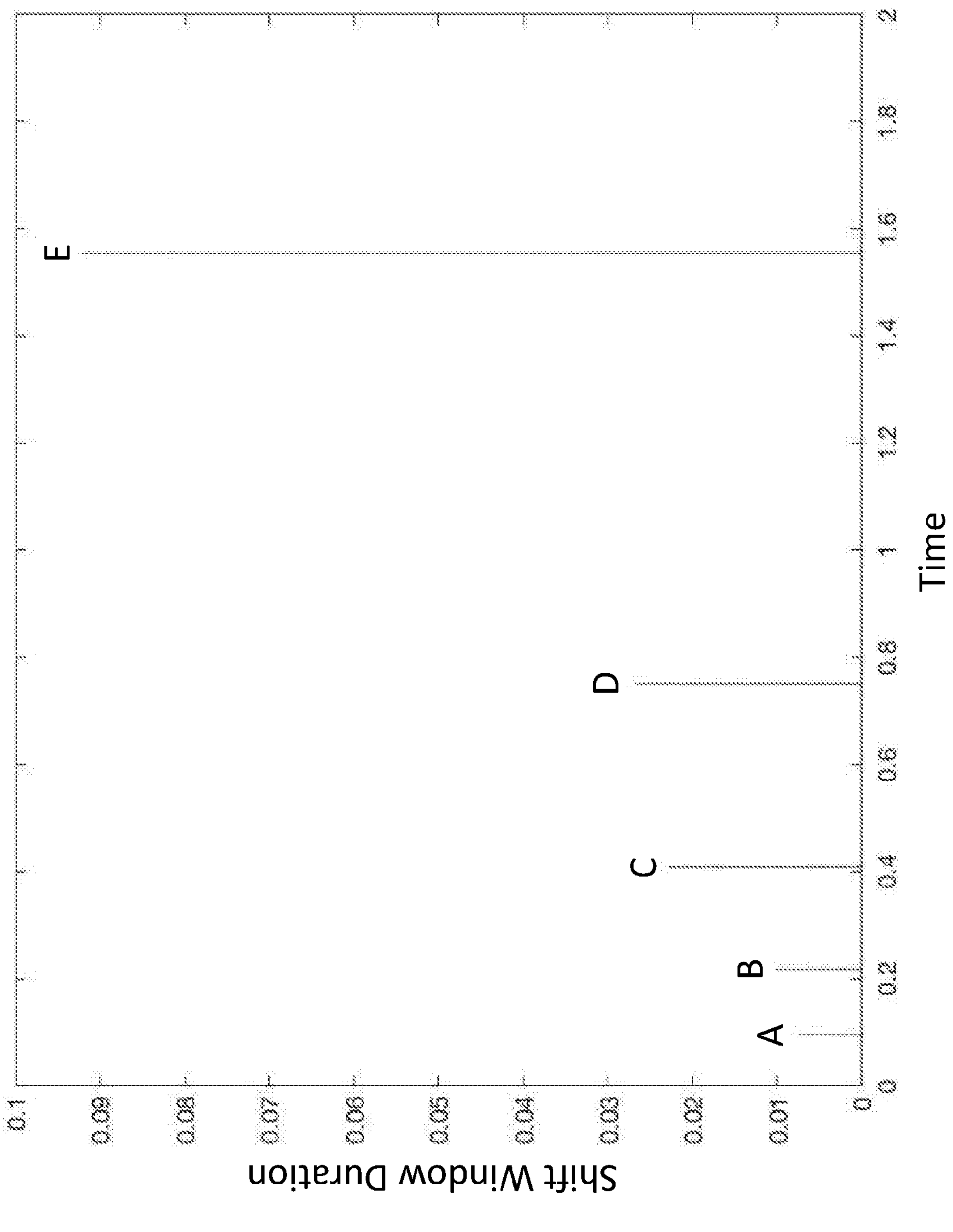

Shift Window Duration
0.1
0.09
0.08
0.07
0.06
0.05
0.04
0.03
0.02
0.01
0
Time
0
0.2
0.4
0.6
0.8
1
1.2
1.4
1.6
1.8
2
A
B
C
D
E

FIG. 14D

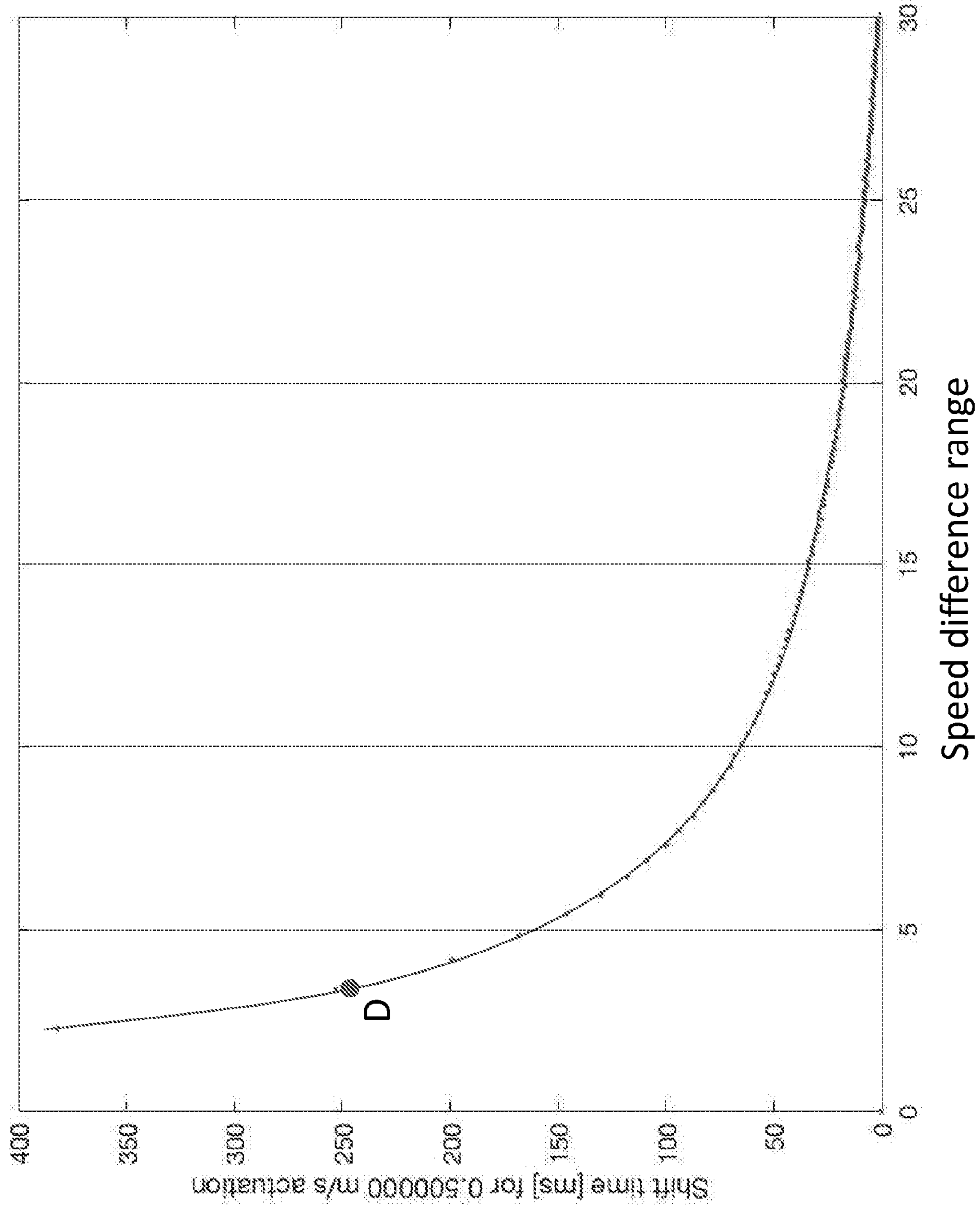

Shift time [ms] for 0.500000 m/s actuation
Speed difference range
D
0
50
100
150
200
250
300
350
400
0
5
10
15
20
25
30

FIG. 14E

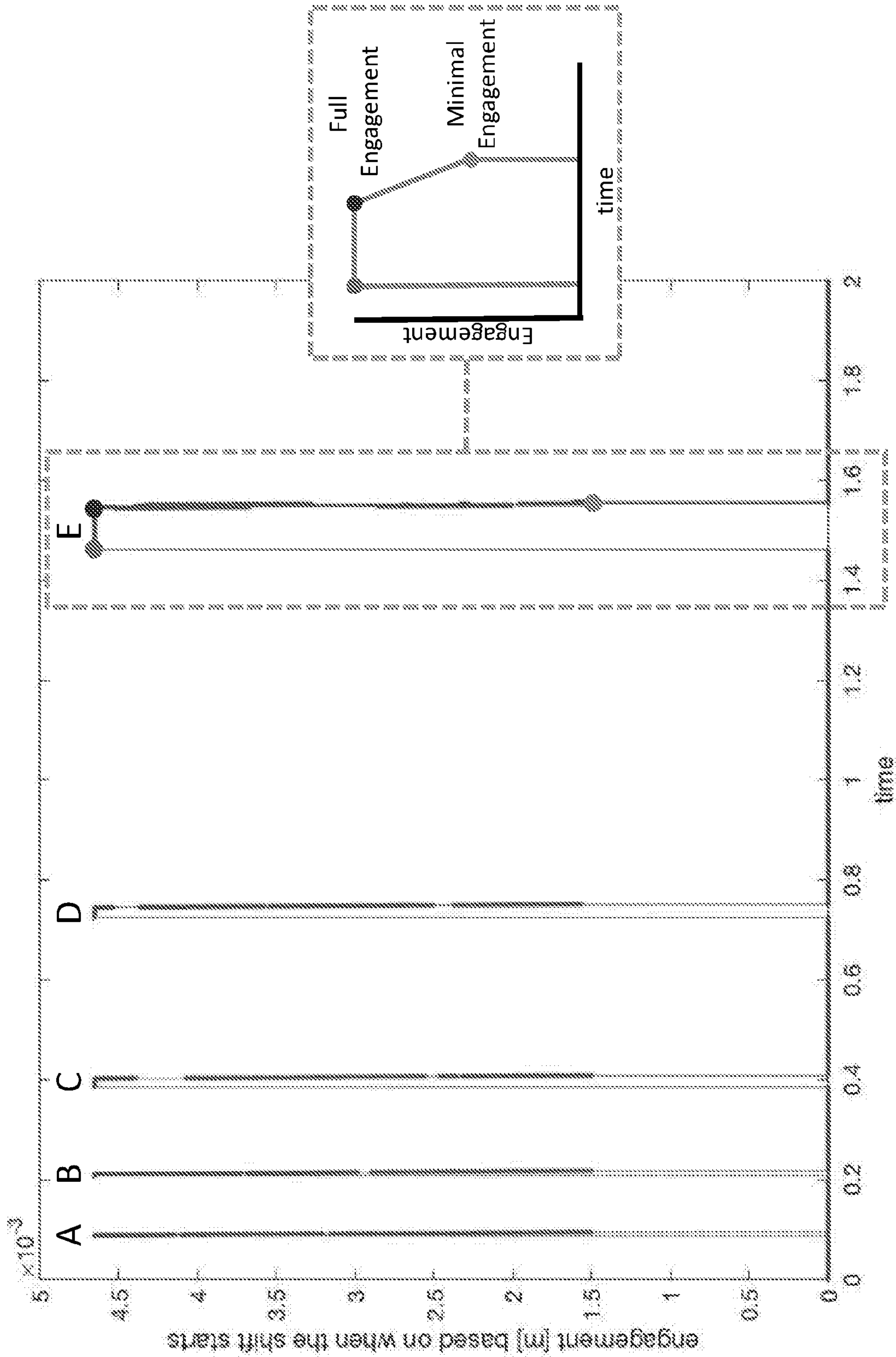

A
B
C
D
E
Full Engagement
Minimal Engagement
Engagement
time
engagement [m] based on when the shift starts
$\times 10^{-3}$
5
4.5
4
3.5
3
2.5
2
1.5
1
0.5
0
0.2
0.4
0.6
0.8
1
1.2
1.4
1.6
1.8
2

