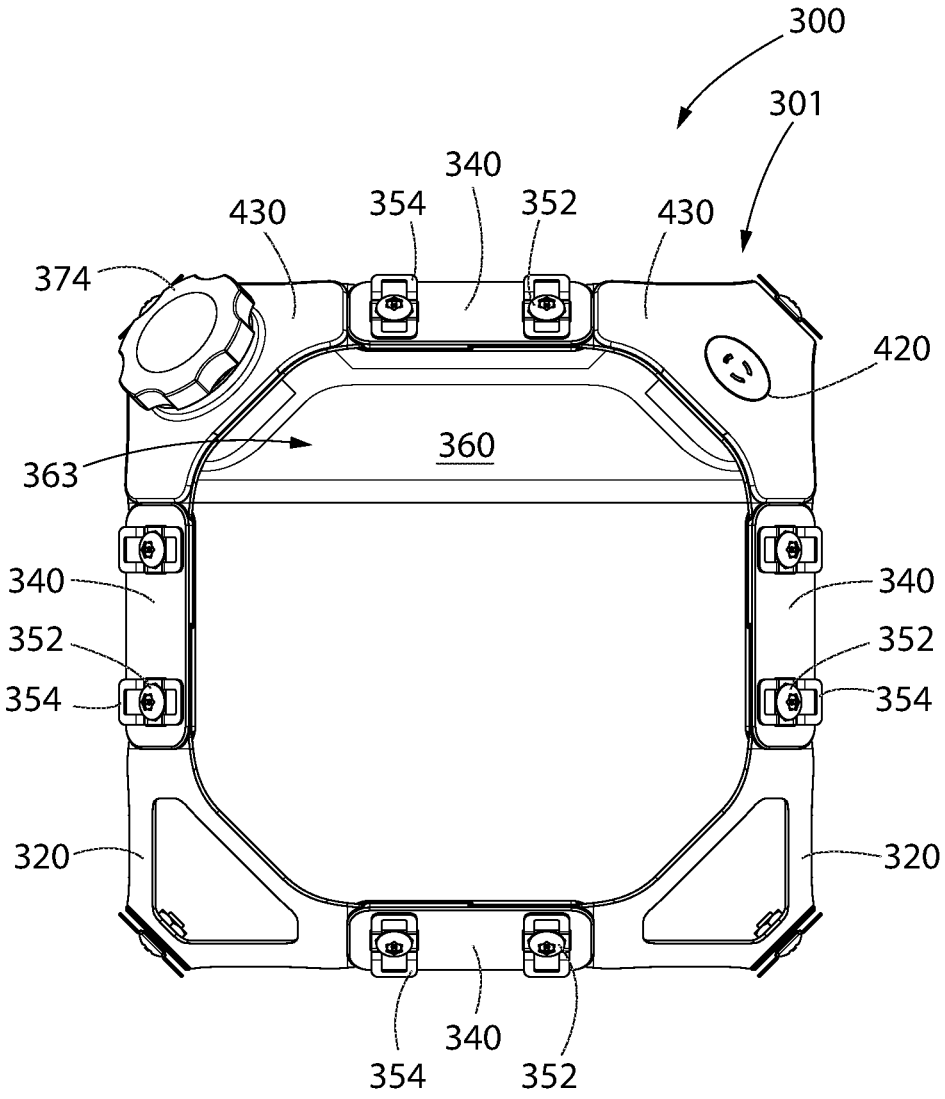

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FIG. 77

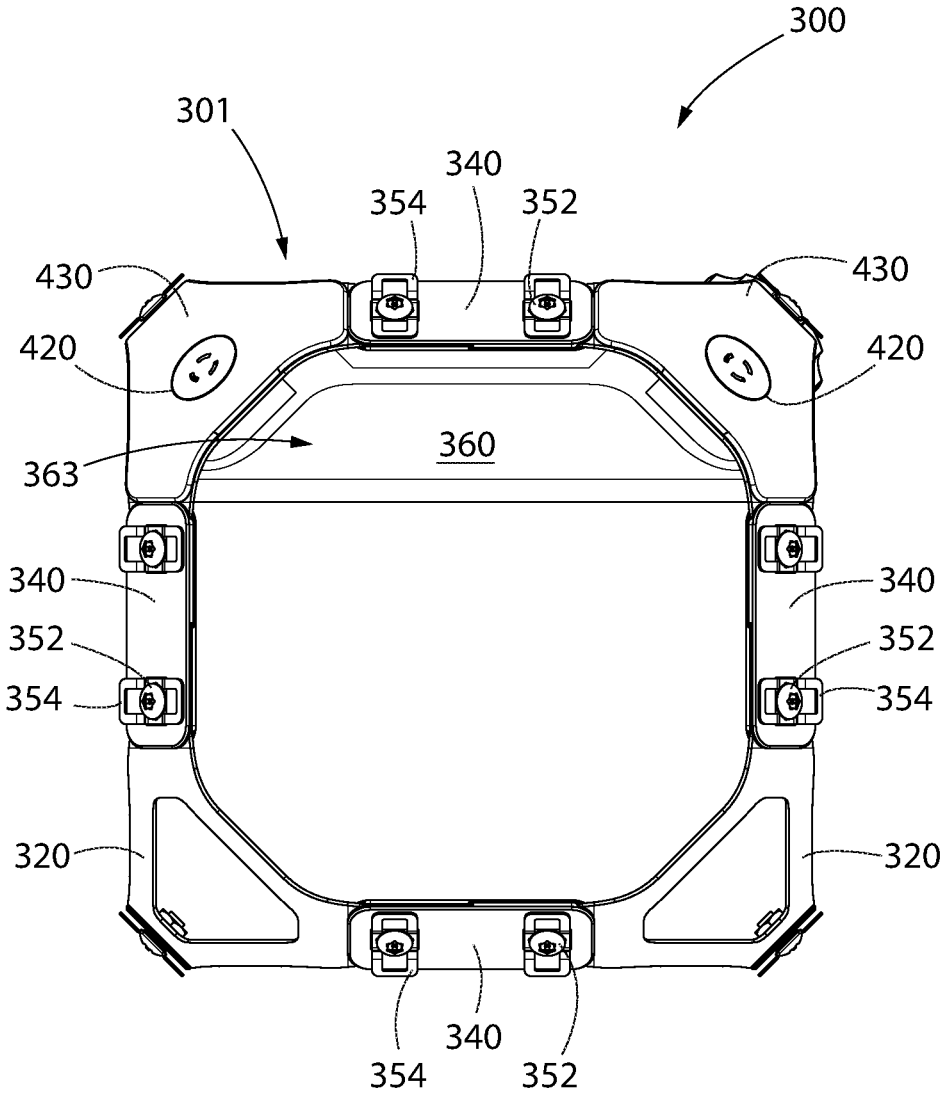

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FIG. 78

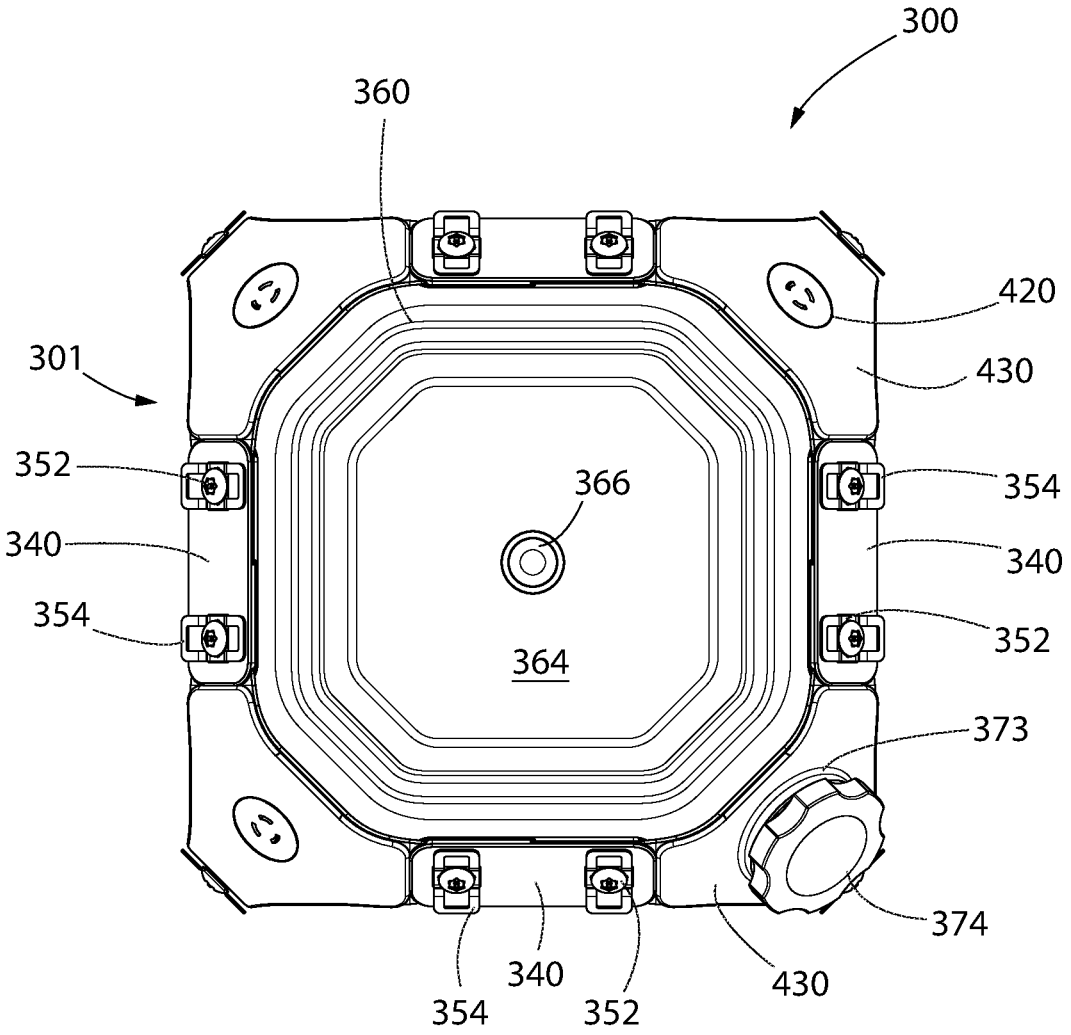