FIG. 14F

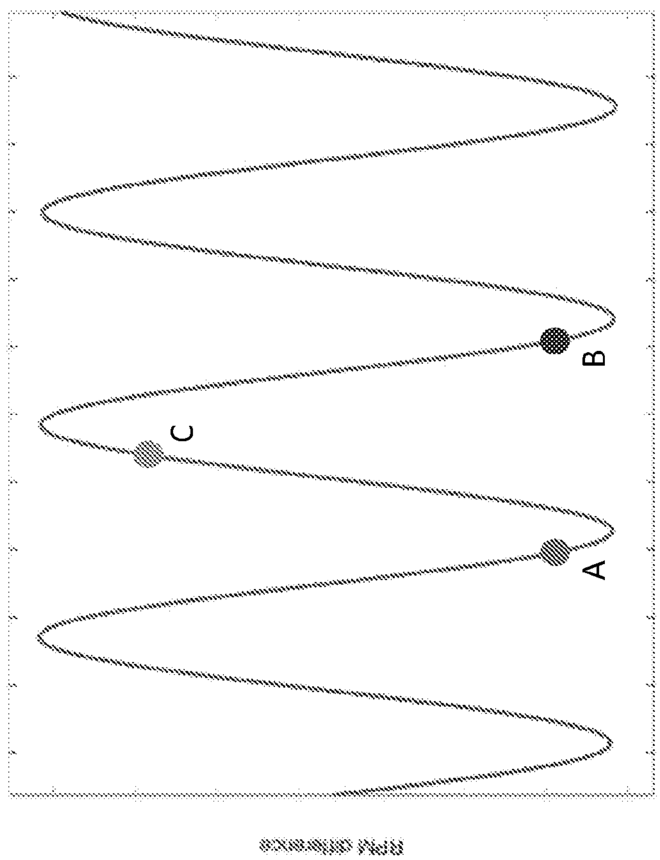


FIG. 15A

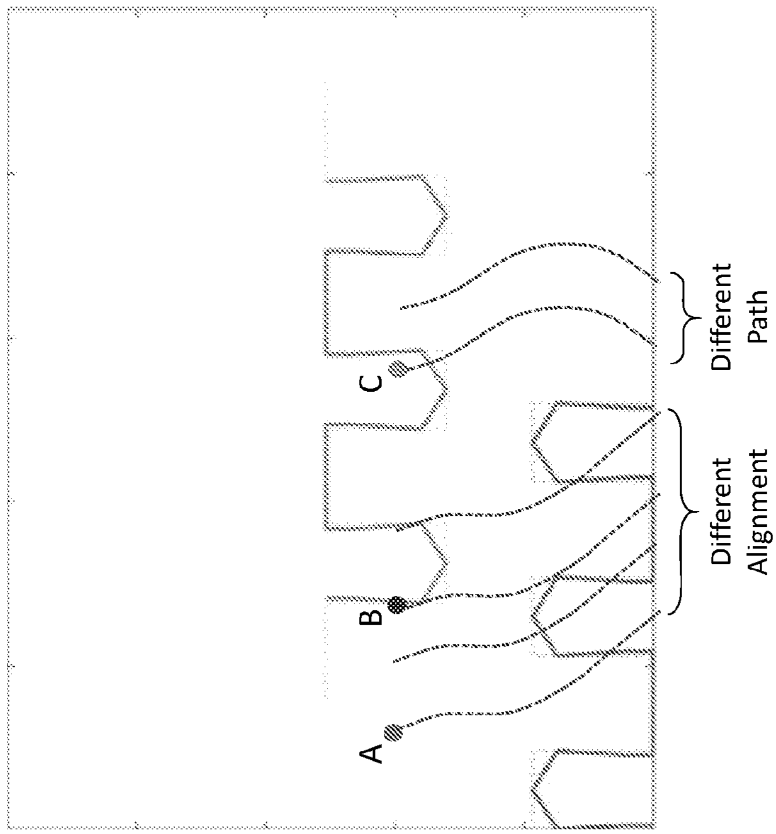


FIG. 15B

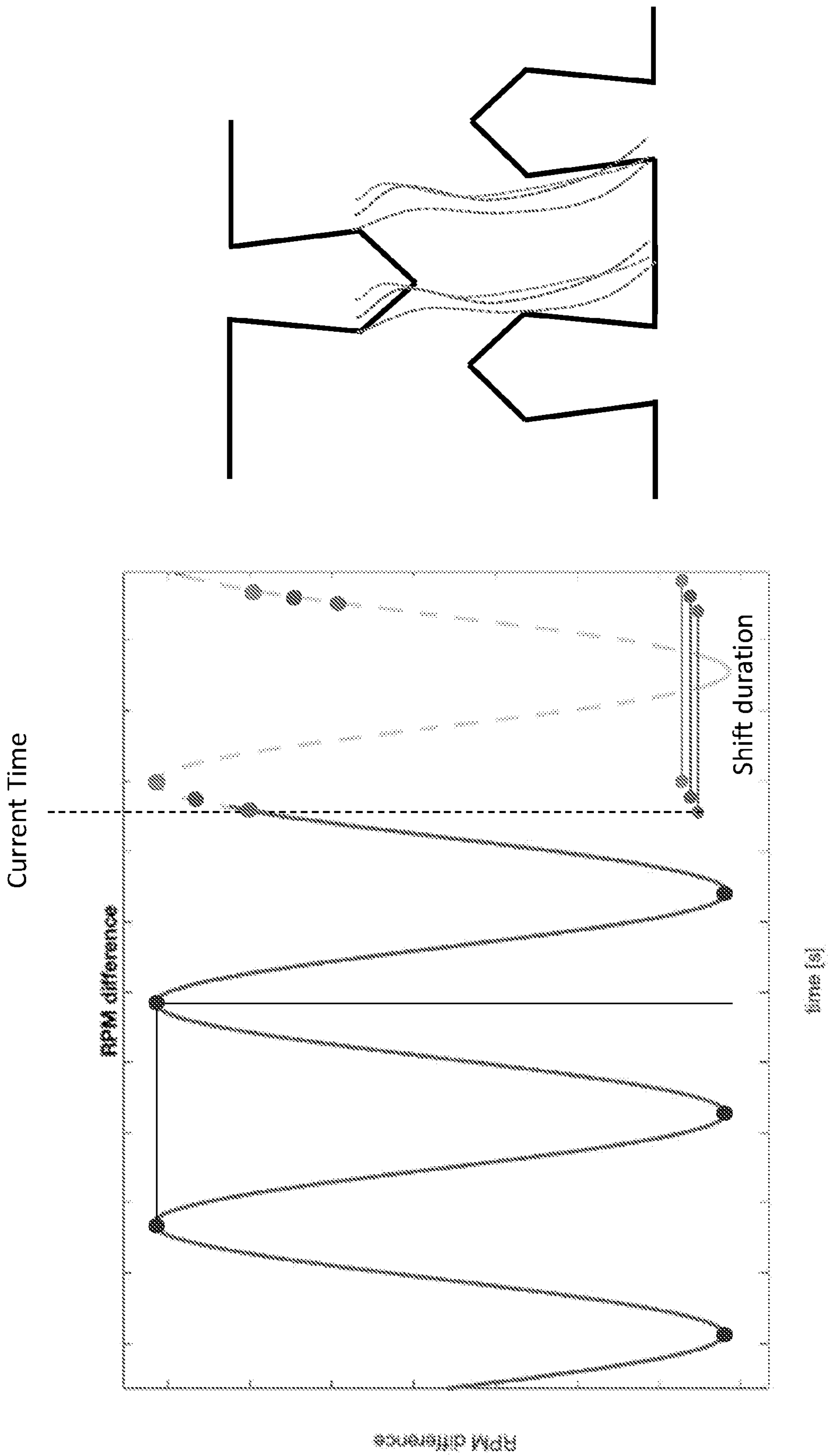

Current Time
RPM difference
Shift duration
time [s]
RPM difference

FIG. 16

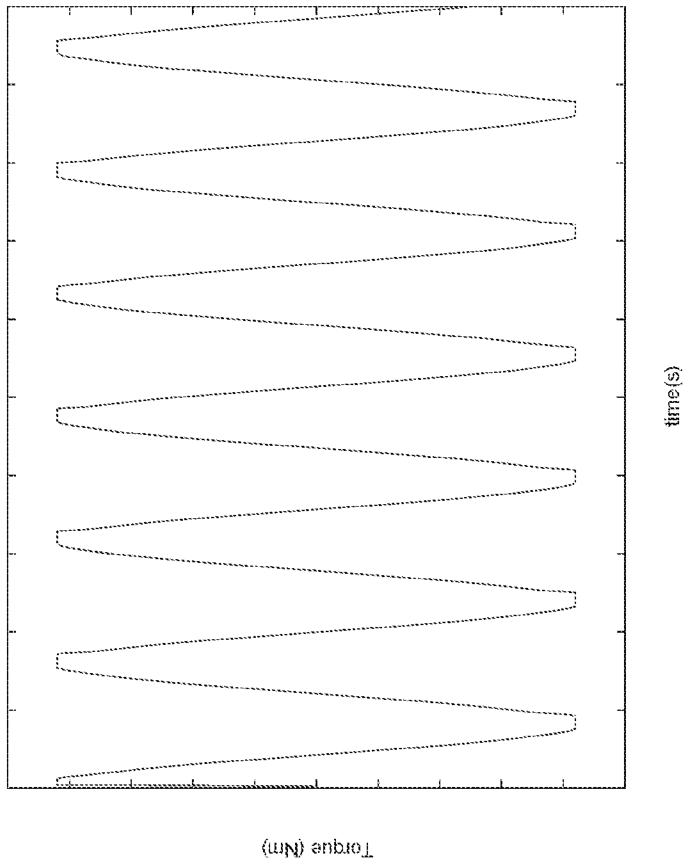


FIG. 17B

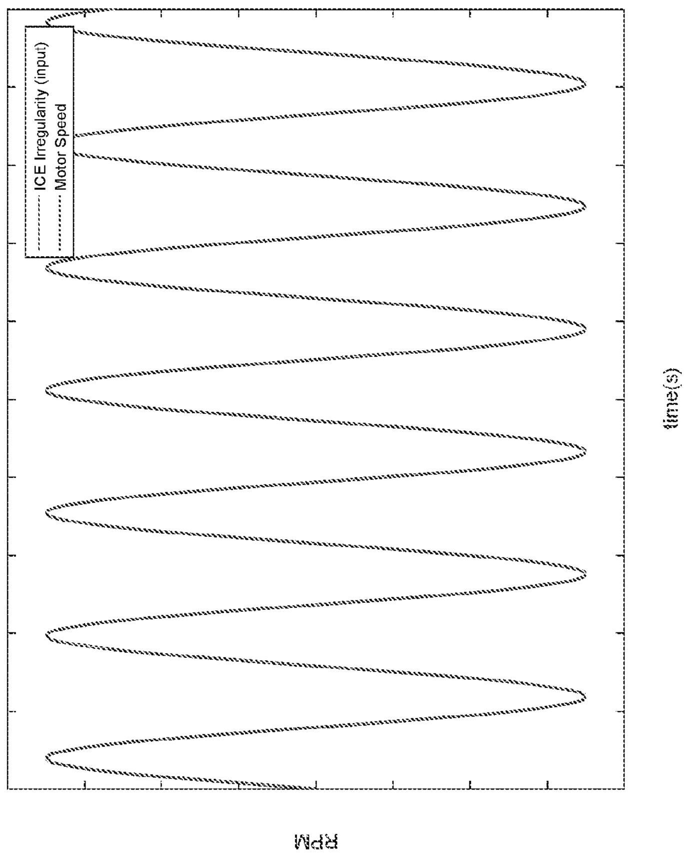


FIG. 17A

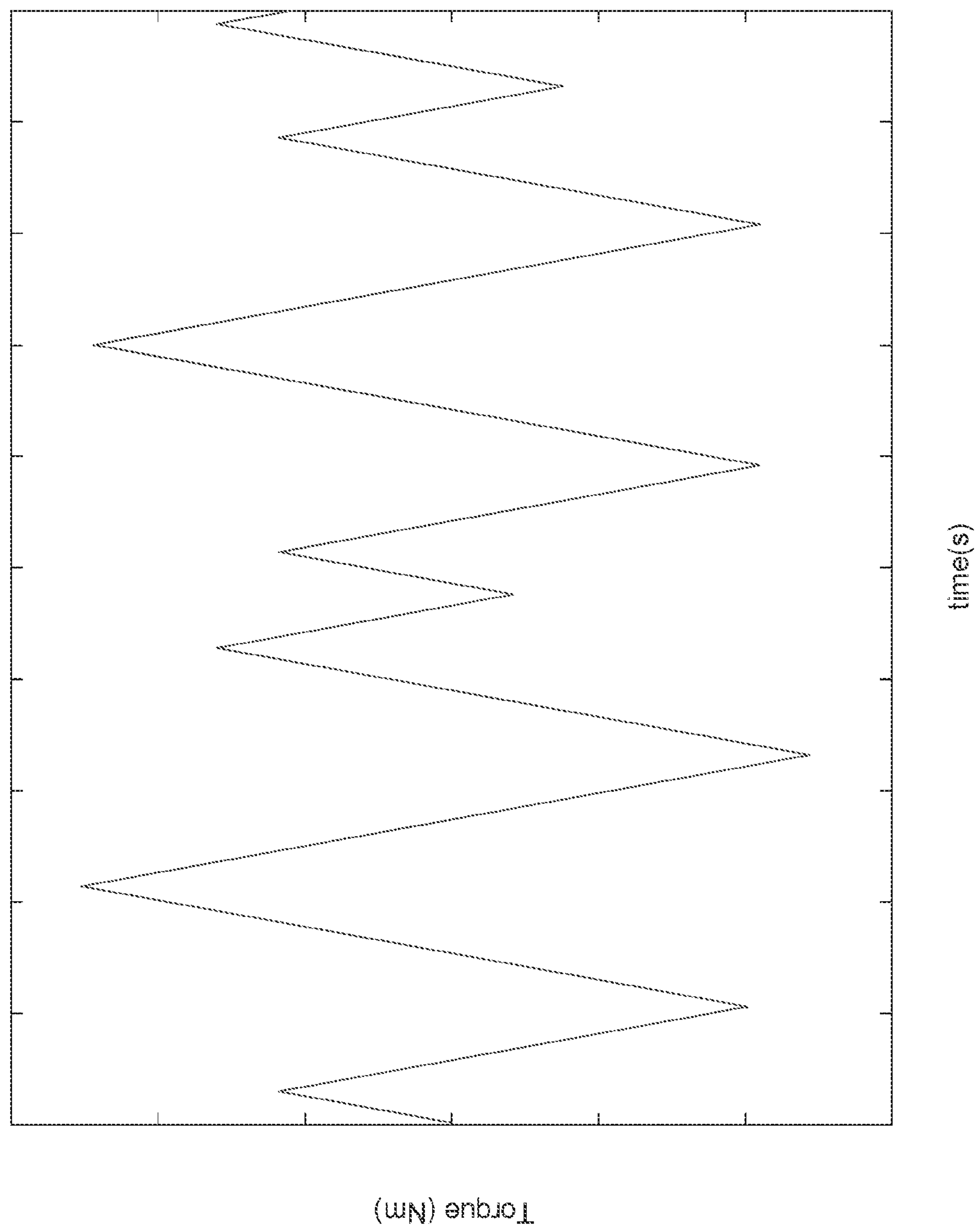


FIG. 18

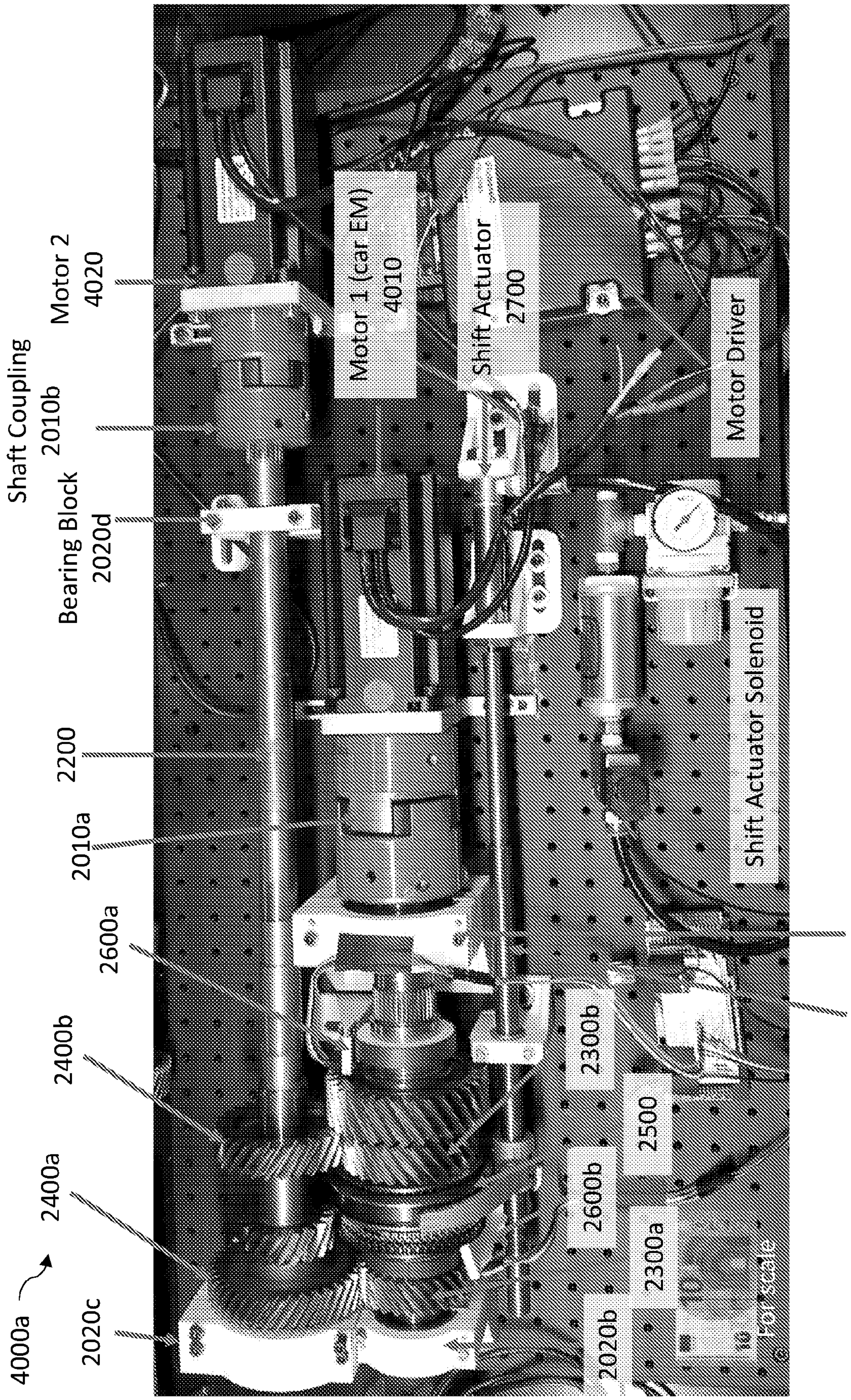

Motor 2
4020
Shaft Coupling
2010b
Bearing Block
2020d
2200
2010a
2600a
2400b
2400a
4000a
2020c
Motor 1 (car EM)
4010
Shift Actuator
2700
Motor Driver
Shift Actuator Solenoid
2020a
2100
2300b
2500
2600b
2300a
2020b
For scale

FIG. 19A

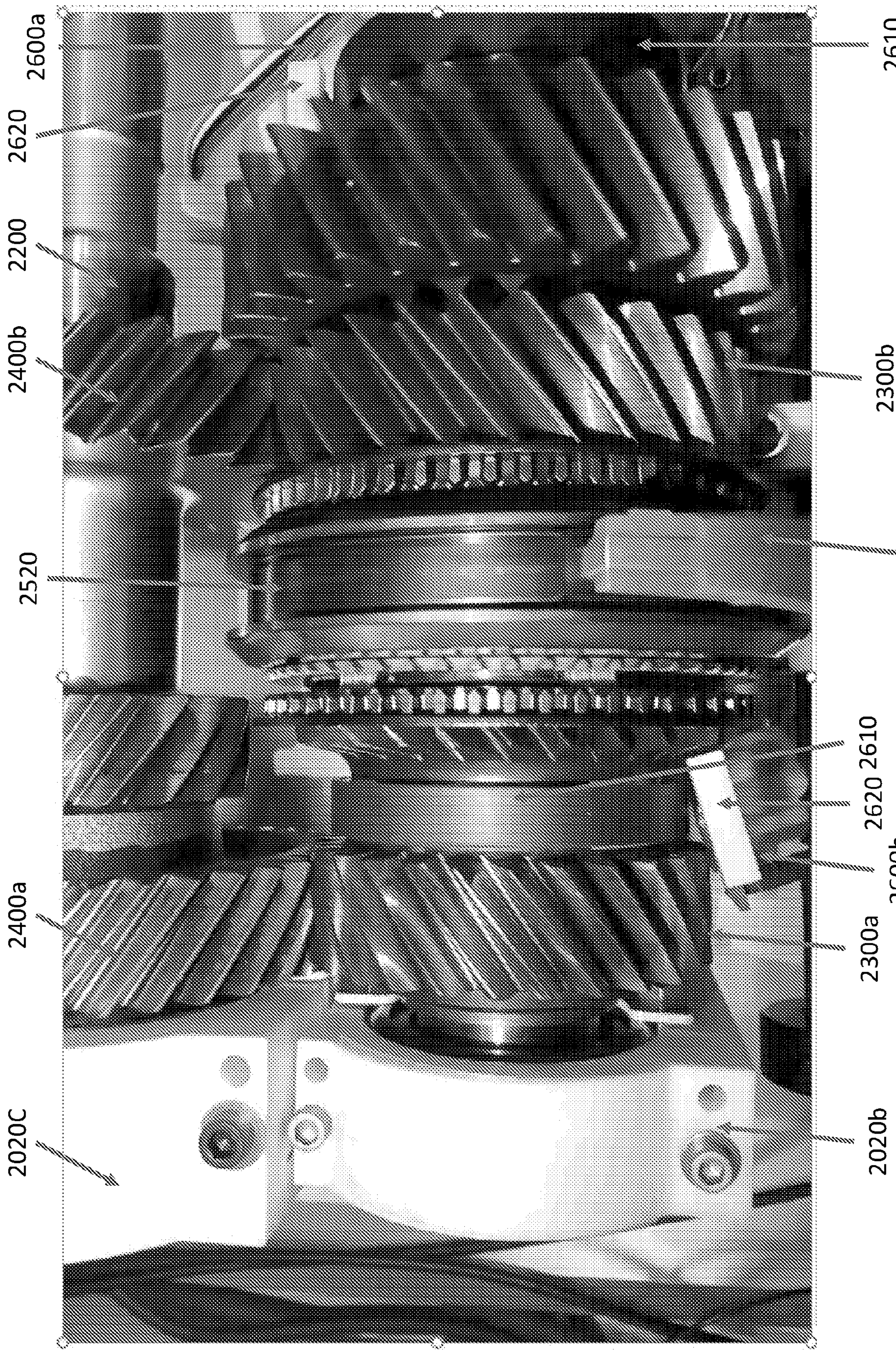

2600a
2620
2200
2400b
2520
2400a
2020C
2610
2300b
2500
2610
2620
2600b
2300a
2020b

FIG. 19B

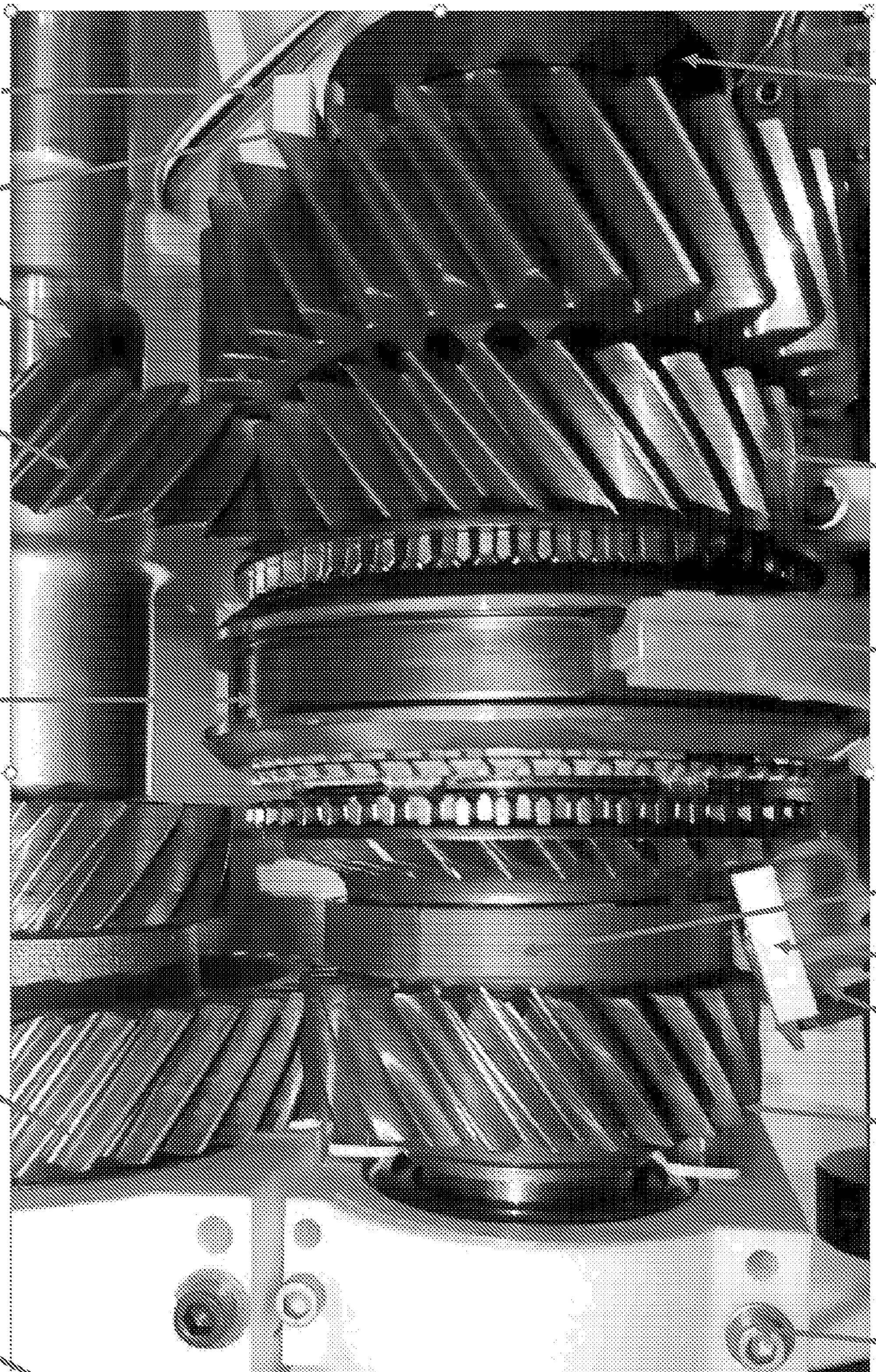

2600a
2620
2200
2400b
2520
2400a
2020C
2610
2300b
2500
2610
2620
2600b
2300a
2020b

FIG. 19C

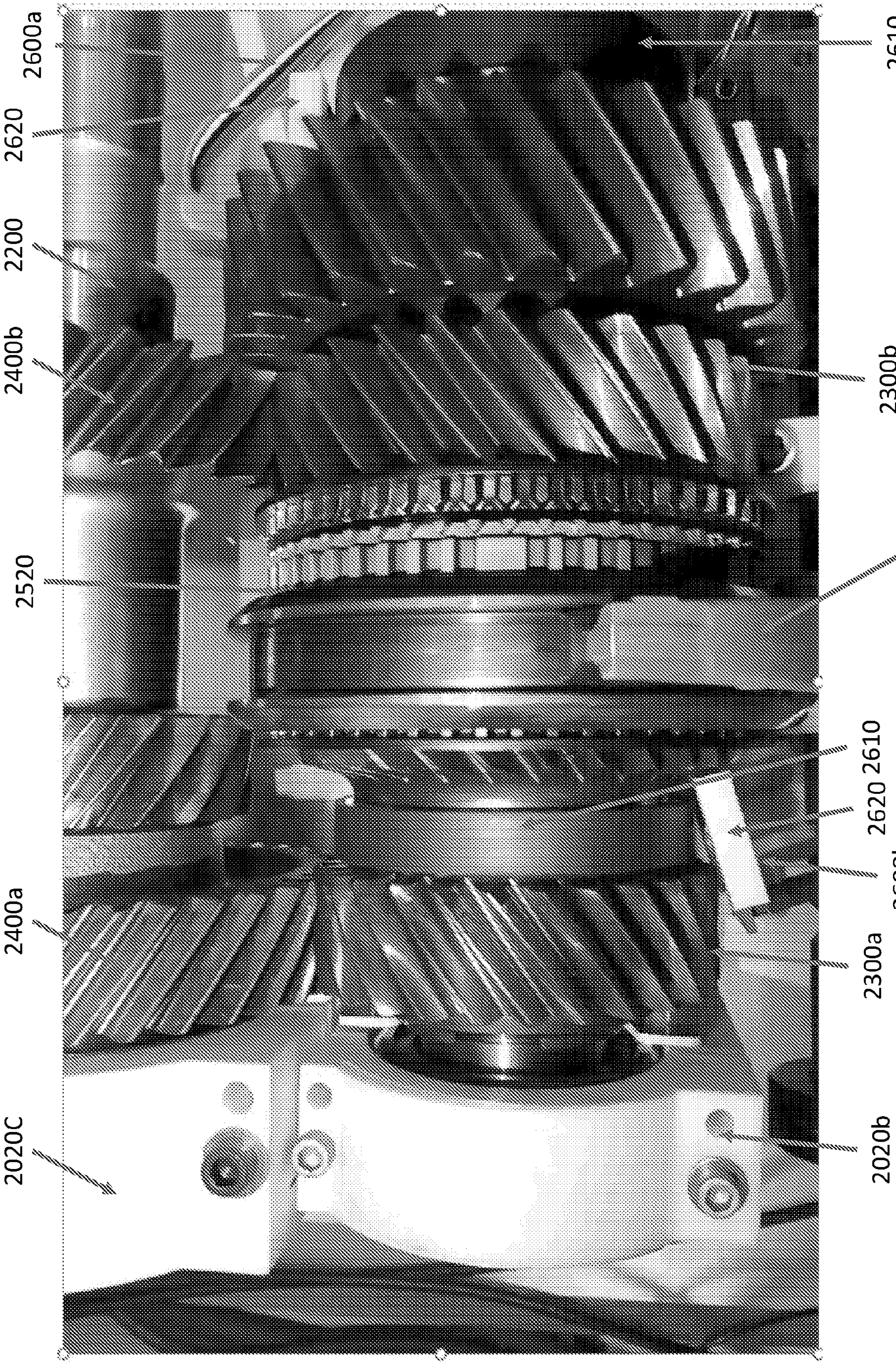

2600a
2620
2200
2400b
2520
2400a
2020C
2610
2300b
2500
2620 2610
2600b
2300a
2020b

FIG. 19D

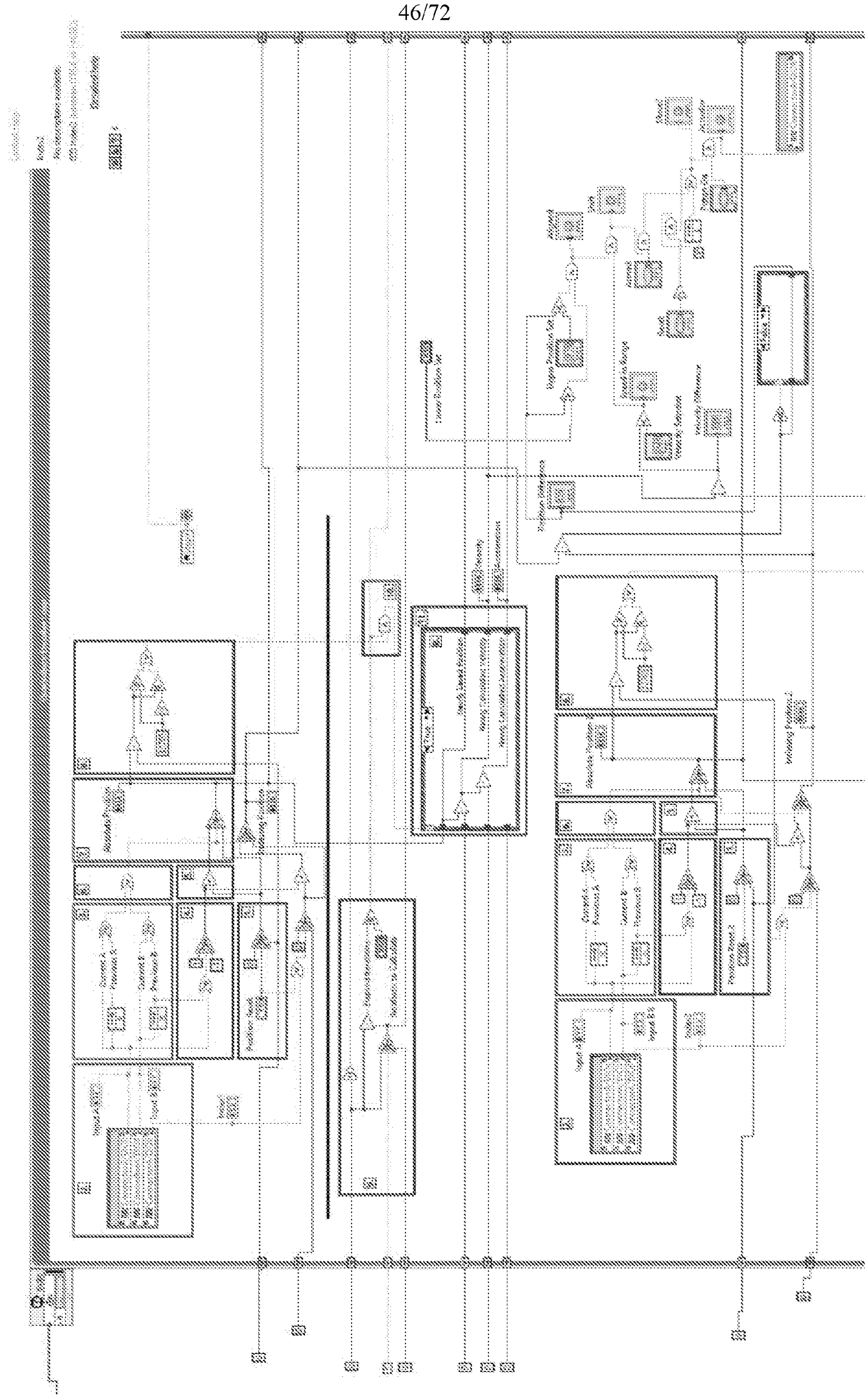


FIG. 19E

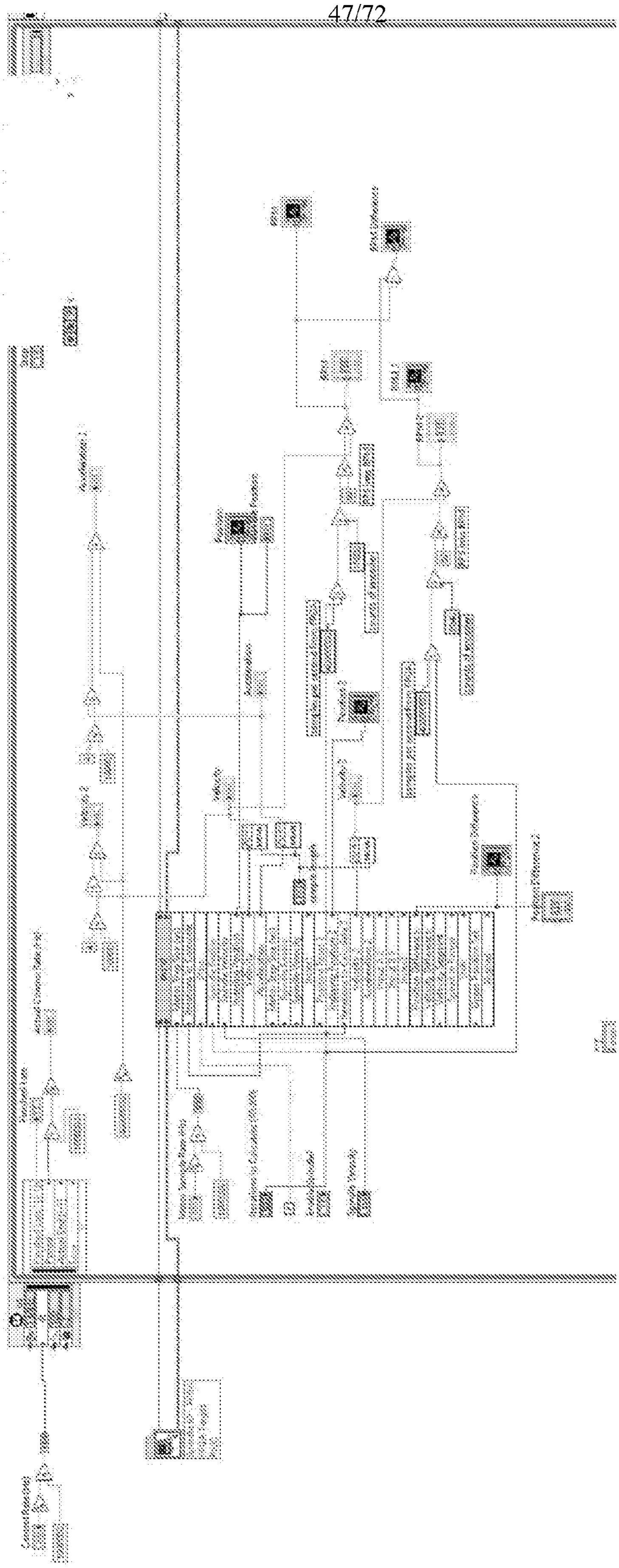


FIG. 19F

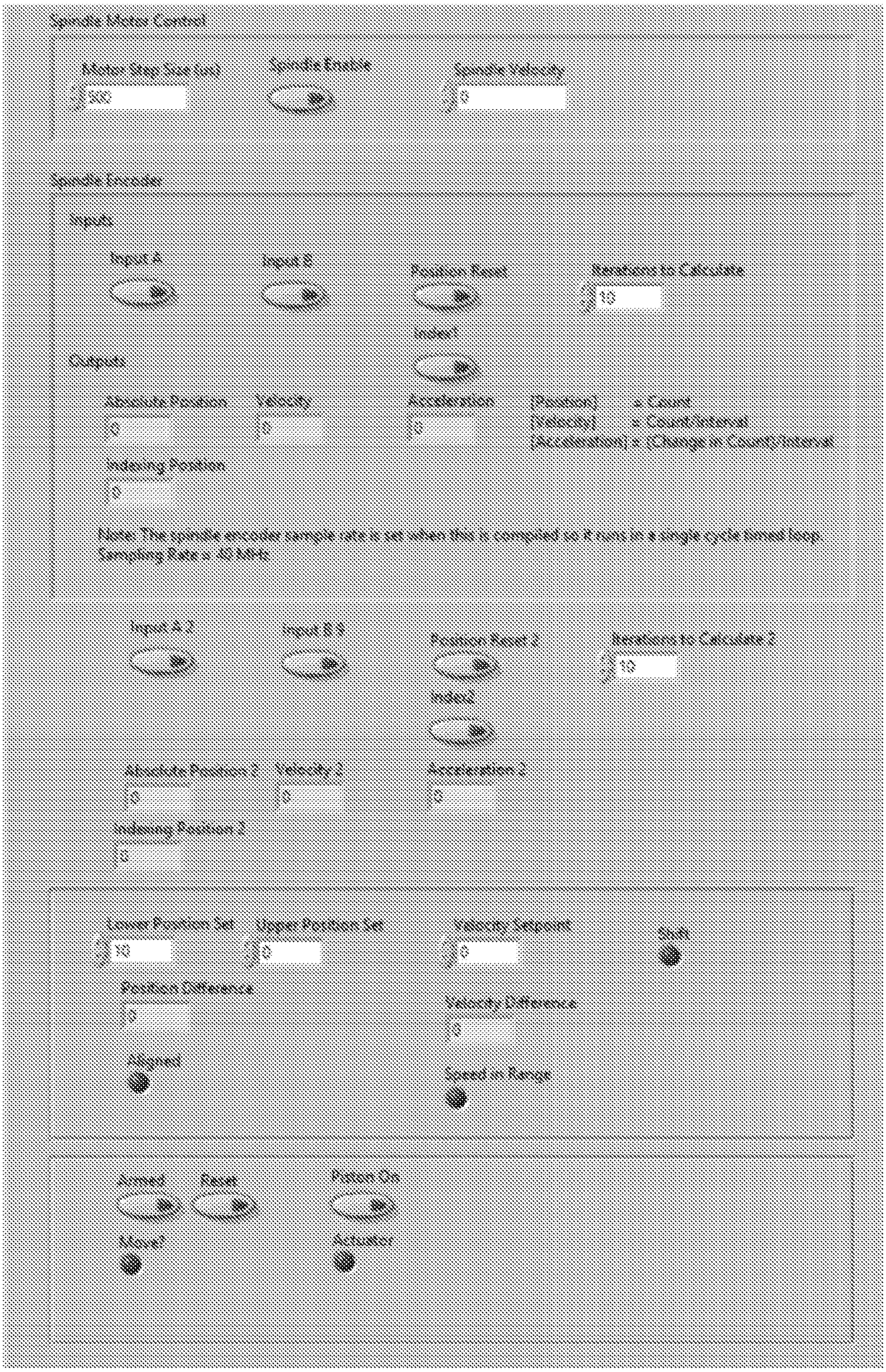

Spindle Motor Control
Motor Step Size (us)
Spindle Enable
Spindle Velocity
Spindle Encoder
Inputs
Input A
Input B
Position Reset
Iterations to Calculate
Index1
Outputs
Absolute Position
Velocity
Acceleration
[Position] = Count
[Velocity] = Count/Interval
[Acceleration] = (Change in Count)/Interval
Indexing Position
Note: The spindle encoder sample rate is set when this is compiled so it runs in a single cycle timed loop.
Sampling Rate = 40 MHz
Input A 2
Input B 2
Position Reset 2
Iterations to Calculate 2
Index2
Absolute Position 2
Velocity 2
Acceleration 2
Indexing Position 2
Lower Position Set
Upper Position Set
Velocity Setpoint
Shift
Position Difference
Velocity Difference
Aligned
Speed in Range
Armed
Reset
Piston On
Move?
Actuator