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FIG. 79

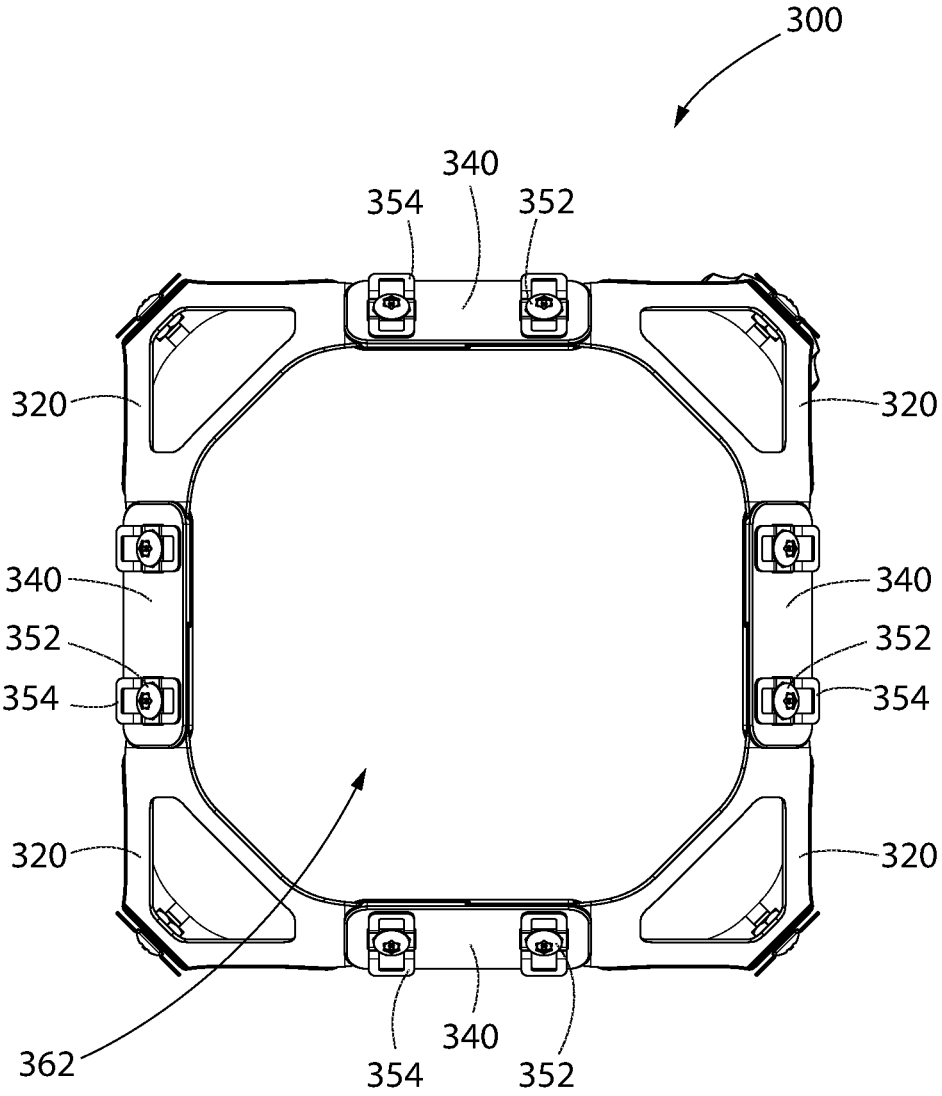

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FIG. 80

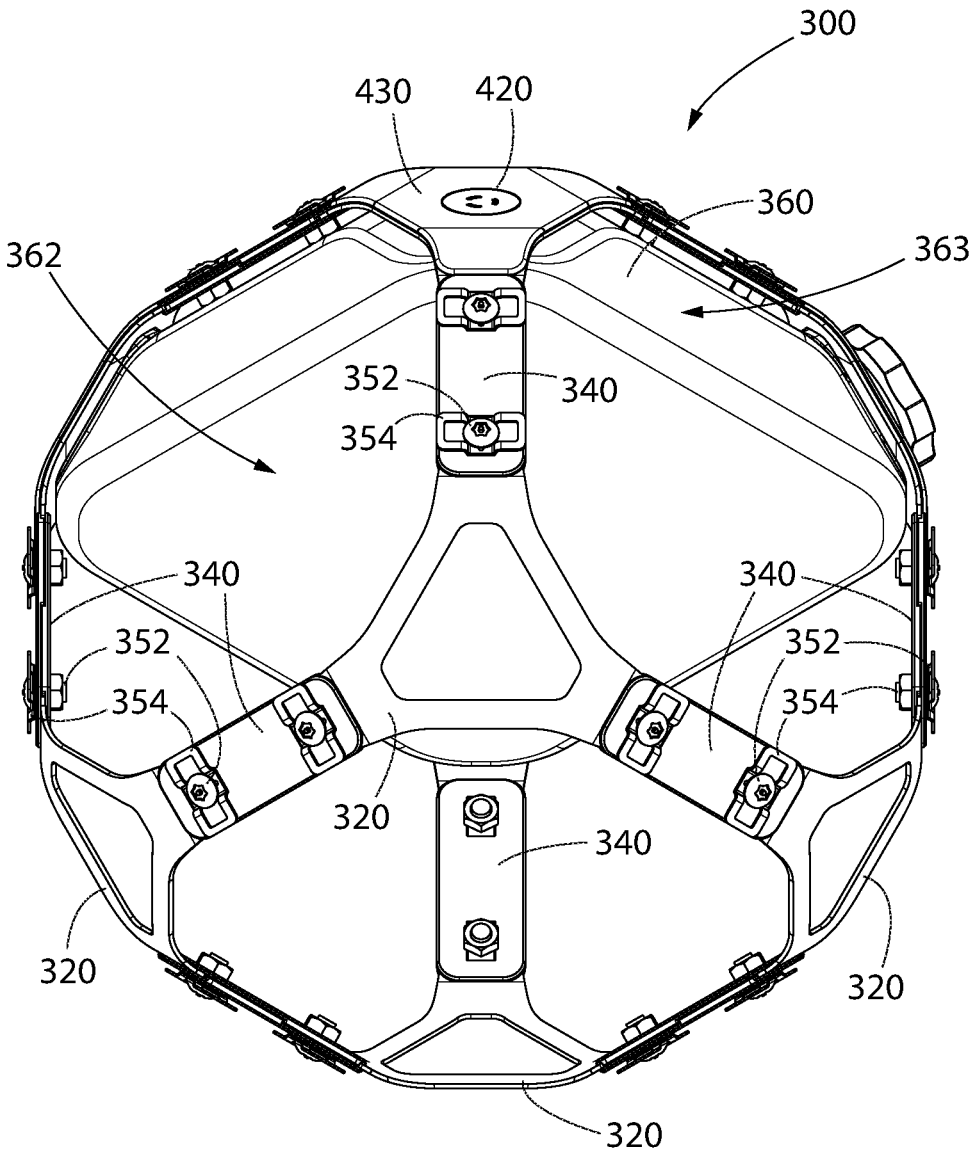