FIG. 19G

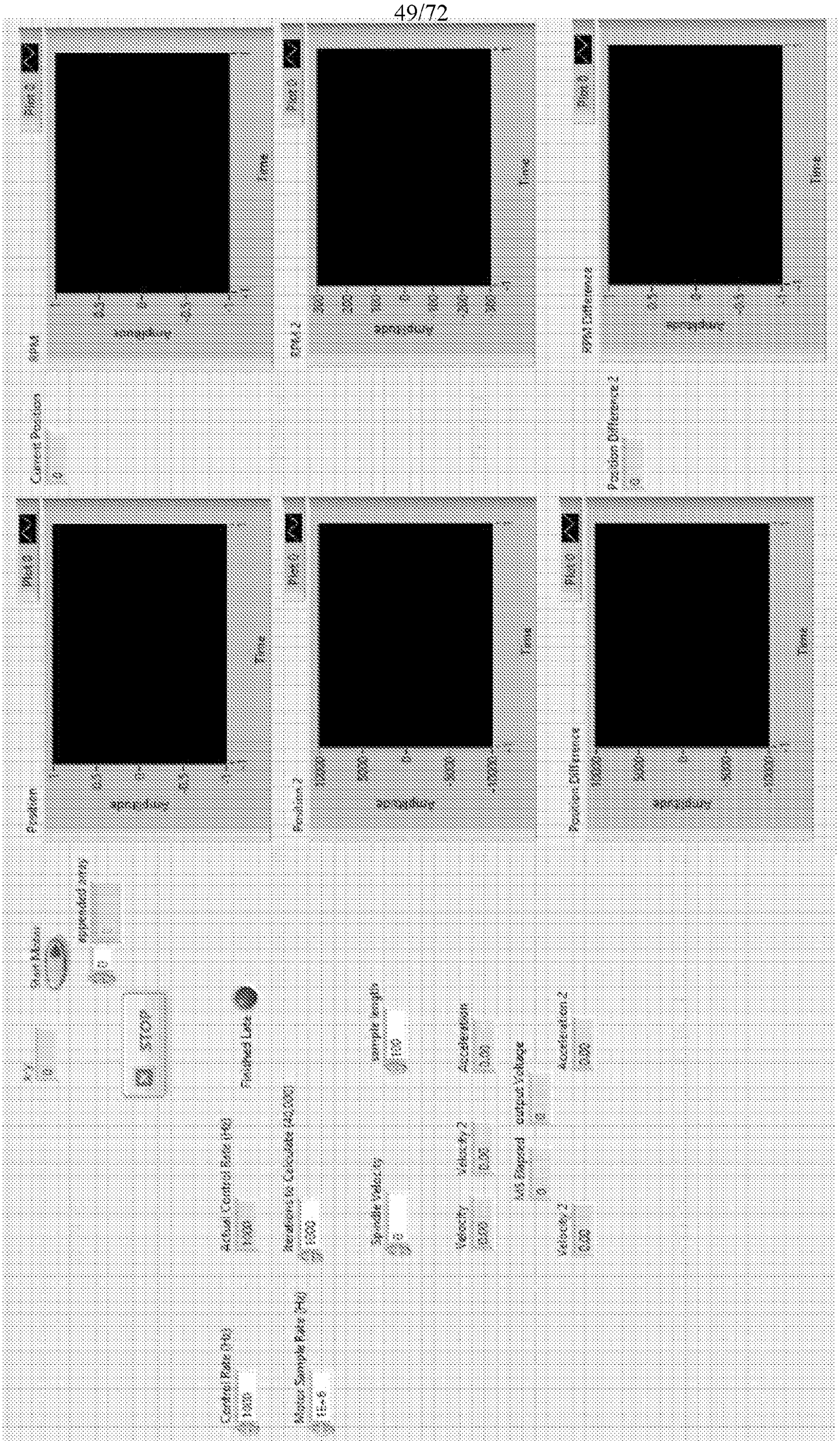


FIG. 19H

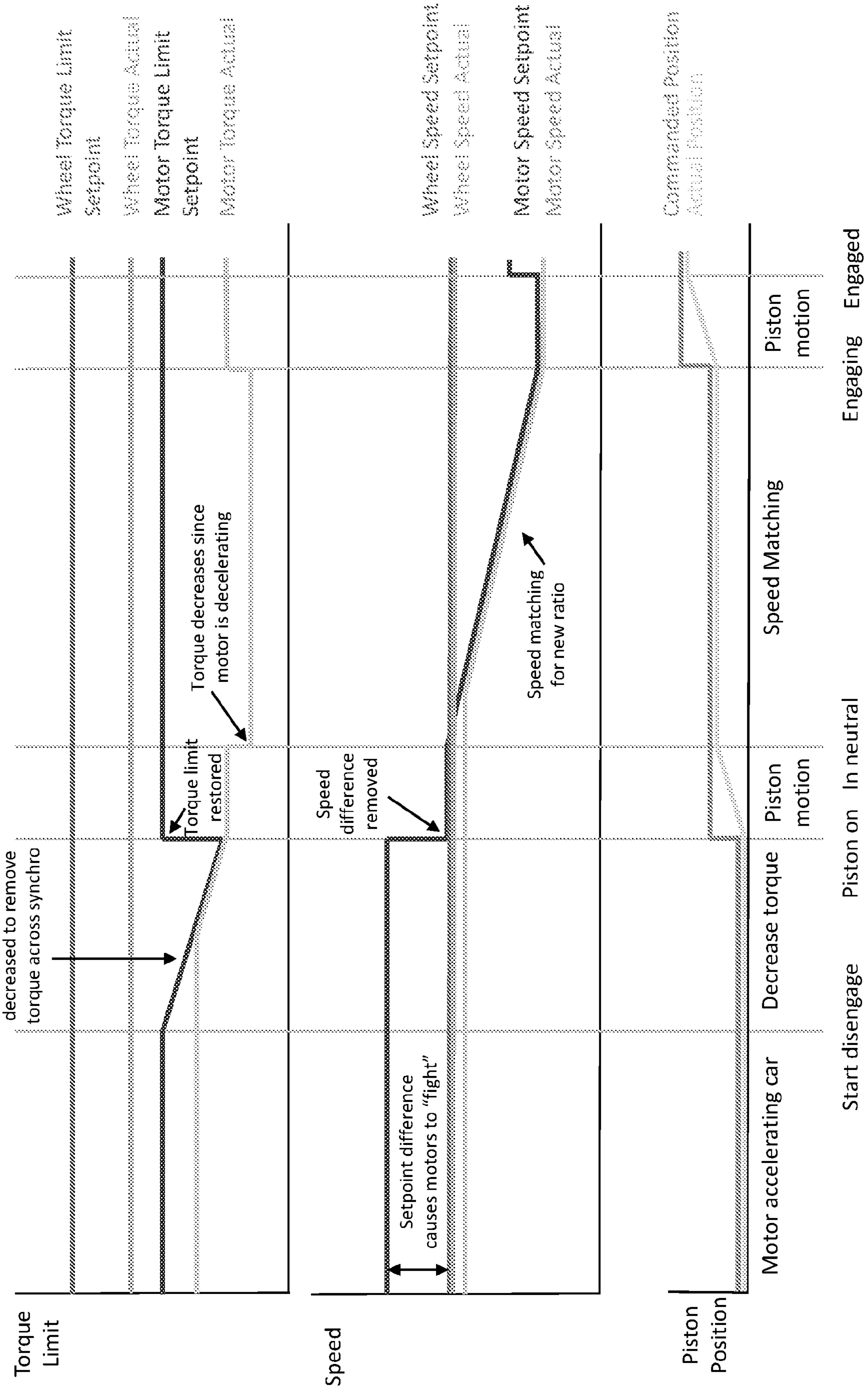

Torque Limit
Speed
Piston Position
Wheel Torque Limit Setpoint
Wheel Torque Actual
Motor Torque Limit Setpoint
Motor Torque Actual
Wheel Speed Setpoint
Wheel Speed Actual
Motor Speed Setpoint
Motor Speed Actual
Commanded Position
Actual Position
Torque limit decreased to remove torque across synchro
Torque limit restored
Torque decreases since motor is decelerating
Setpoint difference causes motors to "fight"
Speed difference removed
Speed matching for new ratio
Motor accelerating car
Start disengage
Decrease torque
Piston on
Piston motion
In neutral position
Speed Matching
Engaging
Piston motion
Engaged

FIG. 20A

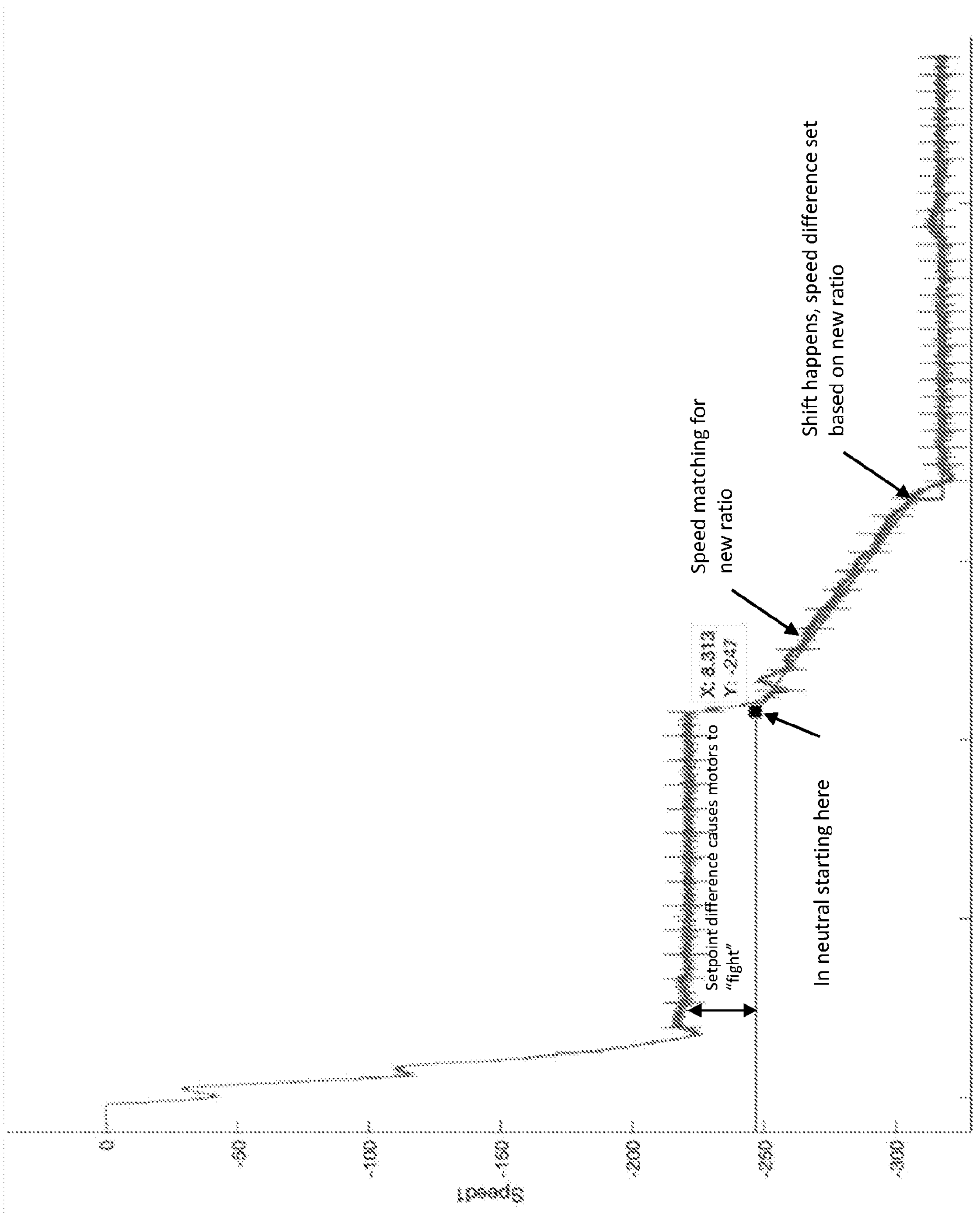

Speed
0
-50
-100
-150
-200
-250
-300
4
6
8
10
12
14
X: 8.313
Y: -247
Setpoint difference causes motors to "fight"
In neutral starting here
Speed matching for new ratio
Shift happens, speed difference set based on new ratio

FIG. 20B

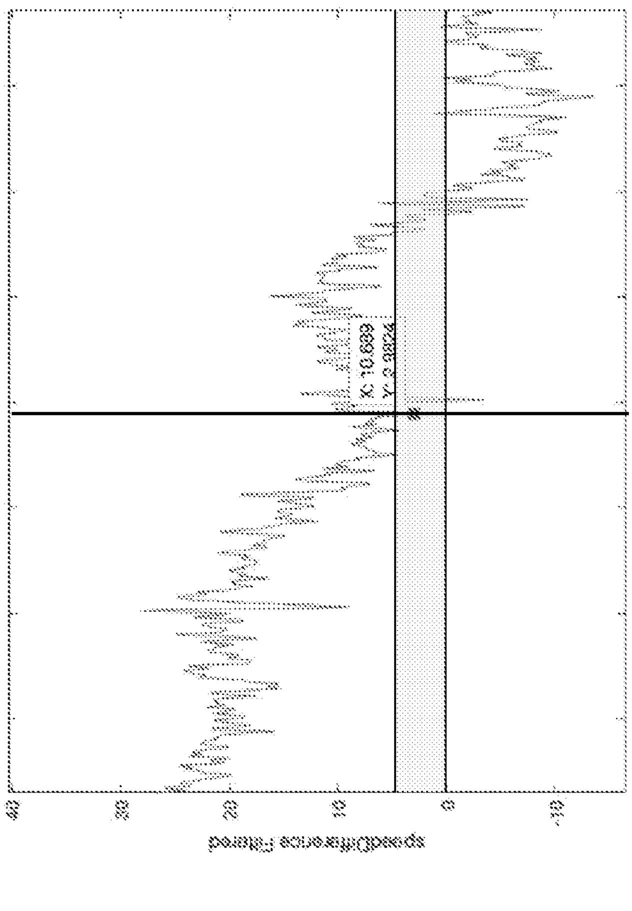


FIG. 21A

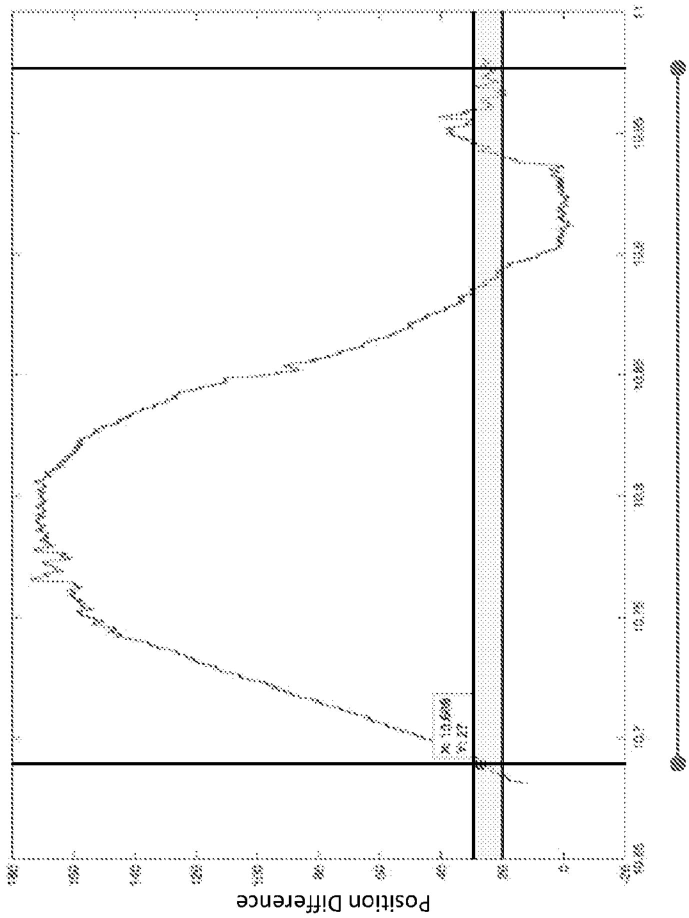


FIG. 21B

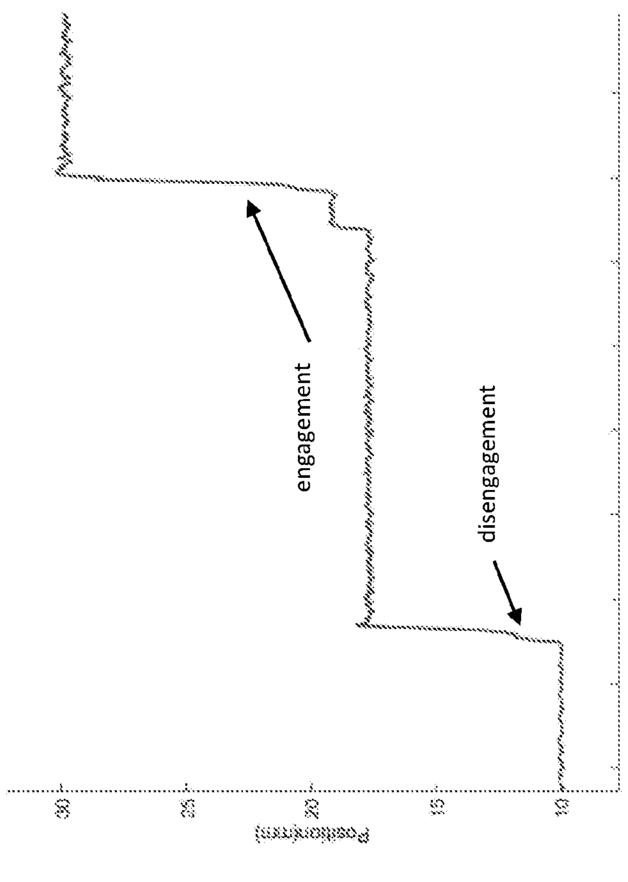


FIG. 22A

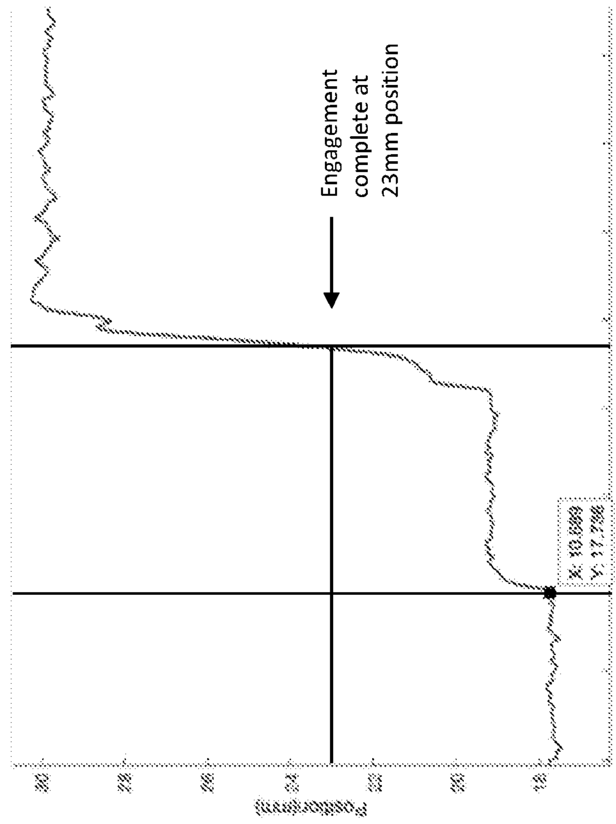


FIG. 22B

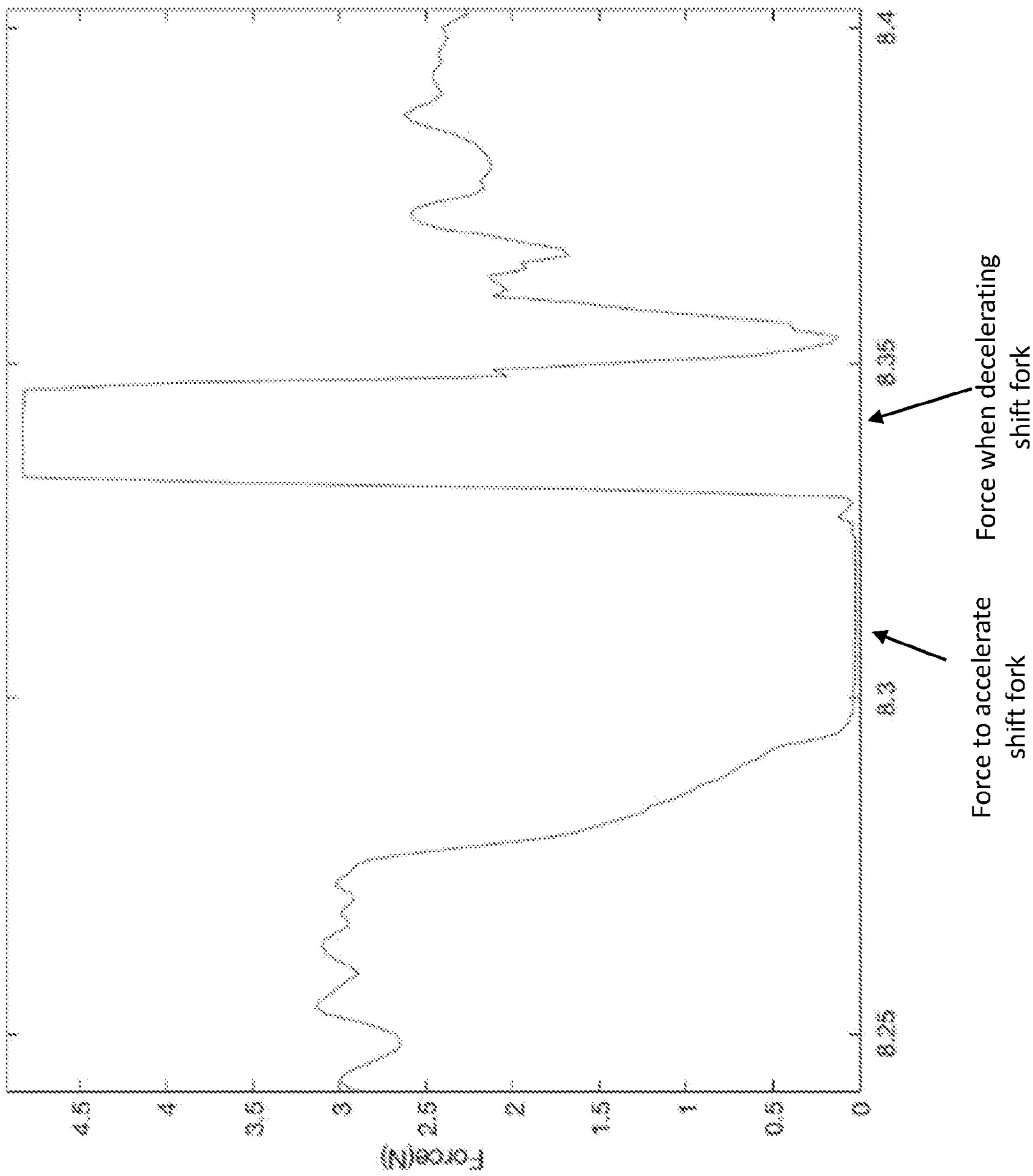

Force(N)
0
0.5
1
1.5
2
2.5
3
3.5
4
4.5
8.25
8.3
8.35
8.4
Force to accelerate shift fork
Force when decelerating shift fork

FIG. 22C

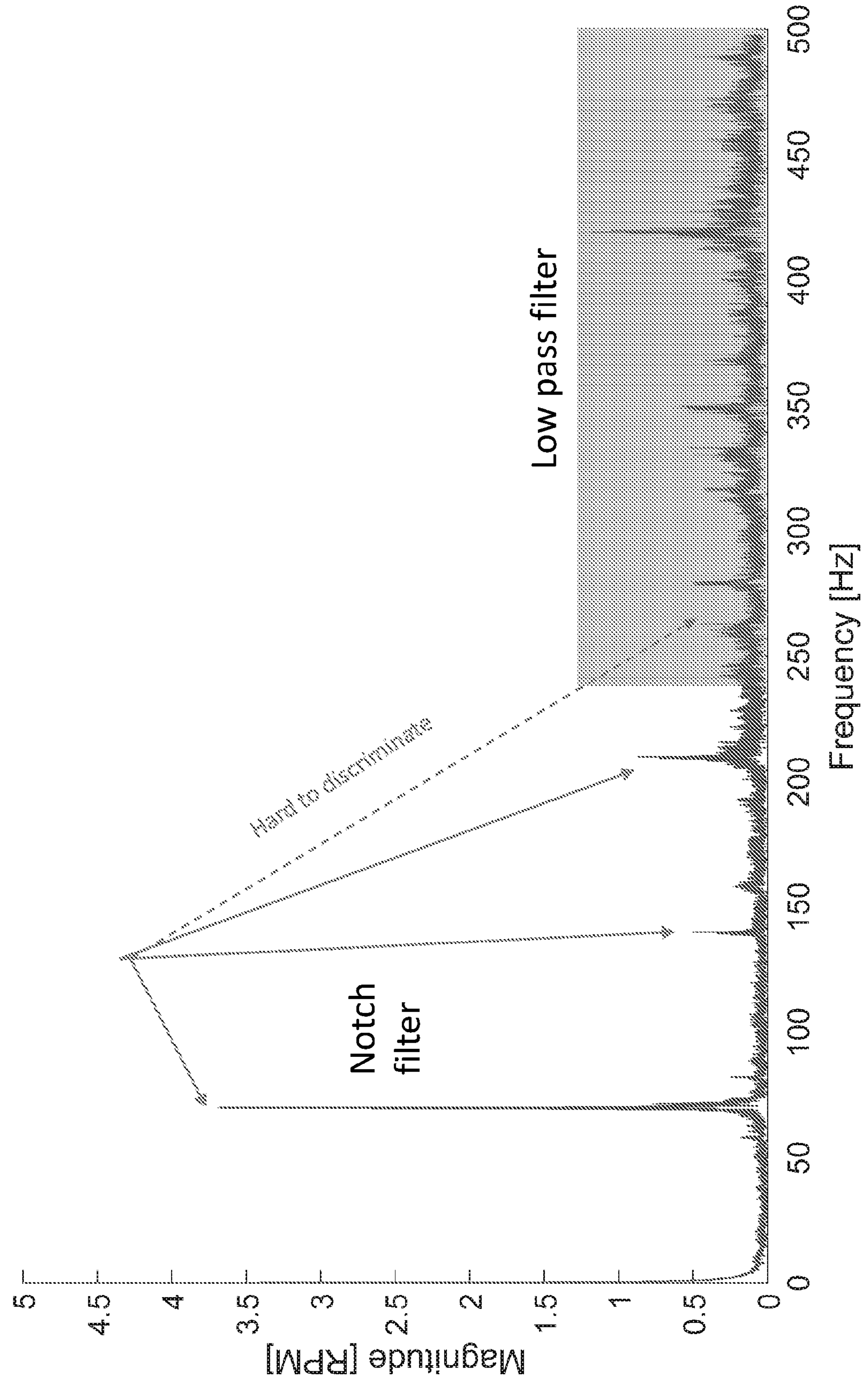

Low pass filter
Notch filter
Hard to discriminate
Frequency [Hz]
0
50
100
150
200
250
300
350
400
450
500
Magnitude [RPM]
0
0.5
1
1.5
2
2.5
3
3.5
4
4.5
5

FIG. 23A

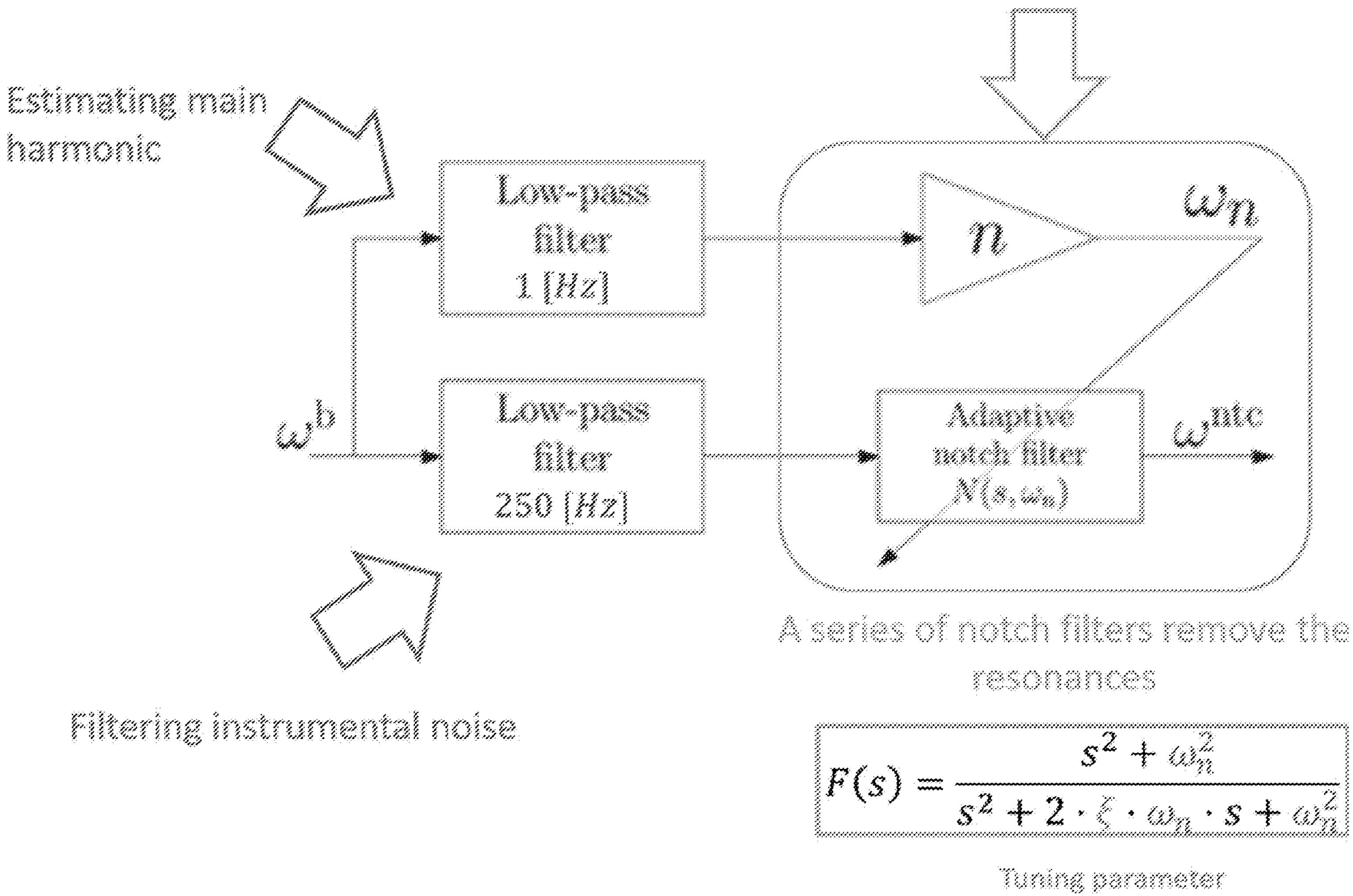

Estimating main harmonic
Low-pass filter 1 [Hz]
Low-pass filter 250 [Hz]
ω^b
n
ω_n
Adaptive notch filter $N(s, \omega_n)$
ω^{ntc}
Filtering instrumental noise
A series of notch filters remove the resonances
$F(s) = \frac{s^2 + \omega_n^2}{s^2 + 2 \cdot \xi \cdot \omega_n \cdot s + \omega_n^2}$
Tuning parameter

FIG. 23B

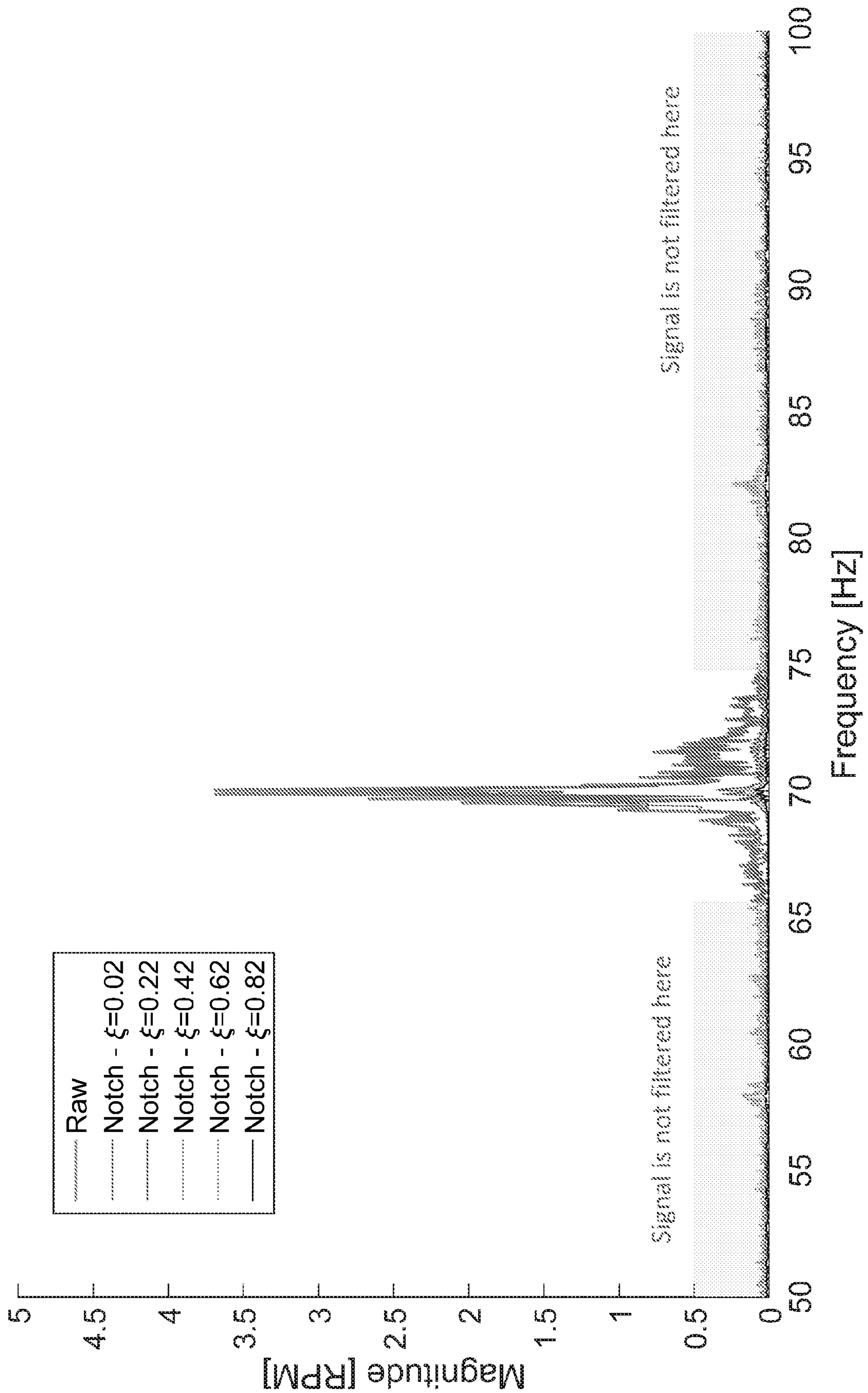

Magnitude [RPM]
5
4.5
4
3.5
3
2.5
2
1.5
1
0.5
0
Frequency [Hz]
50
55
60
65
70
75
80
85
90
95
100
Raw
Notch - ξ=0.02
Notch - ξ=0.22
Notch - ξ=0.42
Notch - ξ=0.62
Notch - ξ=0.82
Signal is not filtered here
Signal is not filtered here