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FIG. 81

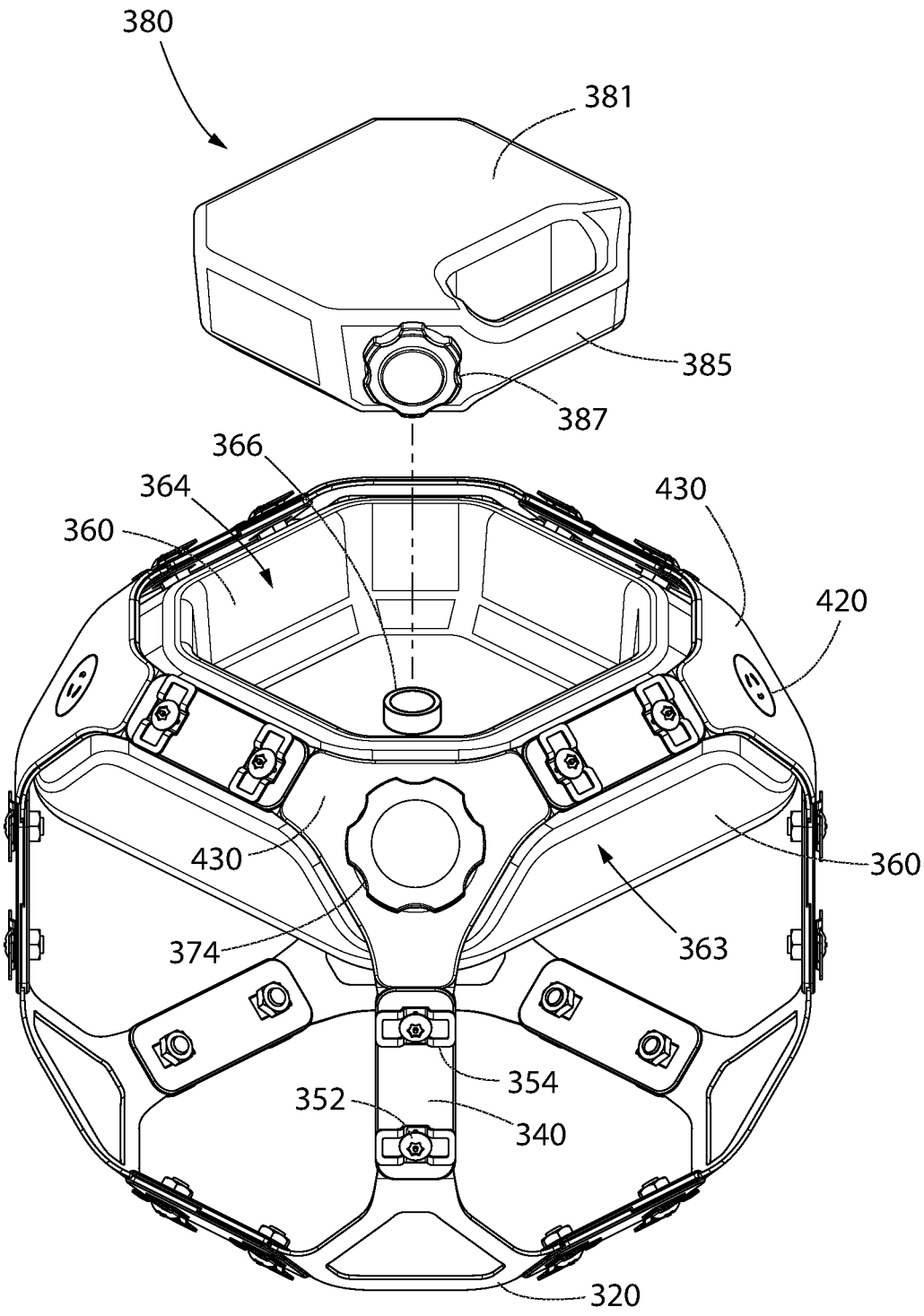

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FIG. 82

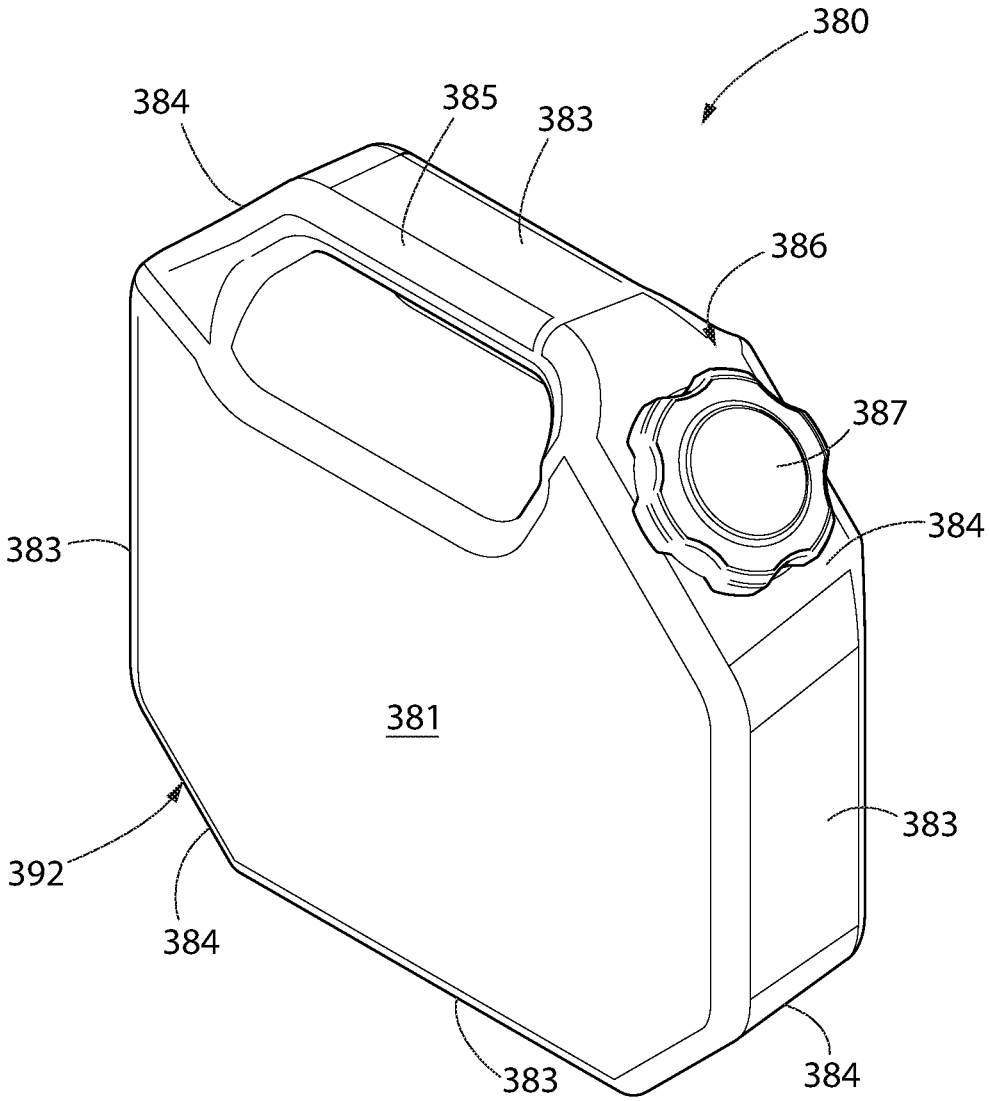


FIG. 83

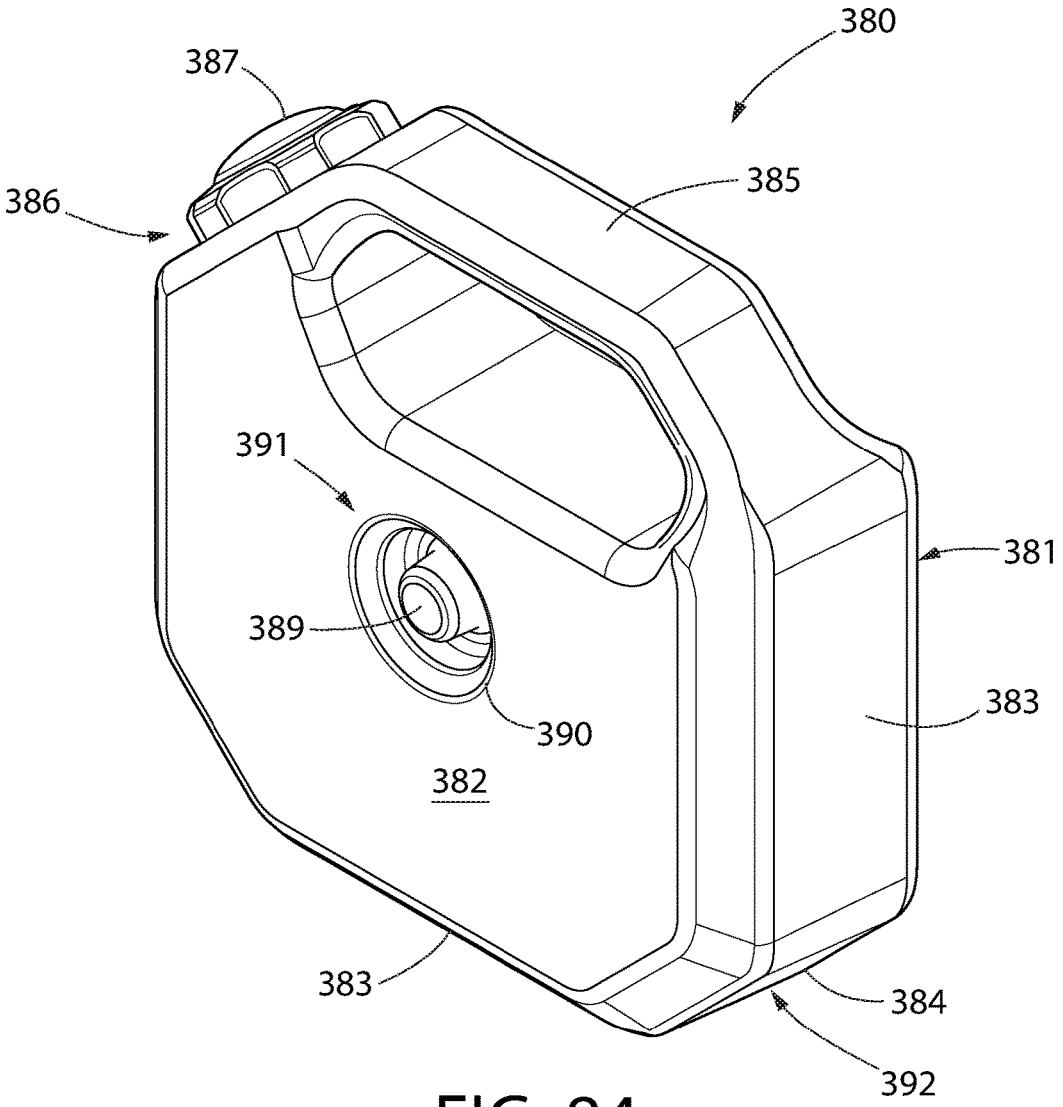


FIG. 84

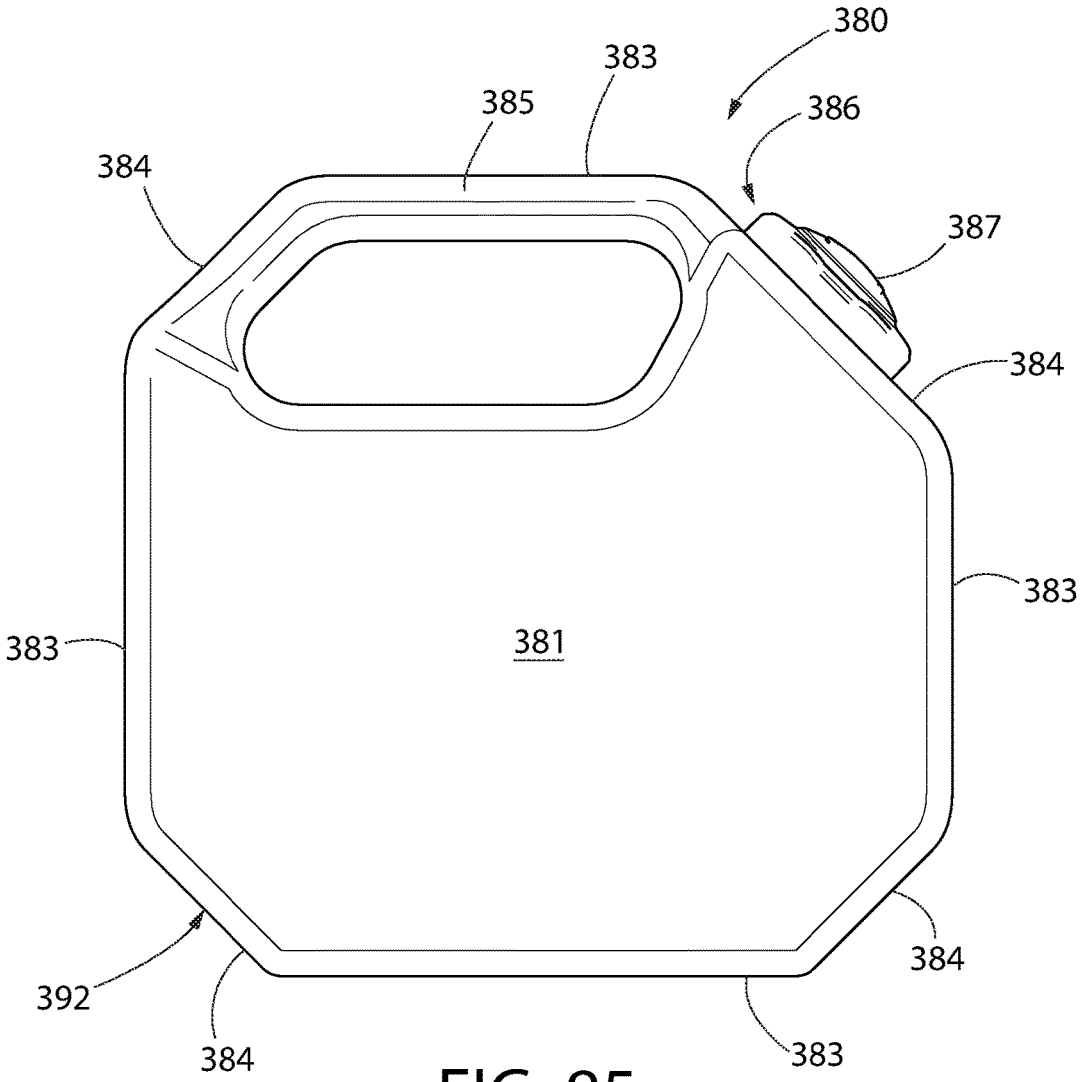


FIG. 85

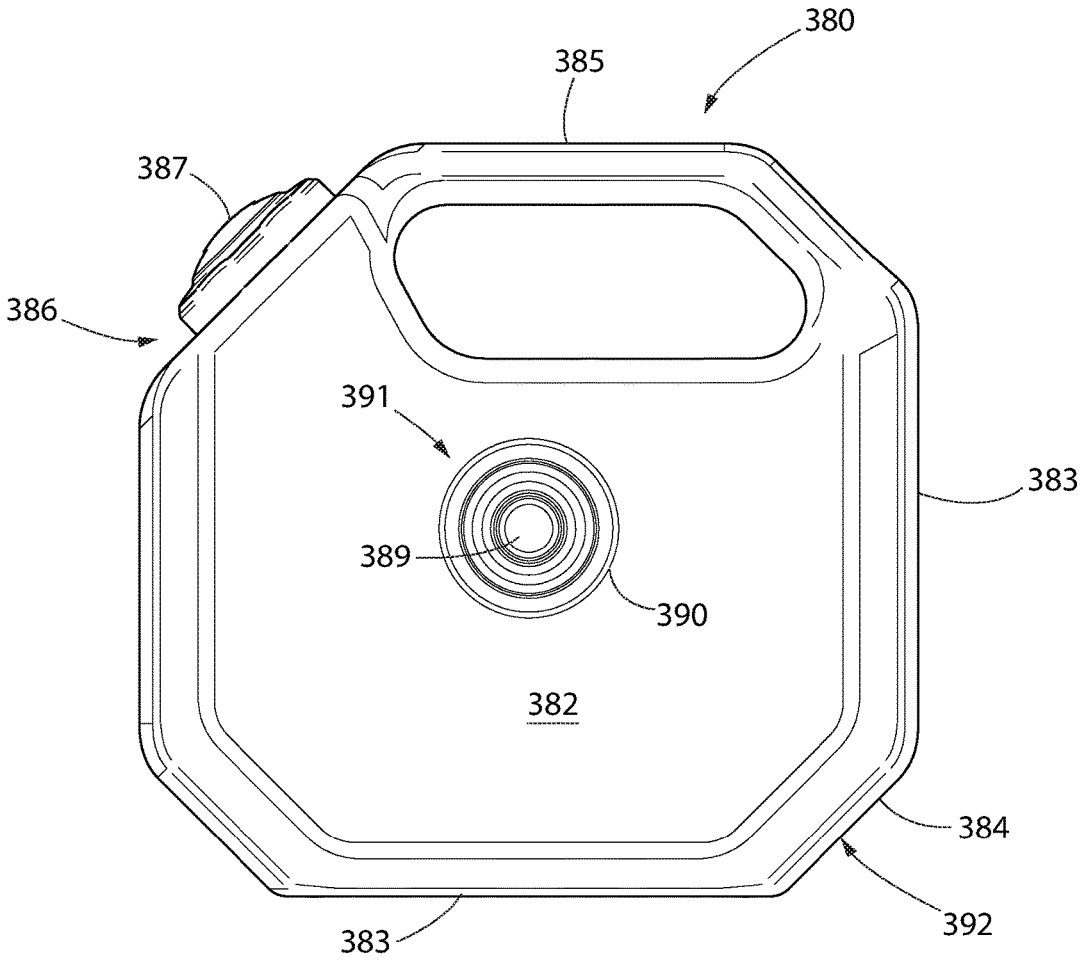


FIG. 86

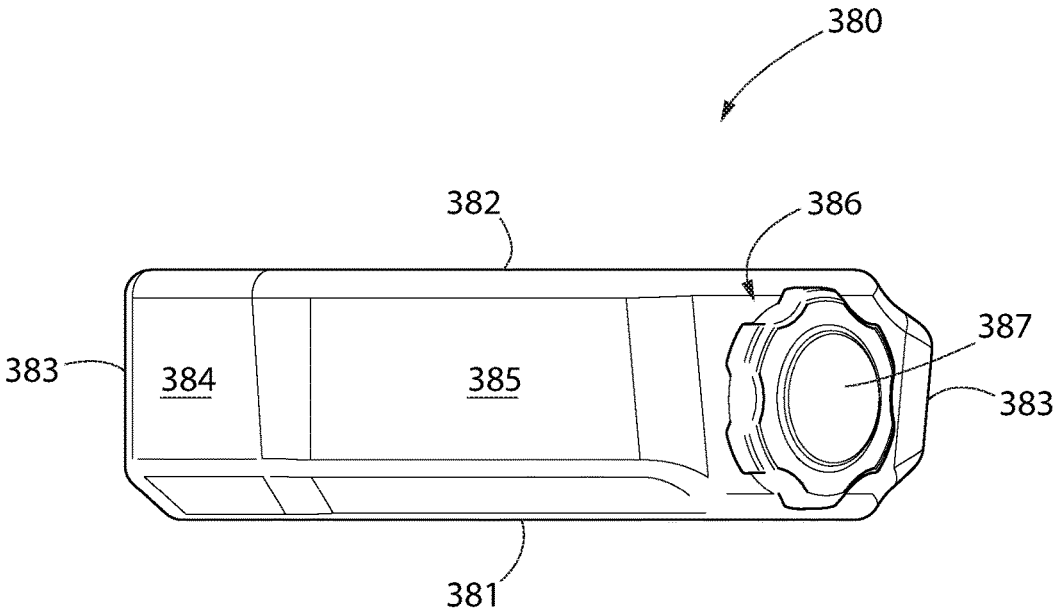


FIG. 87

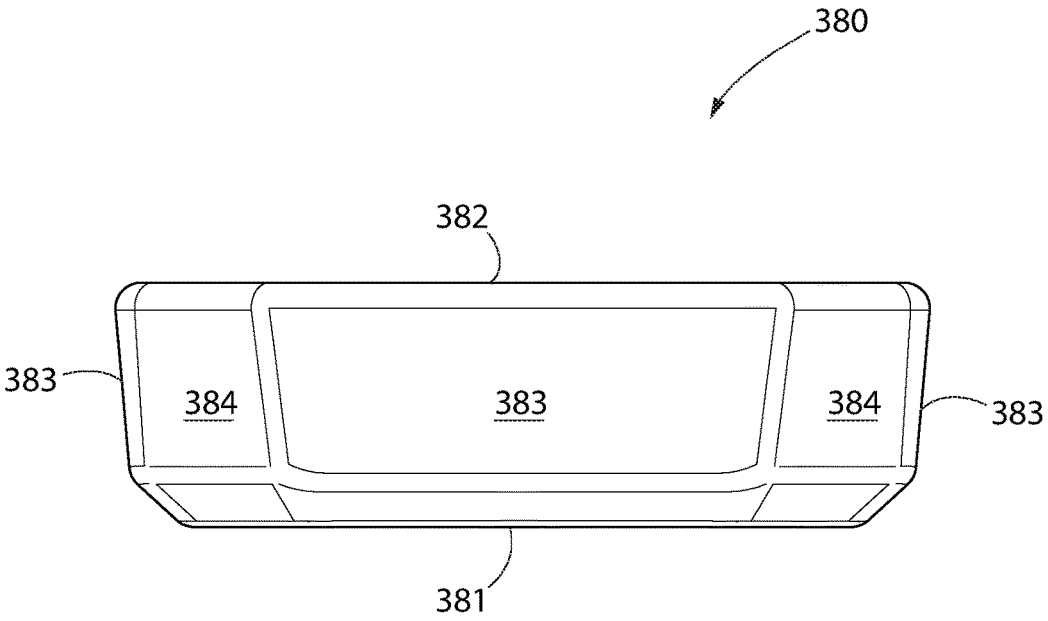


FIG. 88

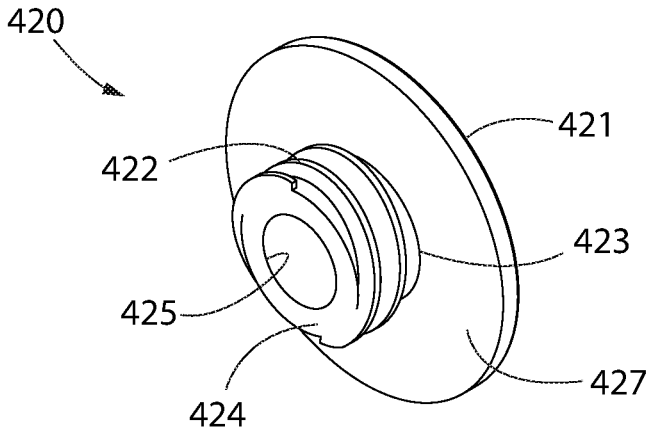