FIG. 23C

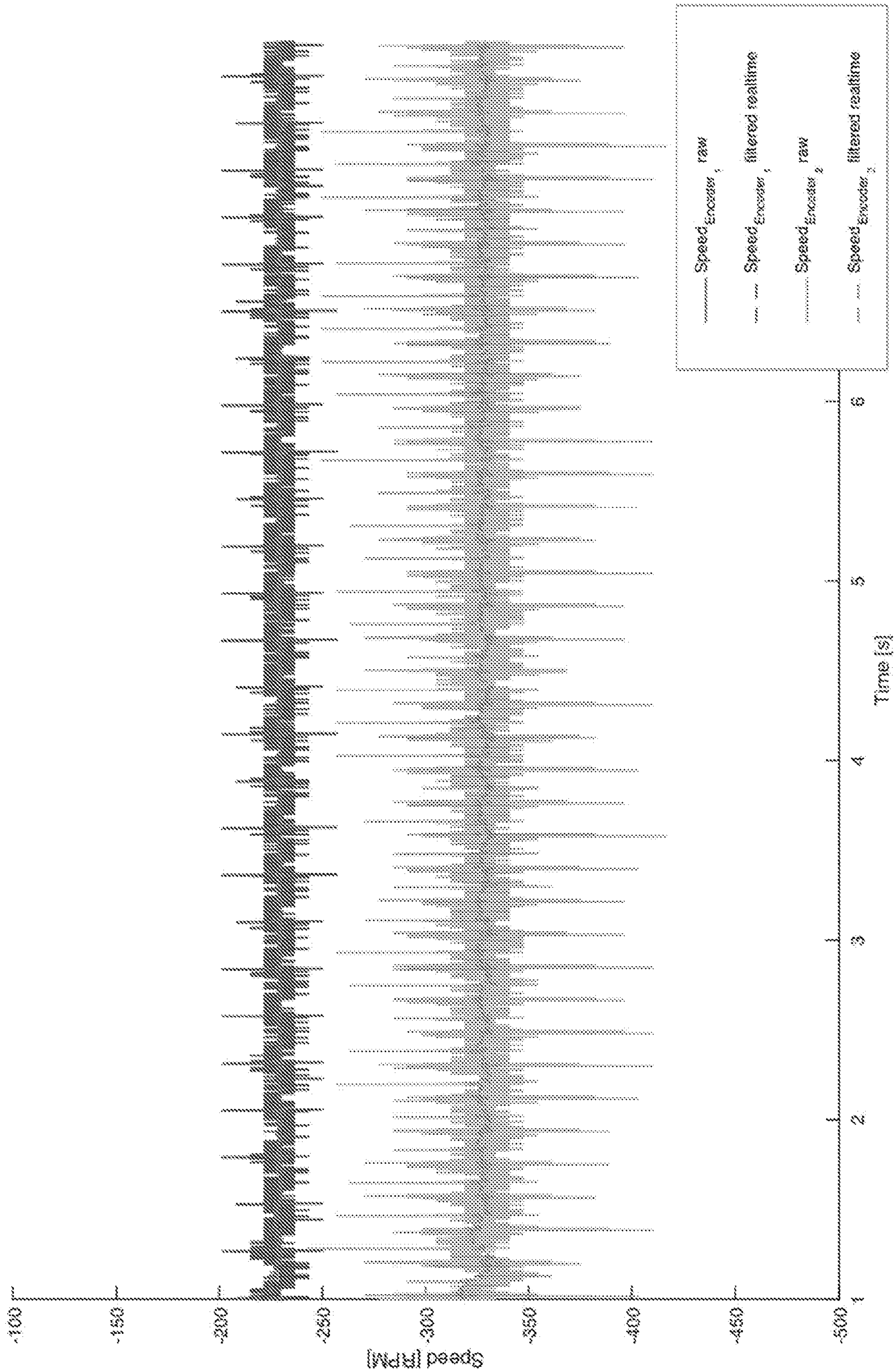

Speed [RPM]
-100
-150
-200
-250
-300
-350
-400
-450
-500
Time [s]
1
2
3
4
5
6
Speed Encoder 1 raw
Speed Encoder 1 filtered realtime
Speed Encoder 2 raw
Speed Encoder 2 filtered realtime

FIG. 23D

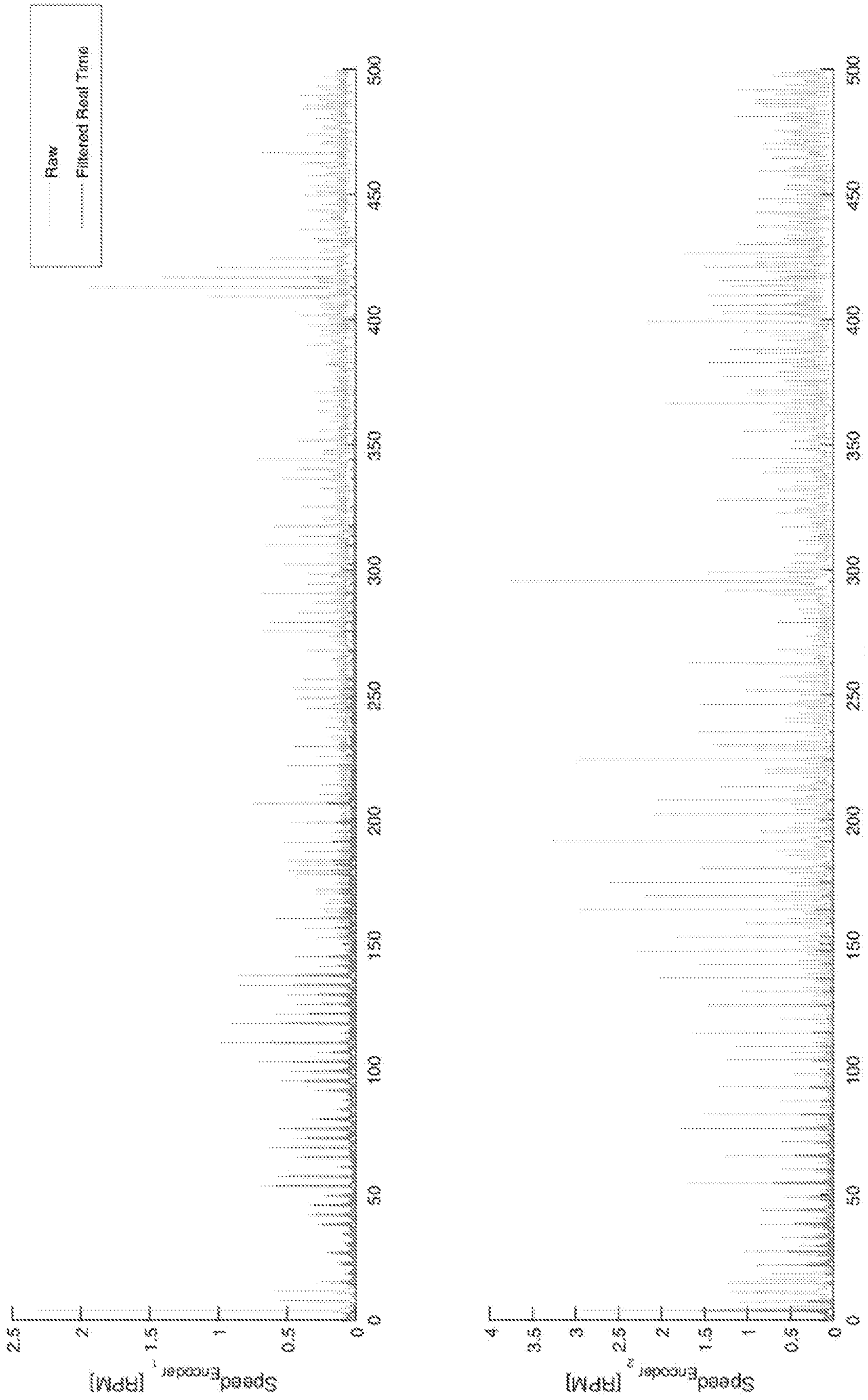

Raw
Filtered Real Time
Speed$_{Encoder_1}$ [RPM]
2.5
2
1.5
1
0.5
0
Speed$_{Encoder_2}$ [RPM]
4
3.5
3
2.5
2
1.5
1
0.5
0
0
50
100
150
200
250
300
350
400
450
500
Frequency [Hz]

FIG. 23E

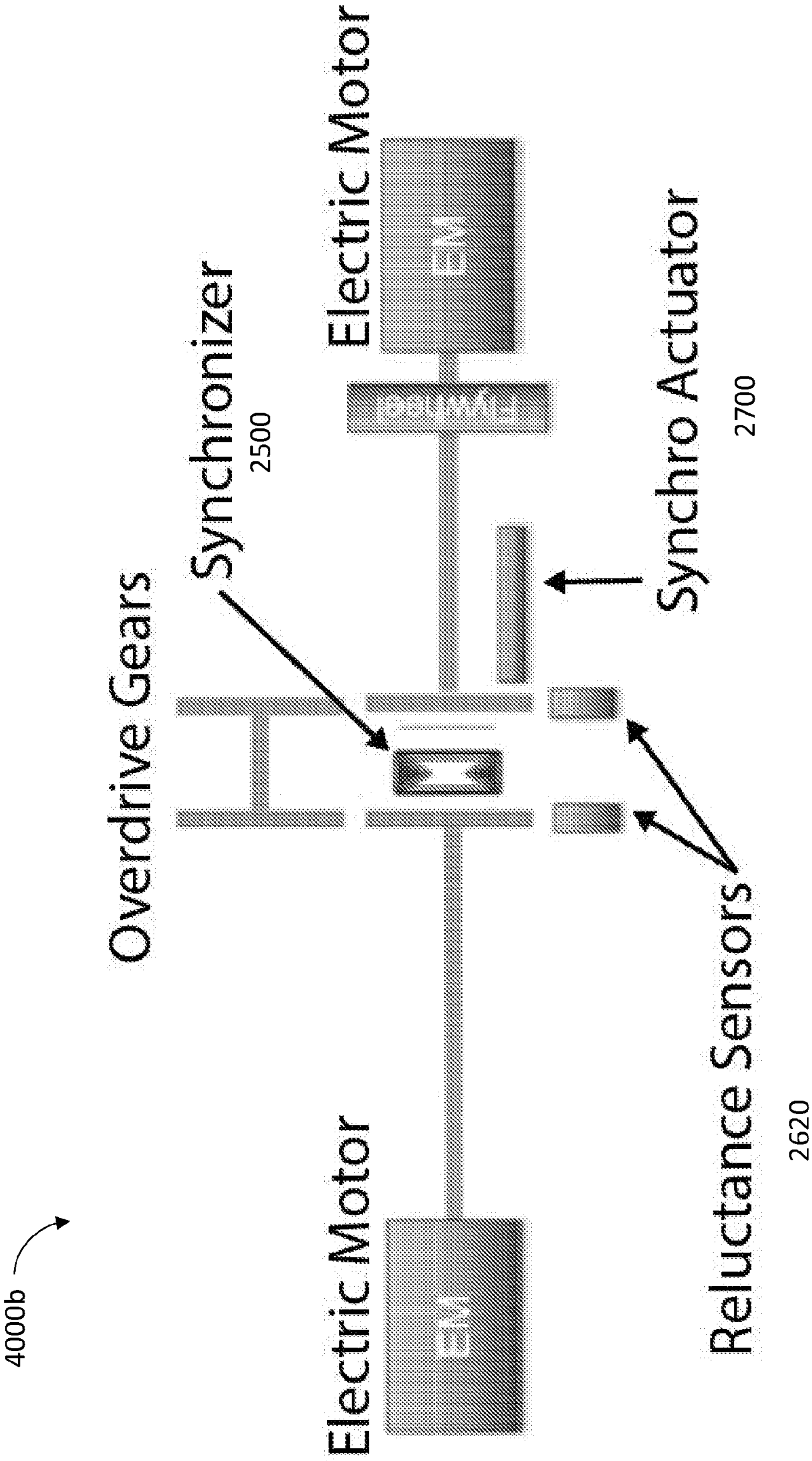

4000b
Electric Motor
EM
Overdrive Gears
Synchronizer
2500
Electric Motor
EM
Synchro Actuator
2700
Reluctance Sensors
2620