FIG. 89

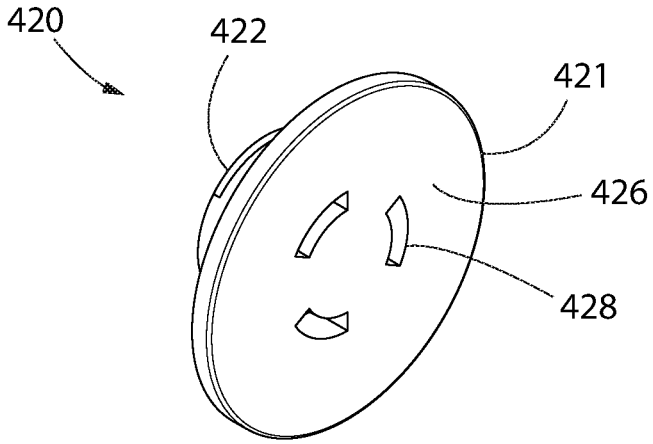


FIG. 90

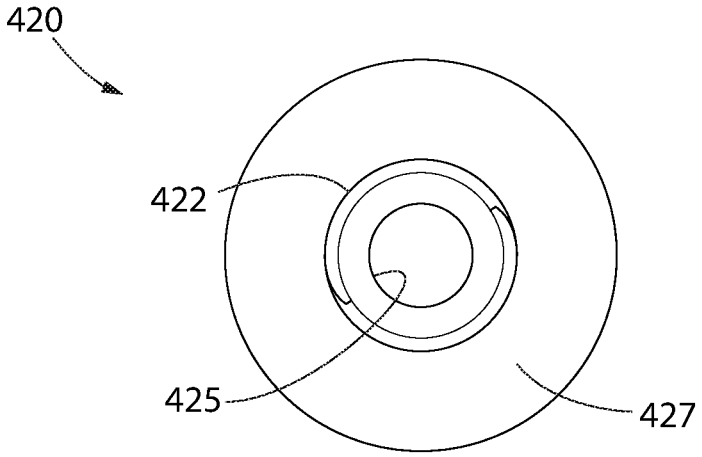


FIG. 91

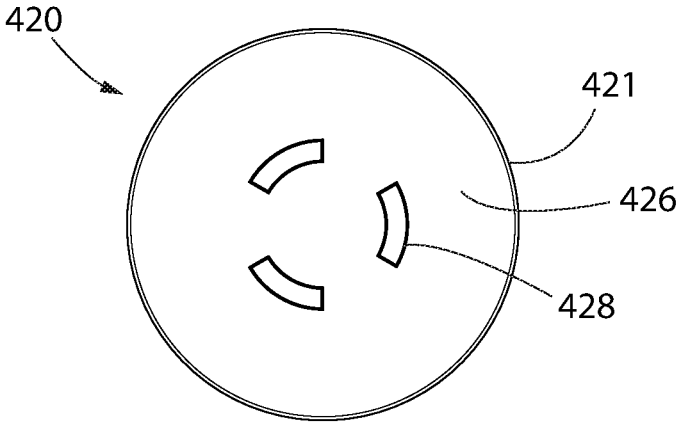


FIG. 92

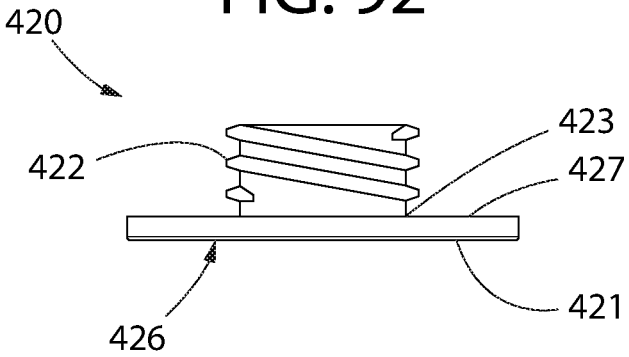


FIG. 93

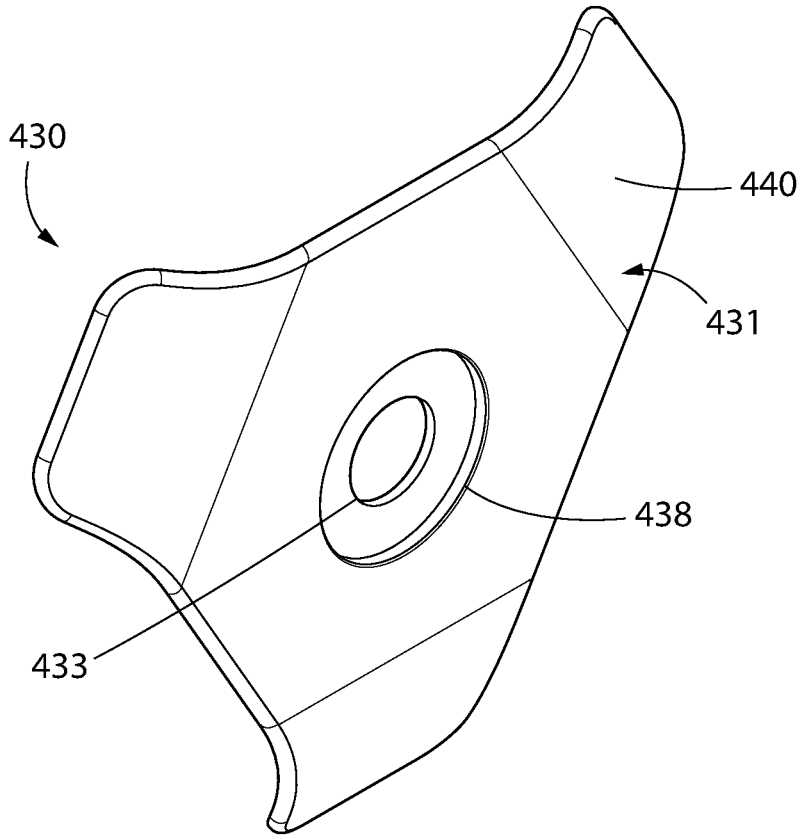


FIG. 94

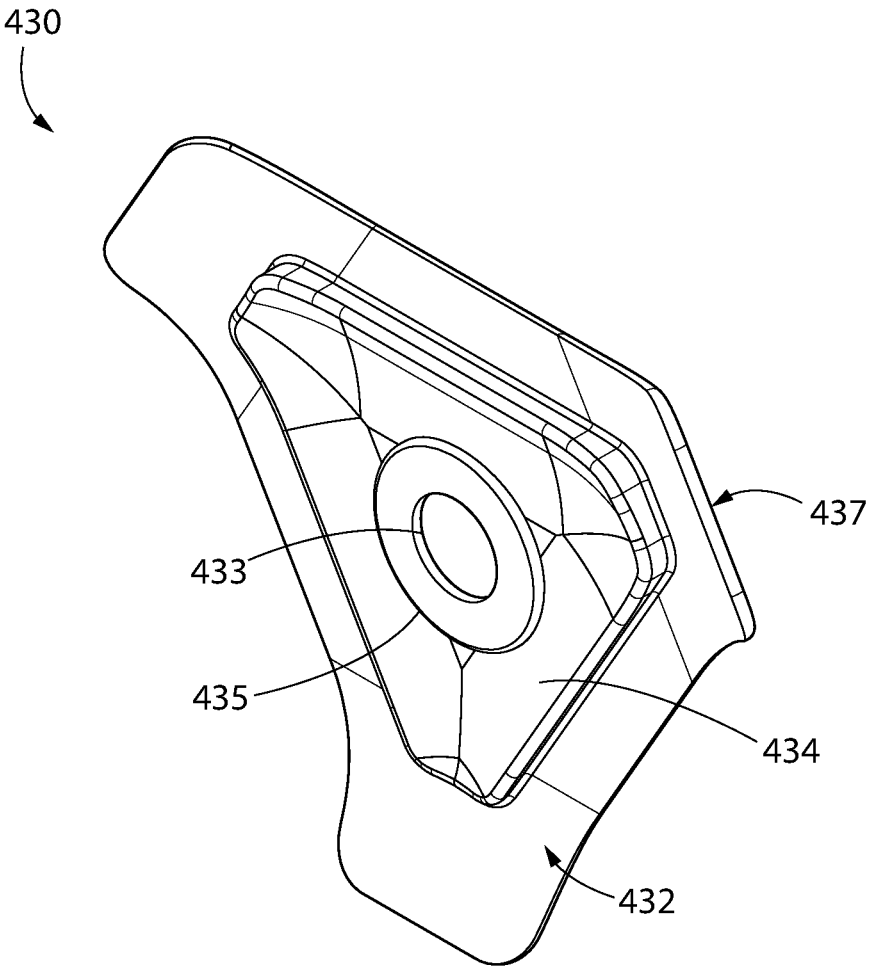


FIG. 95

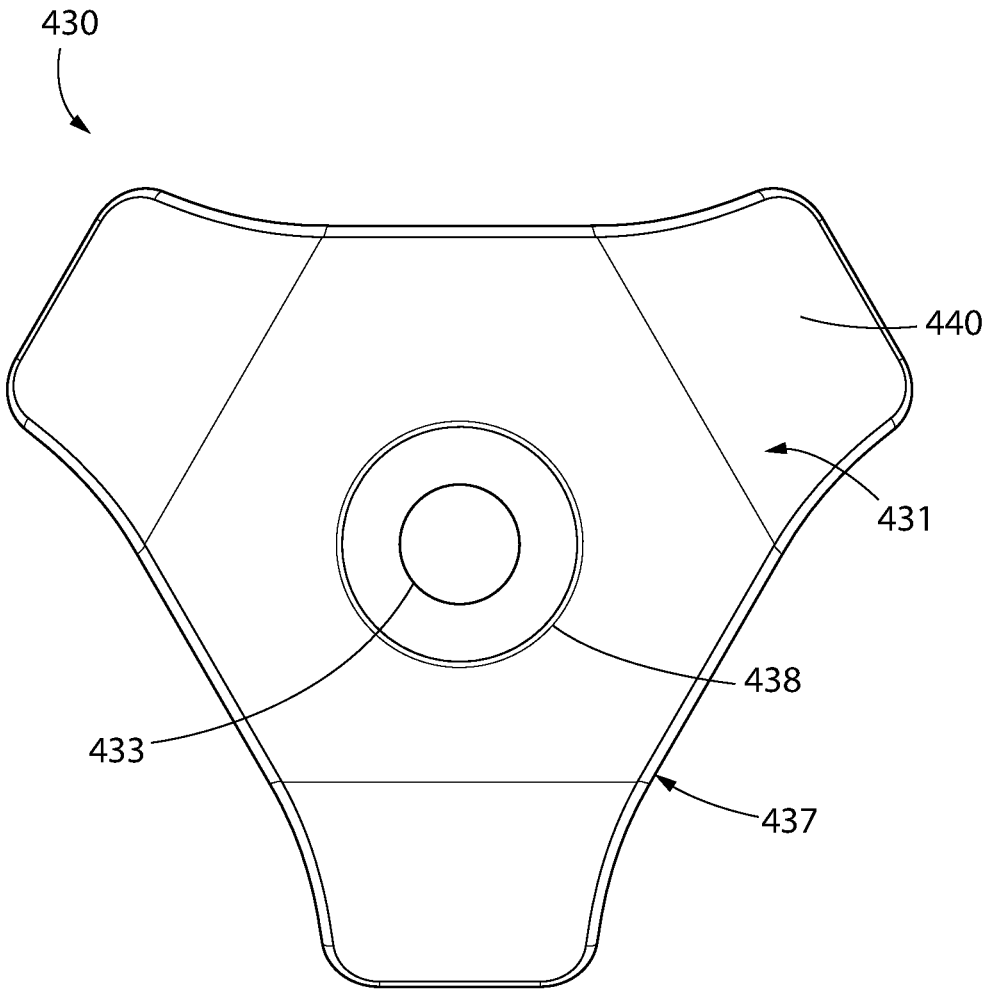


FIG. 96

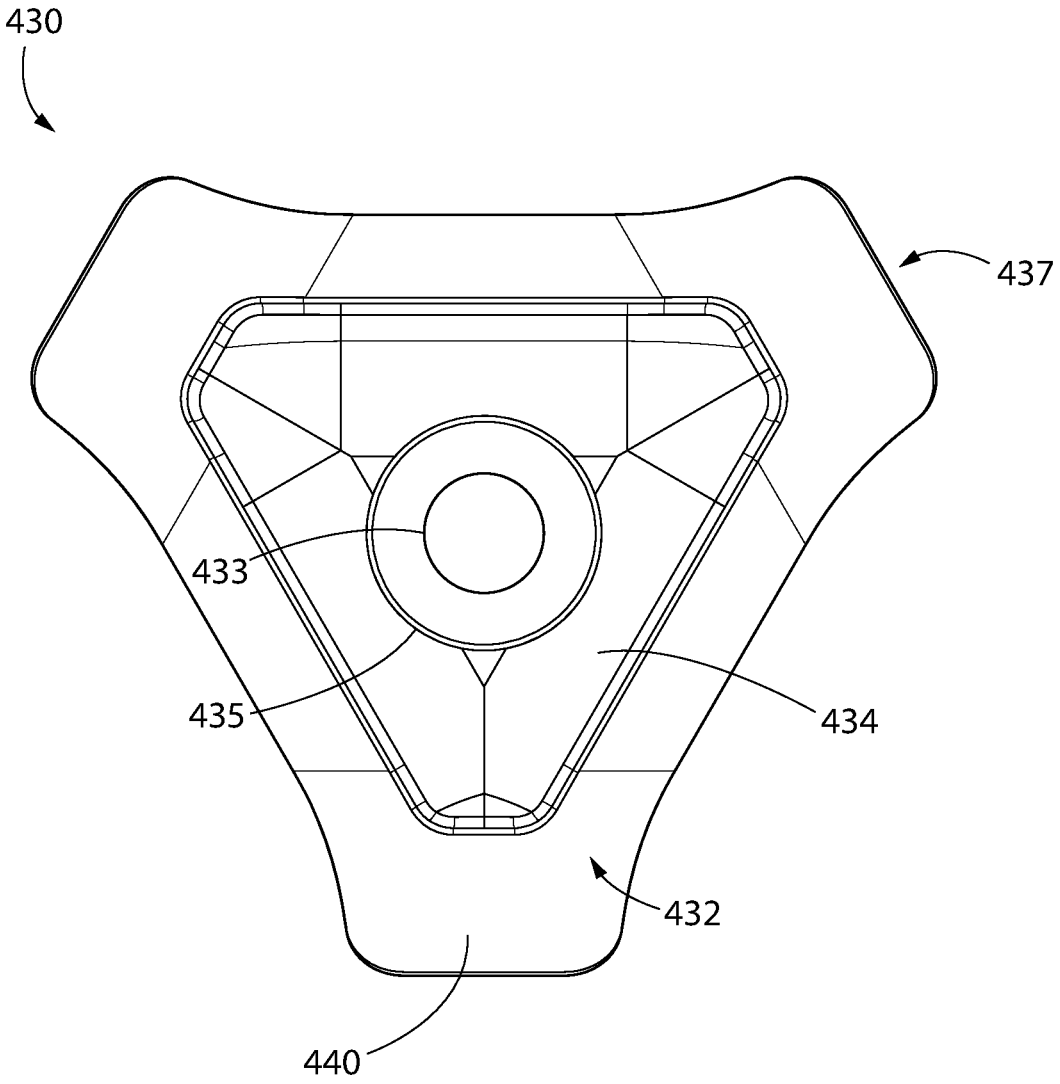


FIG. 97

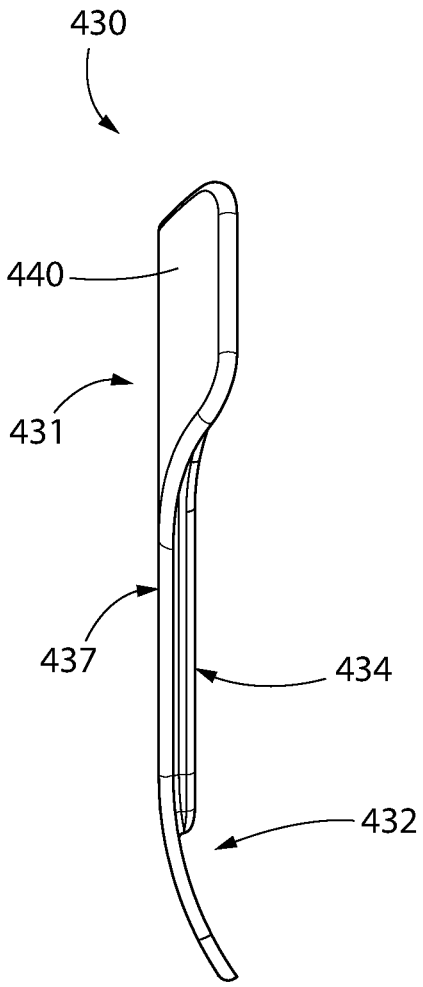