FIG. 24A

4000b

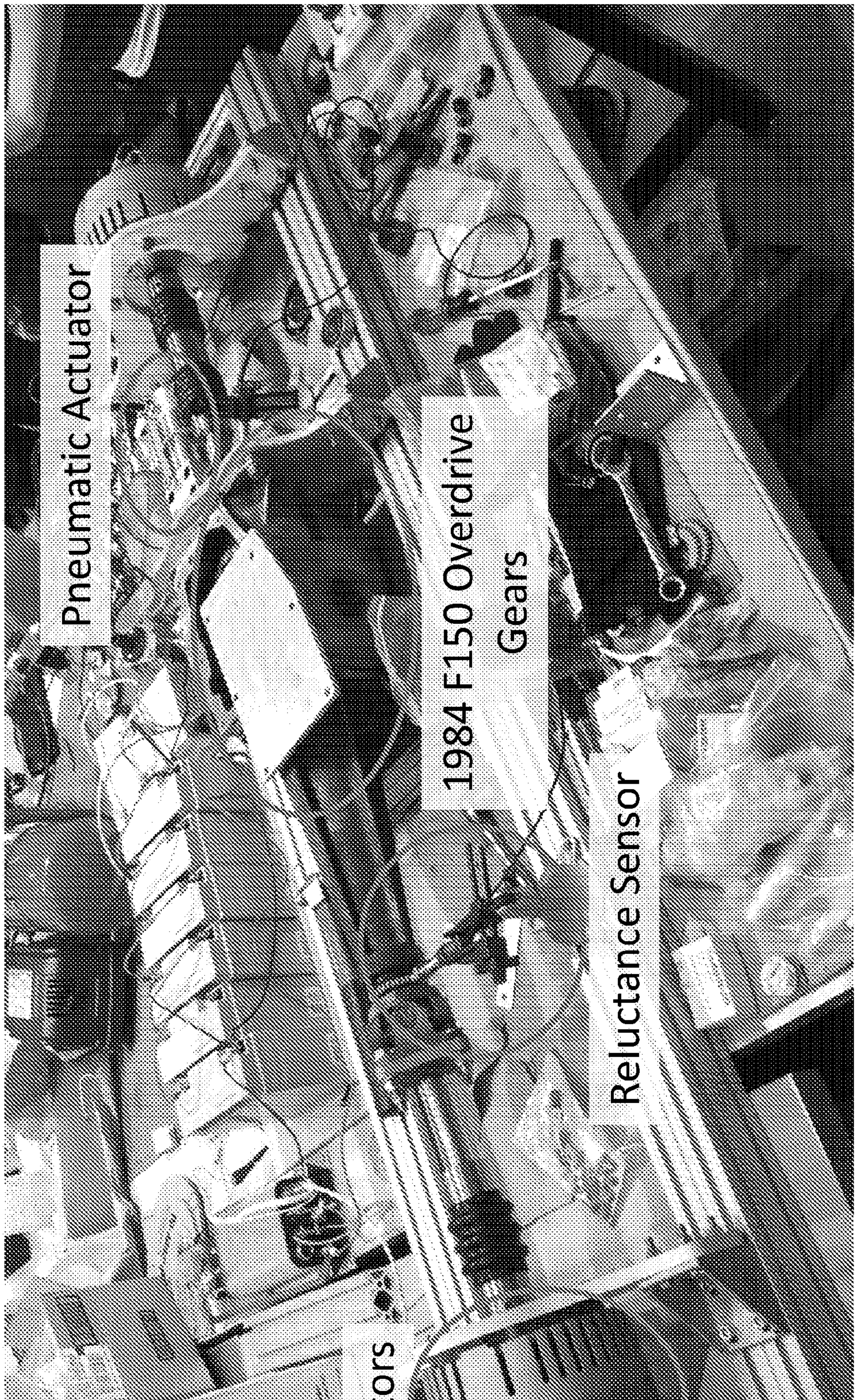


FIG. 24B

FIG. 24C

FIG. 24D

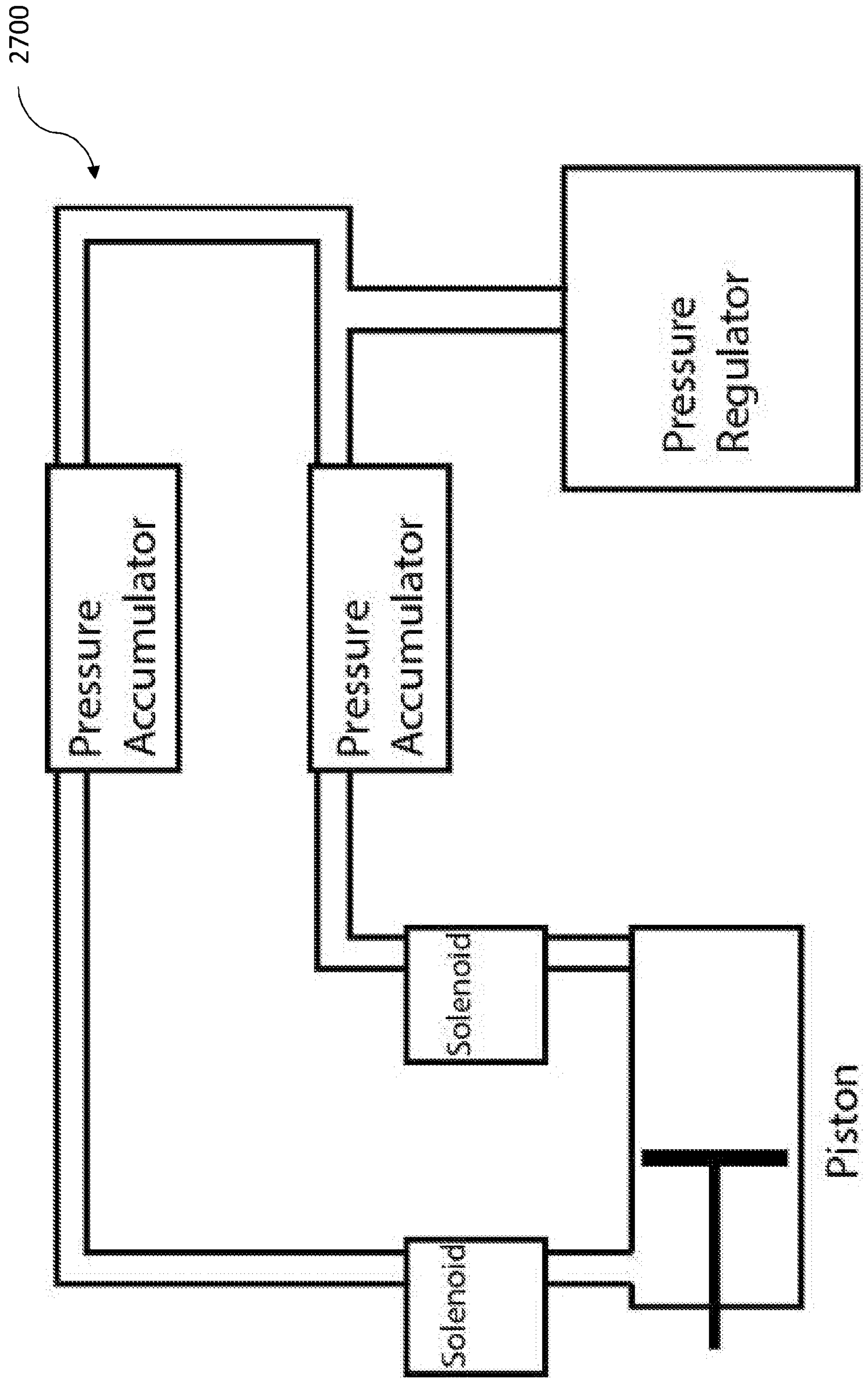

2700
Pressure Accumulator
Pressure Accumulator
Pressure Regulator
Solenoid
Solenoid
Piston

FIG. 24E

2700

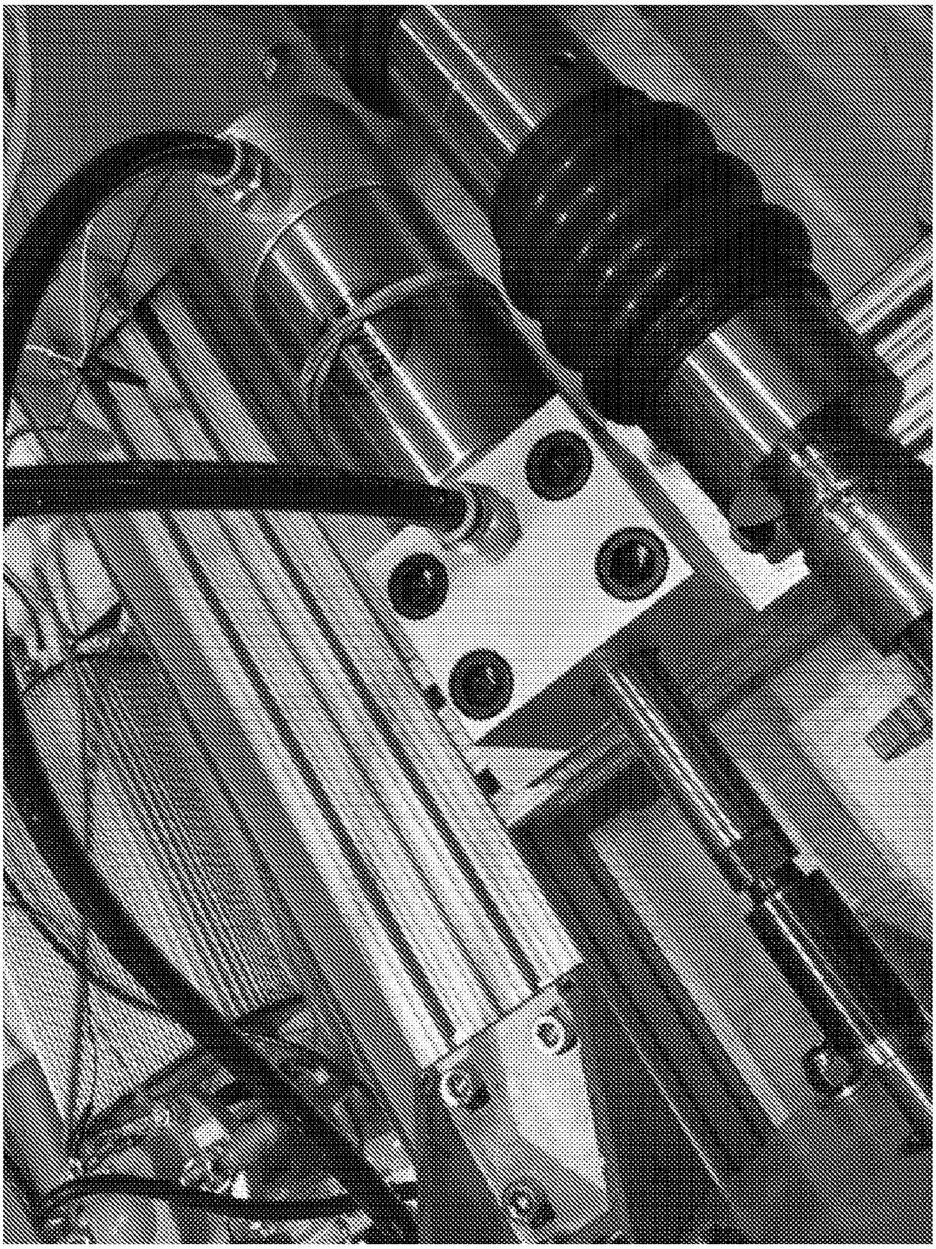


FIG. 24F

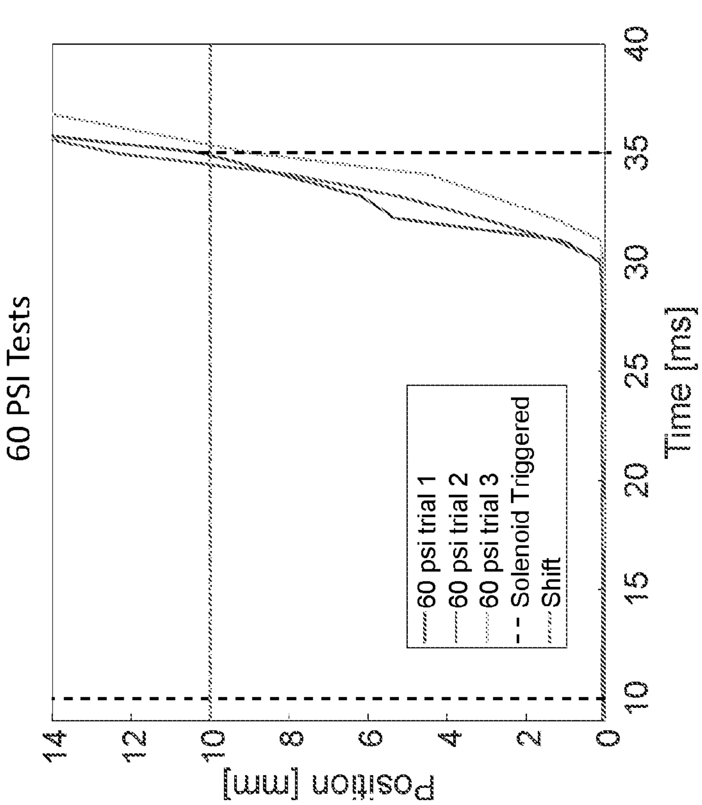


FIG. 25B

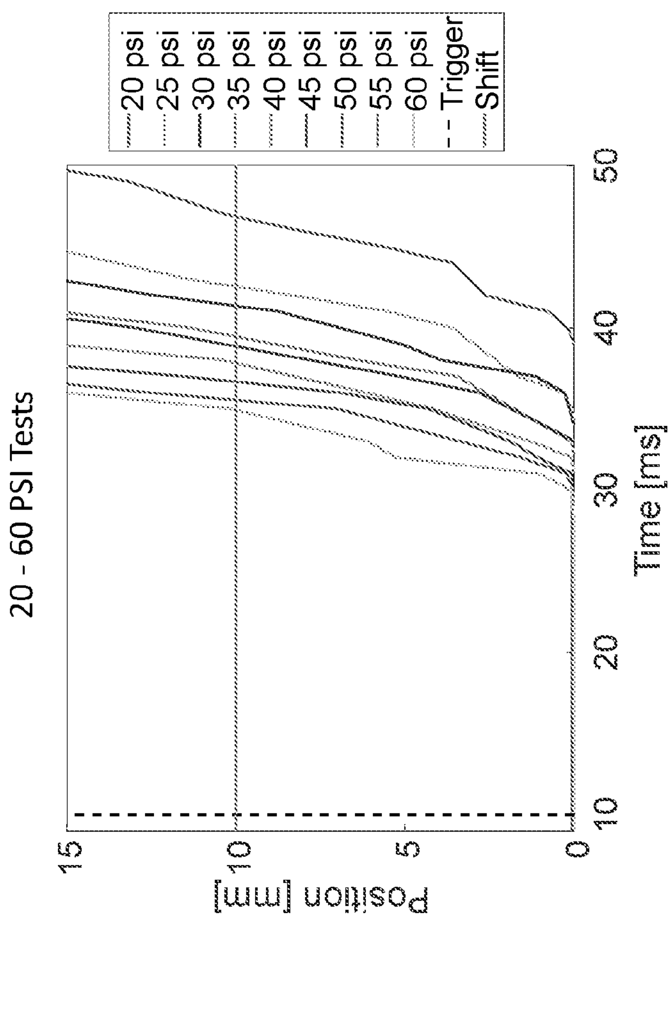


FIG. 25A

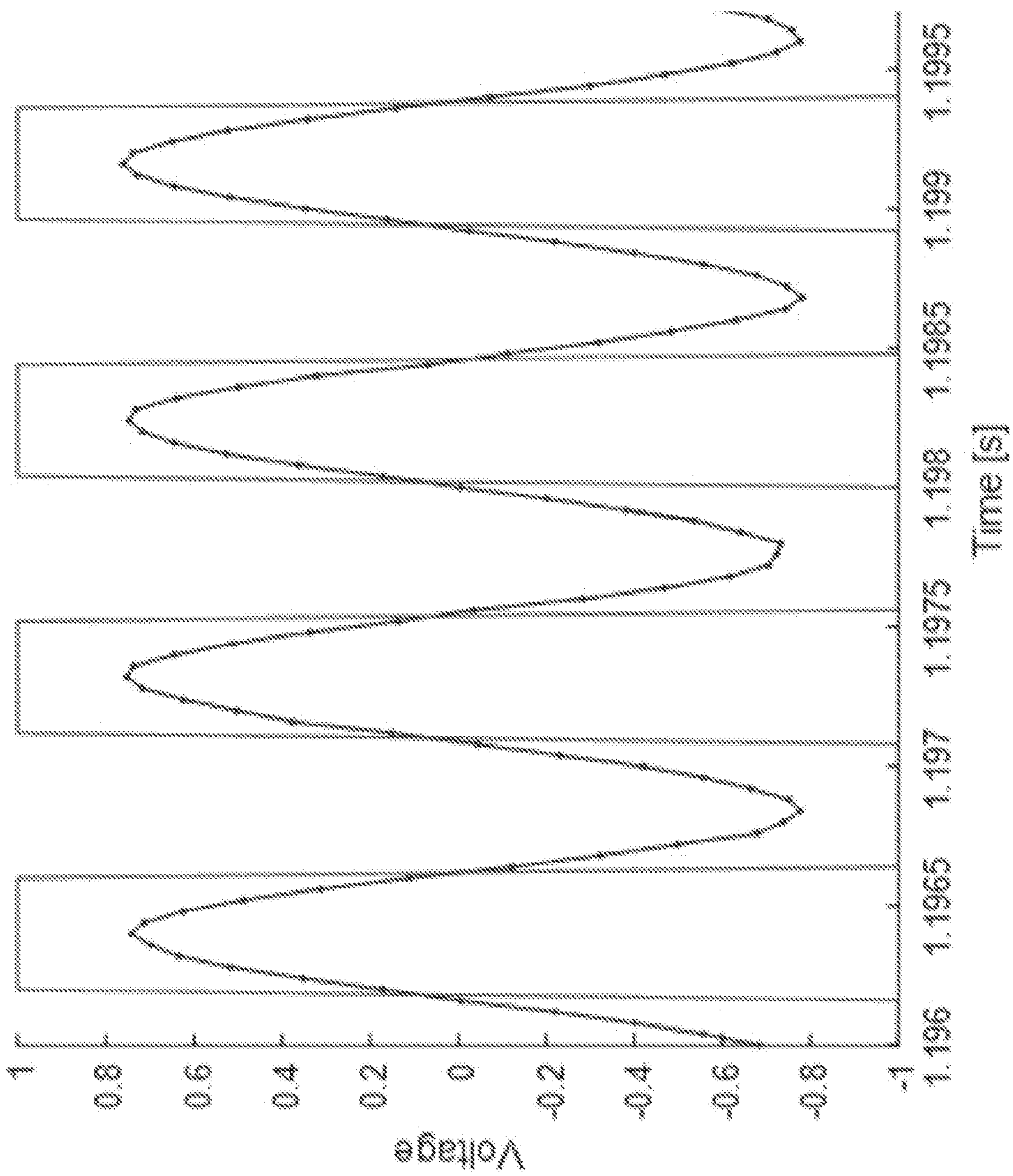

Voltage
1
0.8
0.6
0.4
0.2
0
-0.2
-0.4
-0.6
-0.8
-1
1.196
1.1965
1.197
1.1975
1.198
1.1985
1.199
1.1995
Time [s]