FIG. 98

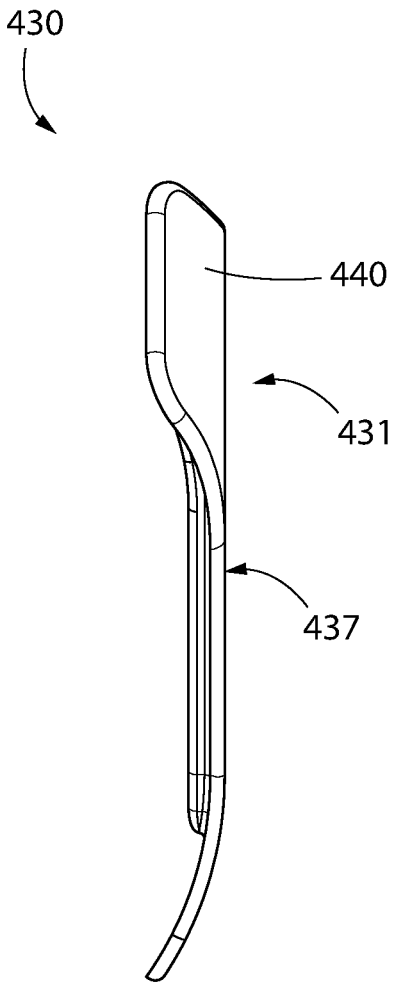


FIG. 99

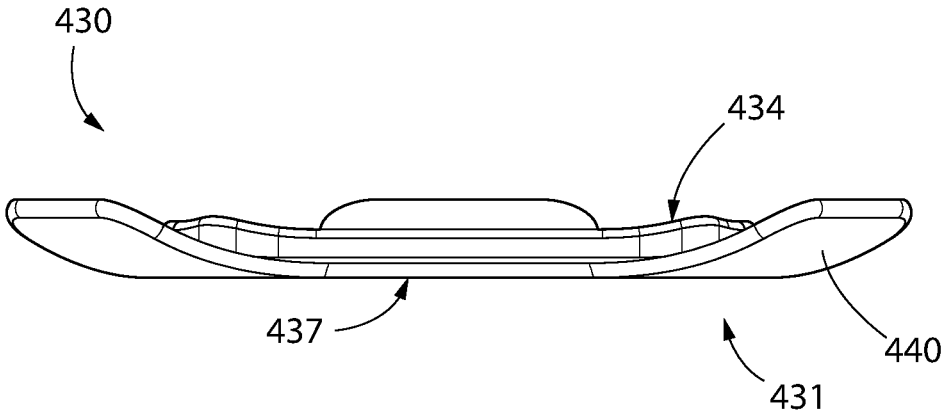


FIG. 100

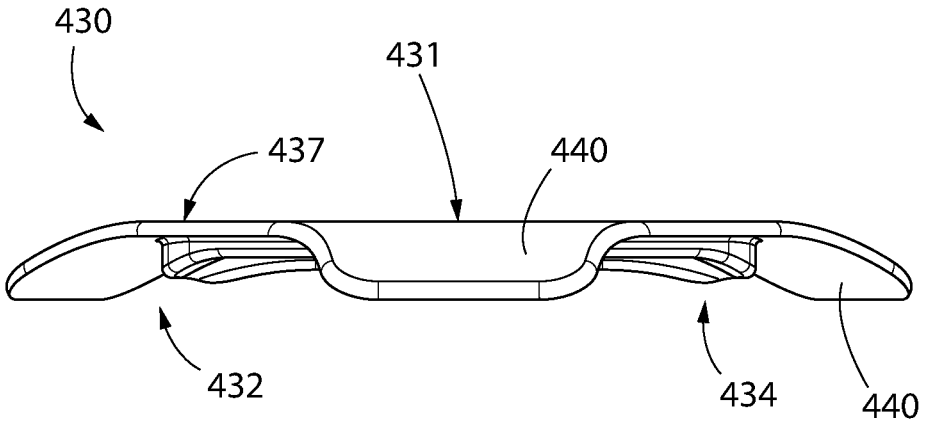


FIG. 101

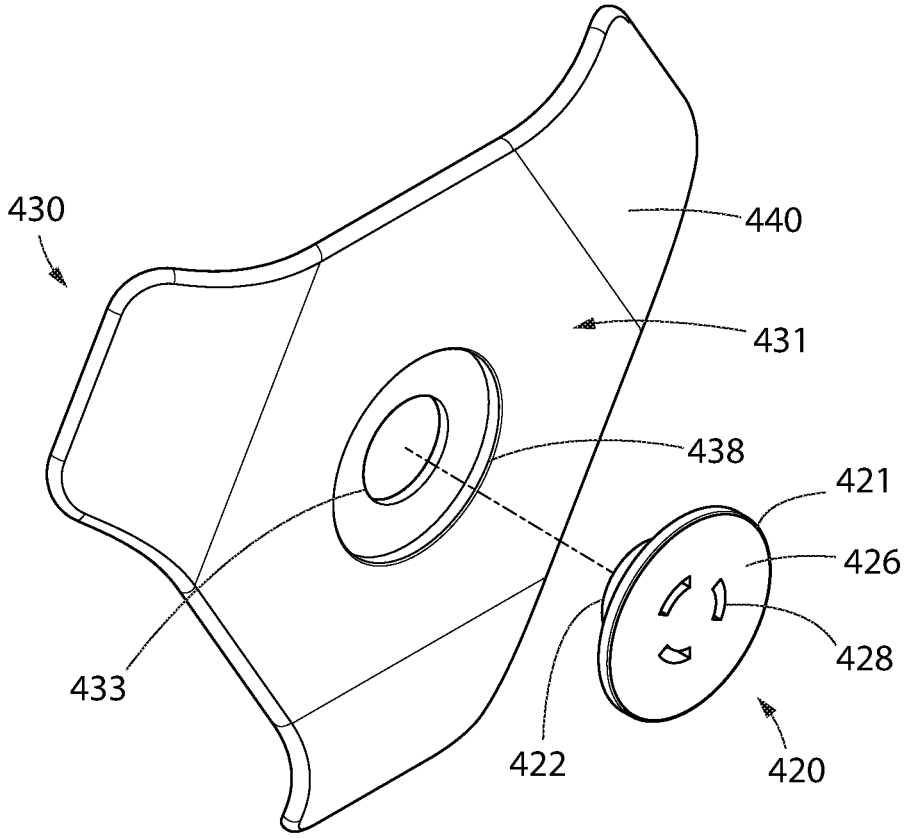


FIG. 102

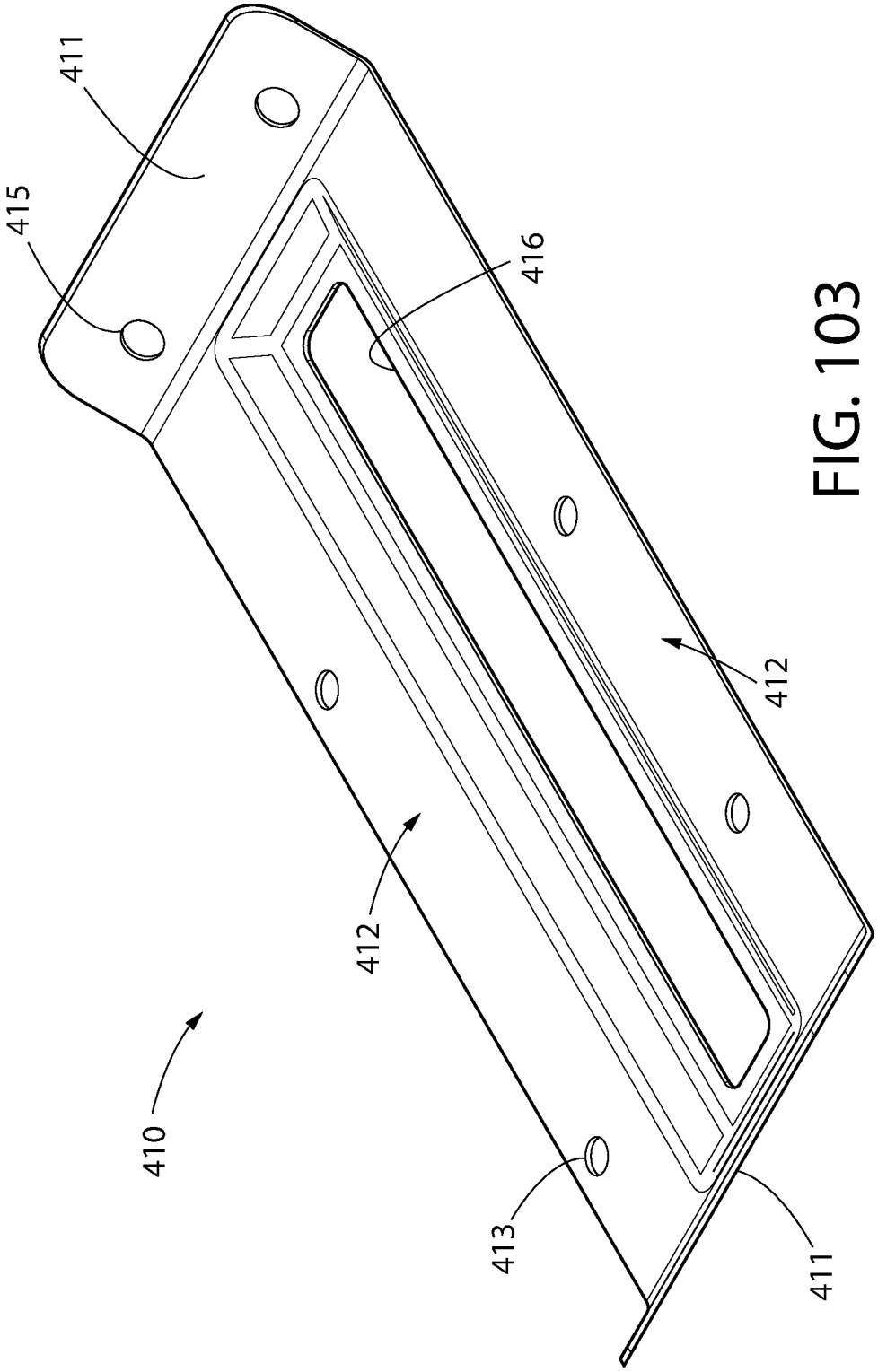