FIG. 26

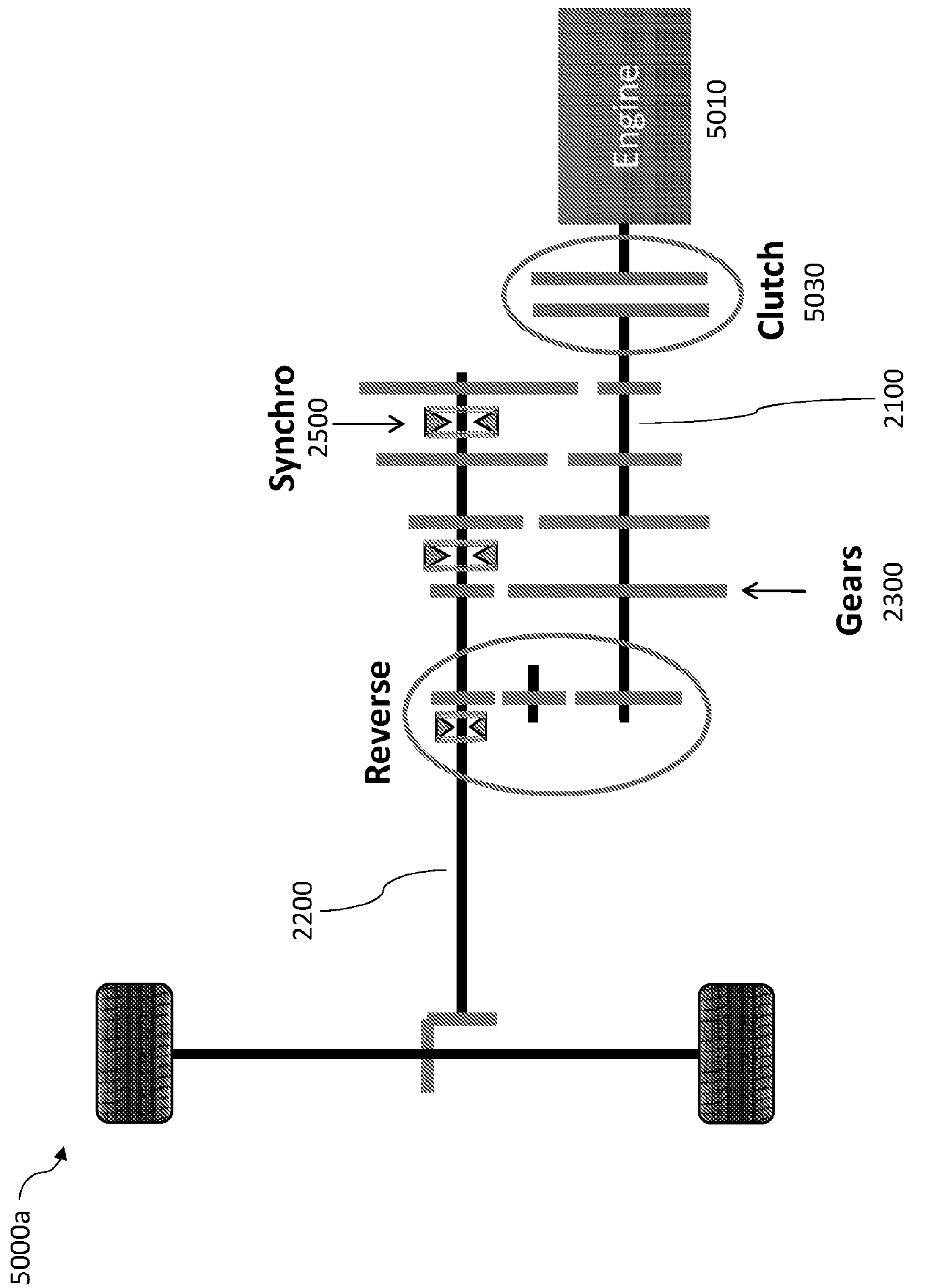

Engine
5010
Clutch
5030
Synchro
2500
2100
Gears
2300
Reverse
2200
5000a

FIG. 27A

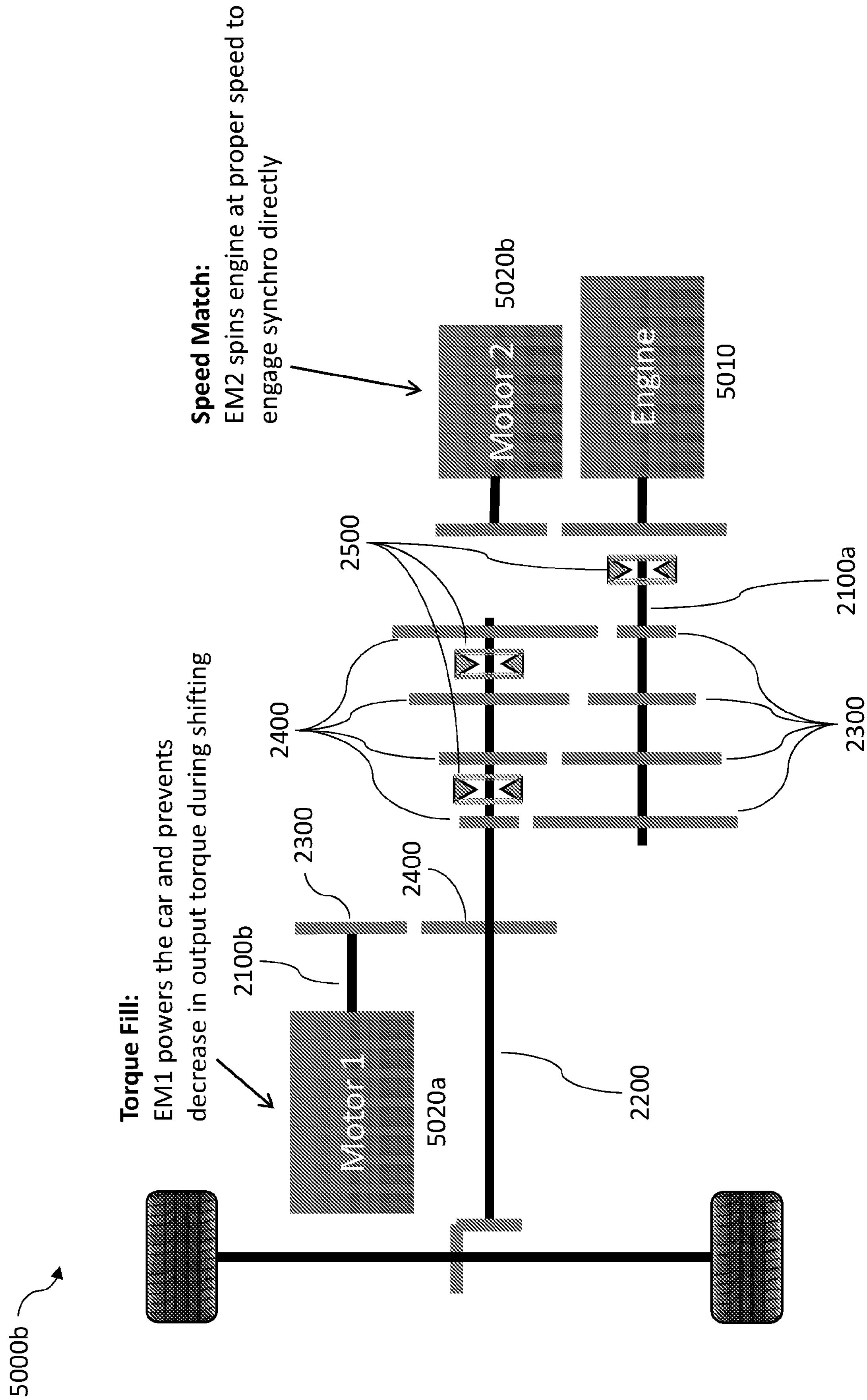

Speed Match:
EM2 spins engine at proper speed to engage synchro directly
Torque Fill:
EM1 powers the car and prevents decrease in output torque during shifting
Motor 1
Motor 2
Engine
5000b
5020a
5020b
5010
2100a
2100b
2200
2300
2400
2500

FIG. 27B

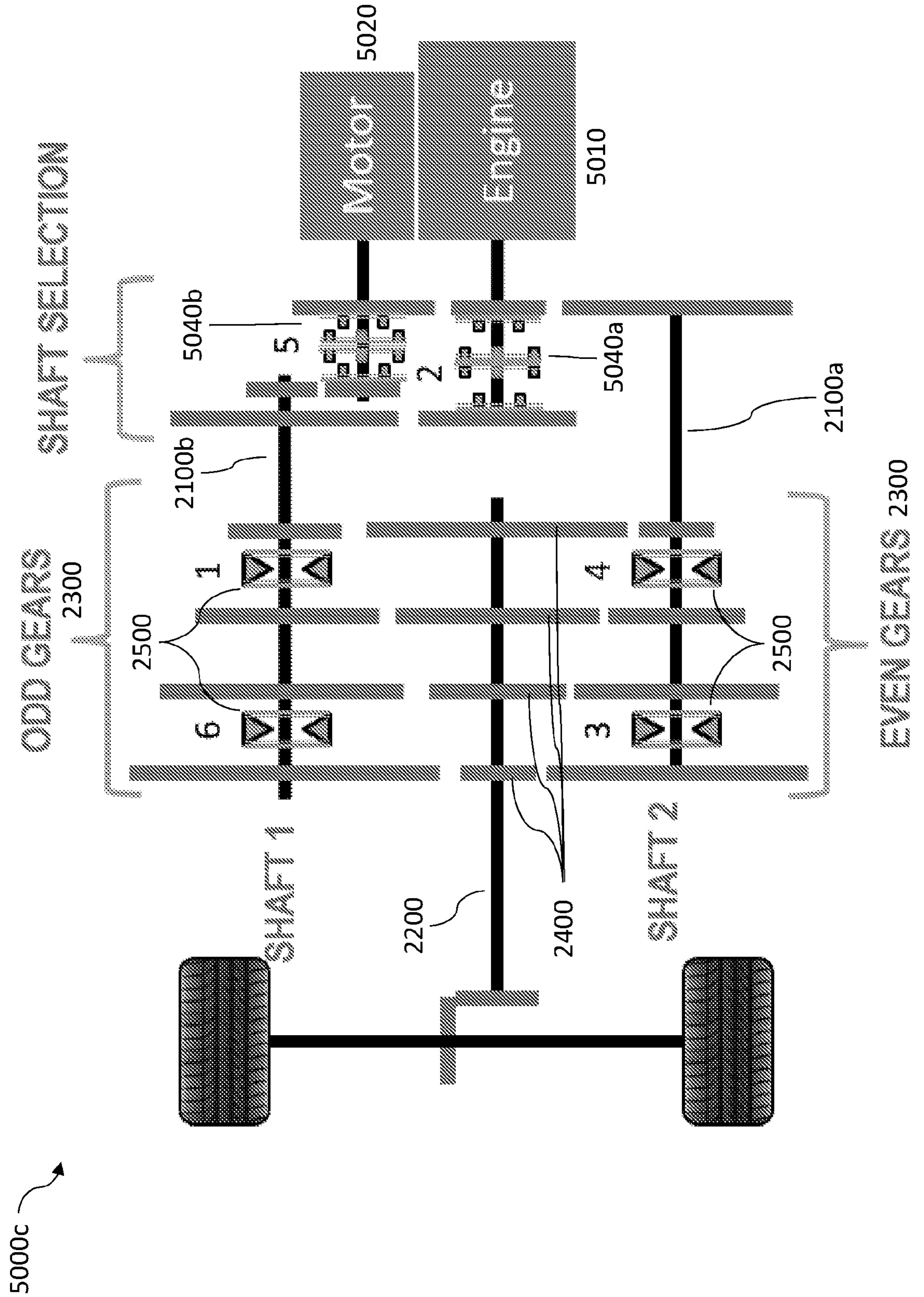

SHAFT SELECTION
Motor
Engine
5020
5010
5040b
5040a
5
2
2100b
2100a
ODD GEARS
2300
EVEN GEARS
2300
1
6
4
3
2500
2500
SHAFT 1
SHAFT 2
2200
2400
5000c

FIG. 27C

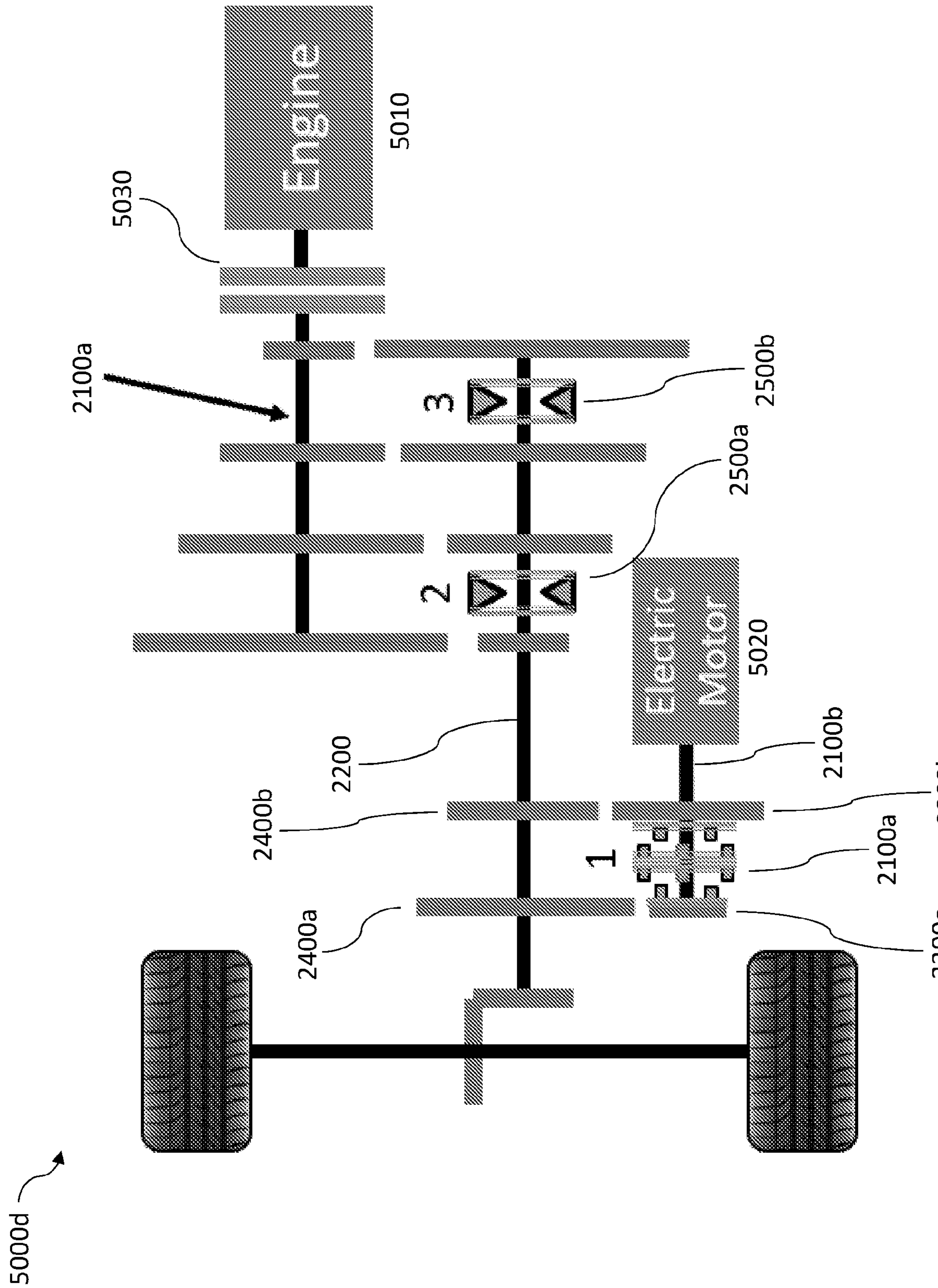

Engine
5010
5030
2100a
3
2500b
2500a
2
Electric Motor
5020
2200
2100b
2400b
2300b
1
2100a
2400a
2300a
5000d

FIG. 27D

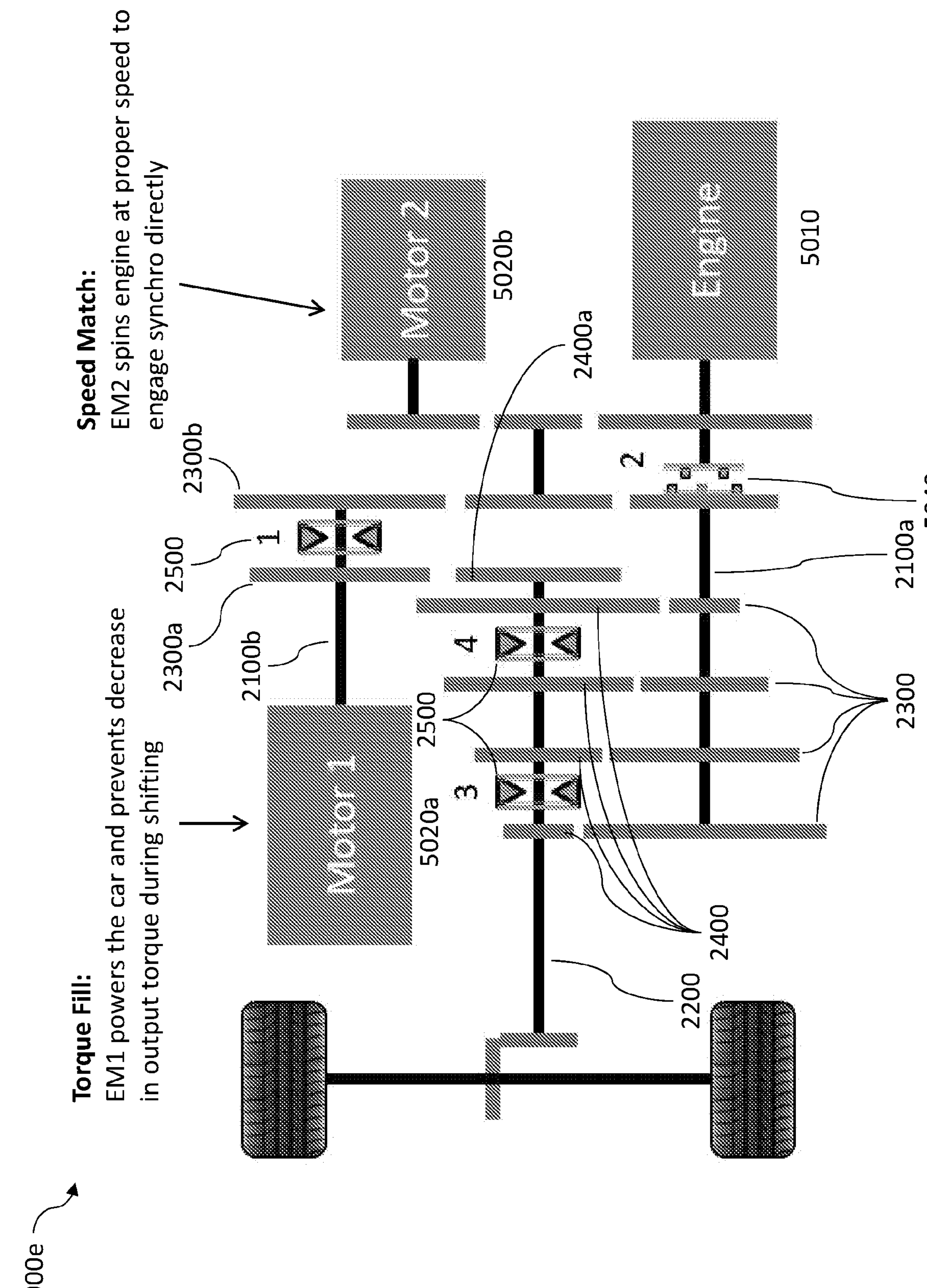

Torque Fill:
EM1 powers the car and prevents decrease
in output torque during shifting
Speed Match:
EM2 spins engine at proper speed to
engage synchro directly
Motor 1
Motor 2
Engine
5000e
5020a
5020b
5010
5040
2100a
2100b
2200
2300
2300a
2300b
2400
2400a
2500
1
2
3
4

FIG. 27E

INTERNATIONAL SEARCH REPORT

International application No.
PCT/US19/48387

A. CLASSIFICATION OF SUBJECT MATTER

IPC - F16D 11/14; F16H 61/04 (2019.01)

CPC - F16D 11/14; F16H 61/0403

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)
See Search History document

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

Electronic data base consulted during the international search (name of data base and, where practicable, search terms used)
See Search History document

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
X	US 2013/0144553 A1 (NIKON CORPORATION) 6 June 2013; figure 1; paragraphs [0076], [0078], [0080]	1-3
Y	US 5,827,148 A (SETO T et al) 27 October 1998; figure 2; column 8, lines 1-15; column 10, lines 20-30	1, 4-8, 10, 13-16, 29, 32, 34
Y	US 2018/0119802 A1 (TOYOTA JIDOSHA KABUSHIKI KAISHA et al) 3 May 2018; figures 1-2; paragraphs [0051], [0058], [0071], [0074]	1, 4-8, 10, 13-18, 23-27, 29, 32, 34
Y	WO 00/15455 A2 (PAICE CORPORATION) 23 March 2000; figure 4; page 33, lines 10-35	17-18, 23-27
A	US 2014/0291102 A1 (AISIN SEIKI KABUSHIKI KAISHA) 2 October 2014; entire document	1-40

☐ Further documents are listed in the continuation of Box C. ☐ See patent family annex.

* Special categories of cited documents:

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

"D" document cited by the applicant in the international application

"E" earlier application or patent 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
28 October 2019 (28.10.2019)	21 NOV 2019

Name and mailing address of the ISA/US	Authorized officer
Mail Stop PCT, Attn: ISA/US, Commissioner for Patents P.O. Box 1450, Alexandria, Virginia 22313-1450 Facsimile No. 571-273-8300	Shane Thomas Telephone No. PCT Helpdesk: 571-272-4300