FIG. 103

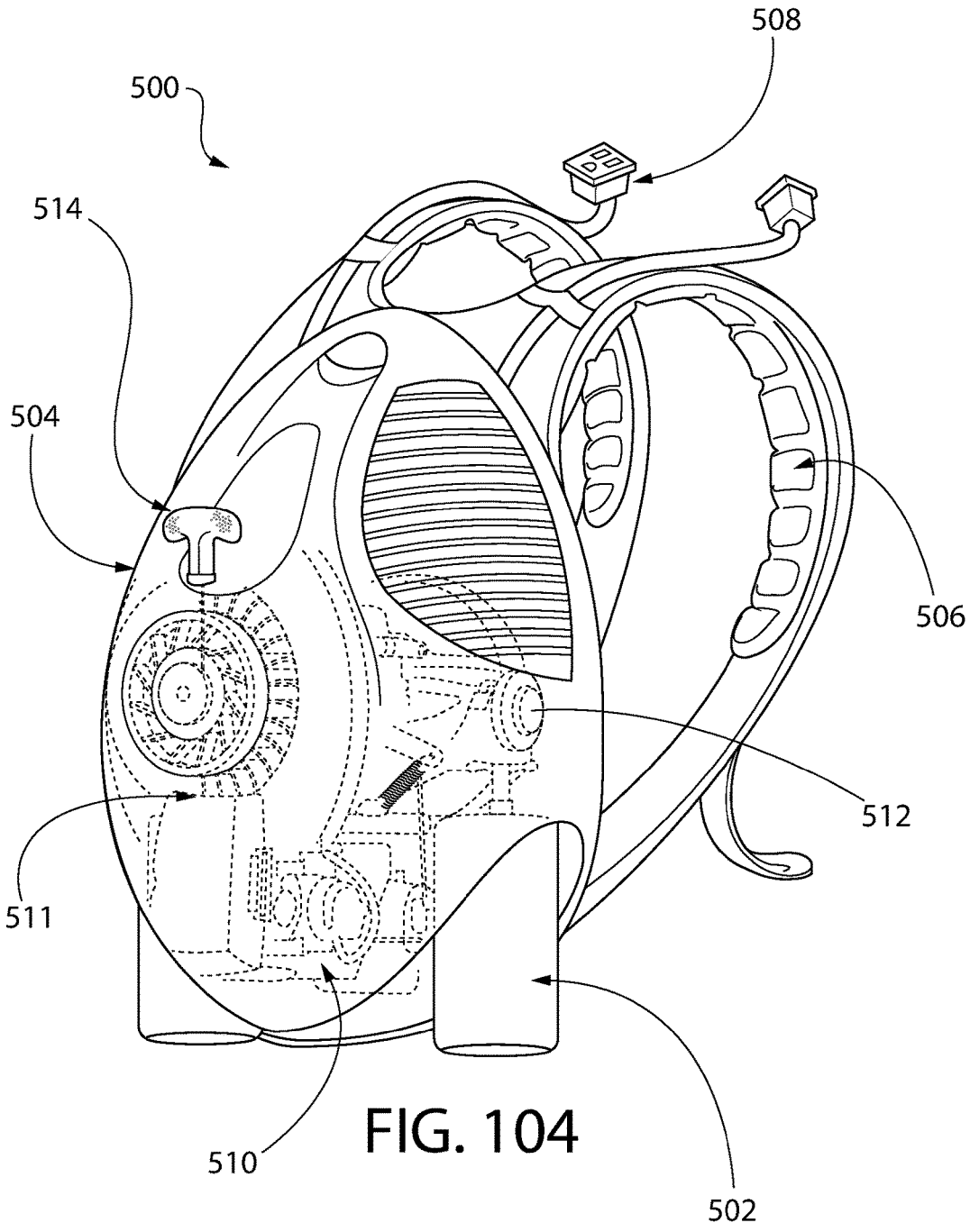


FIG. 104

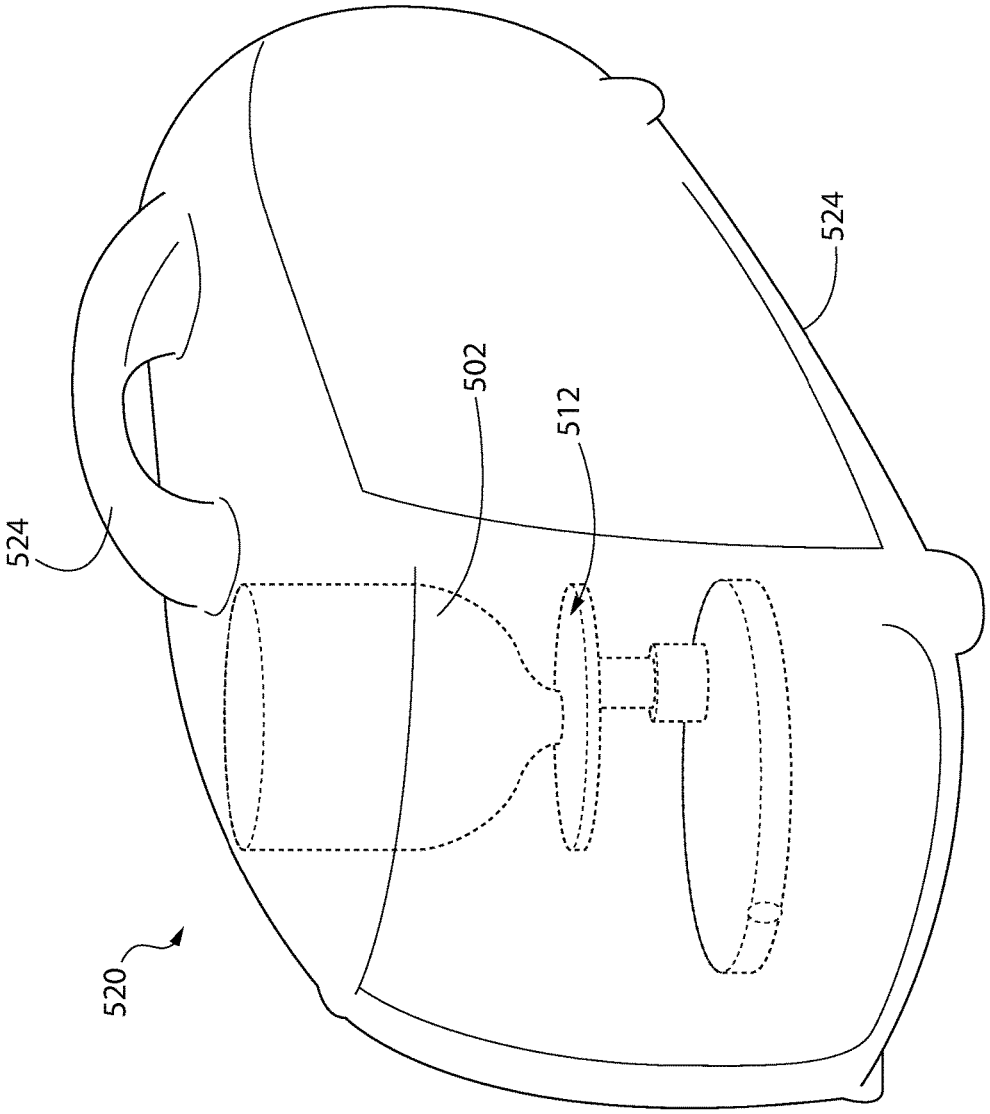


FIG. 105

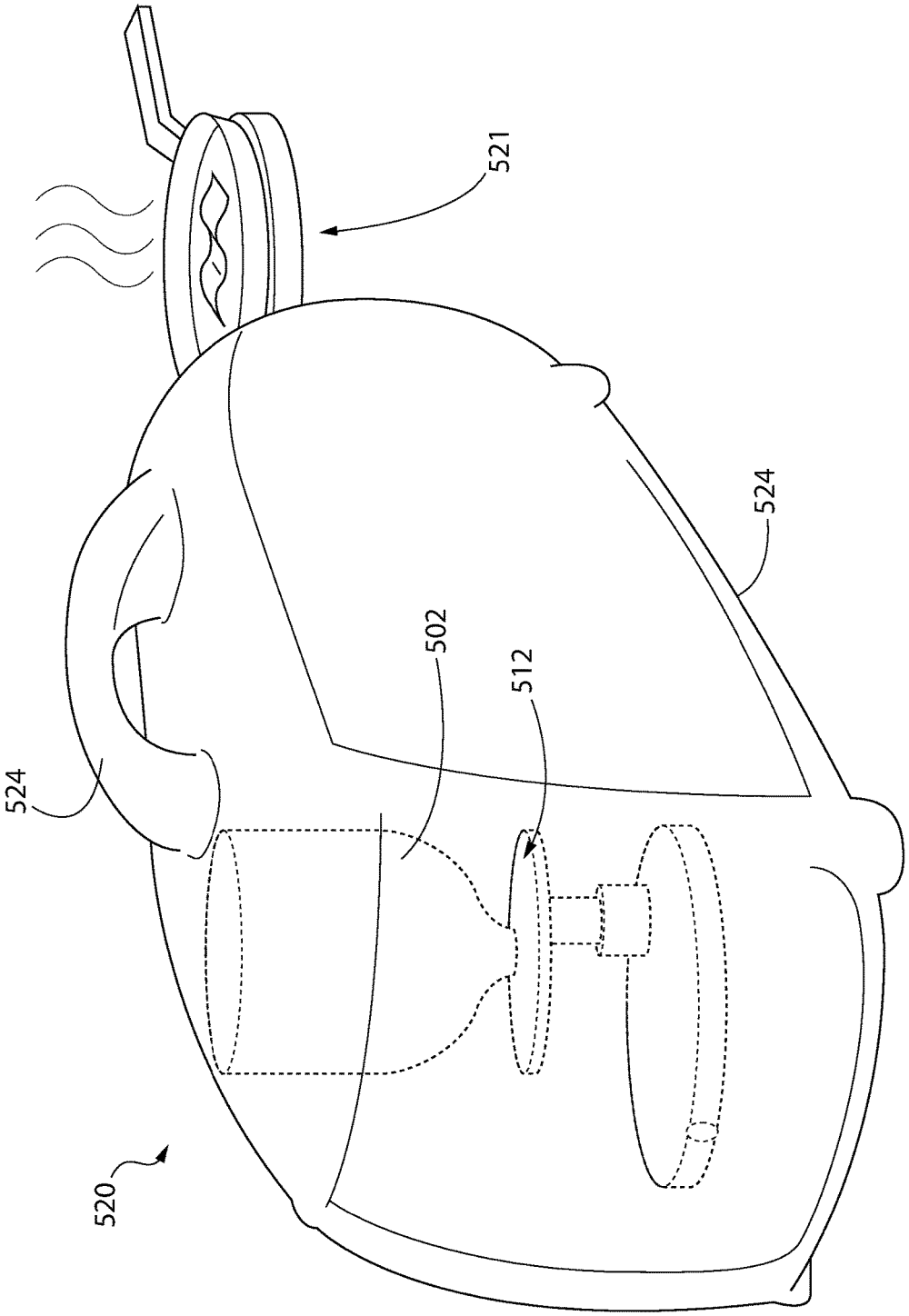


FIG. 106

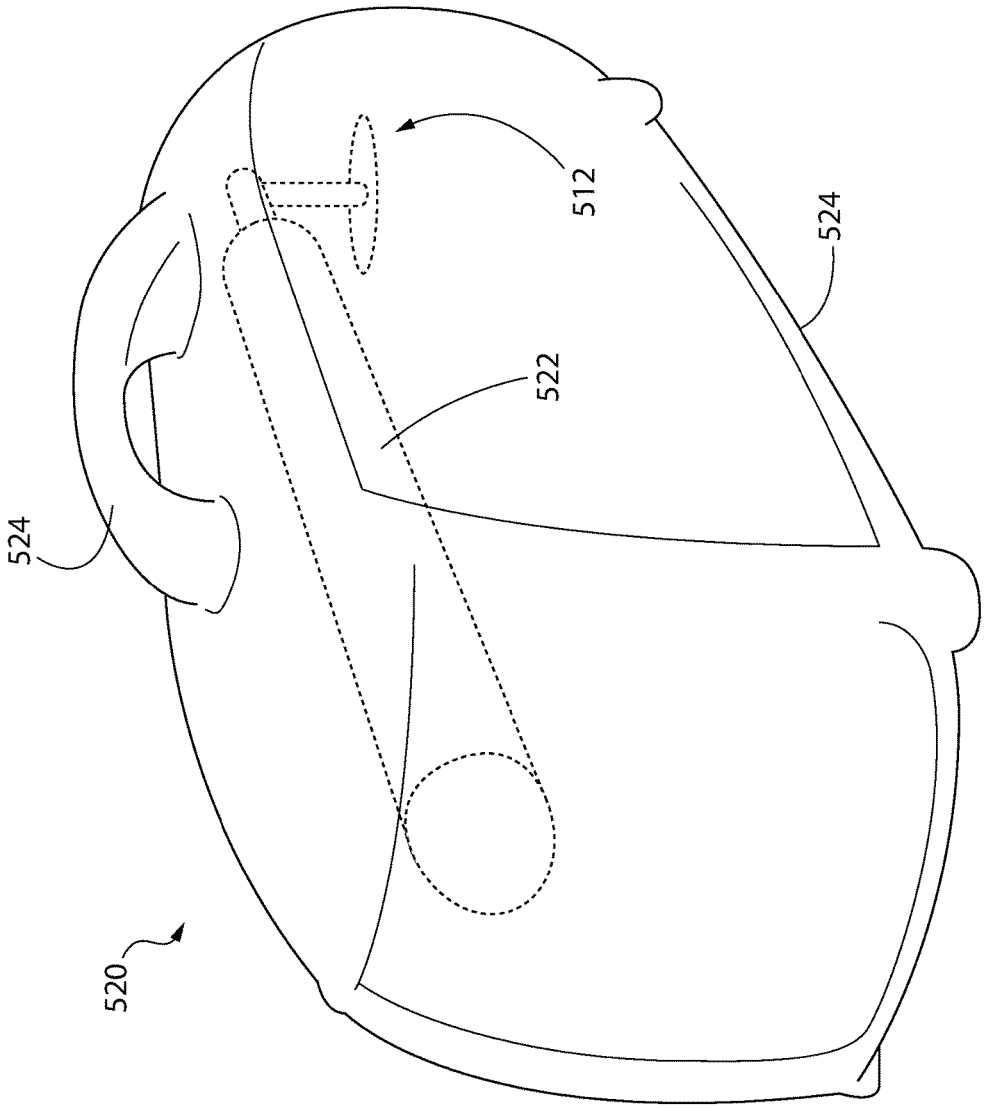


FIG. 107

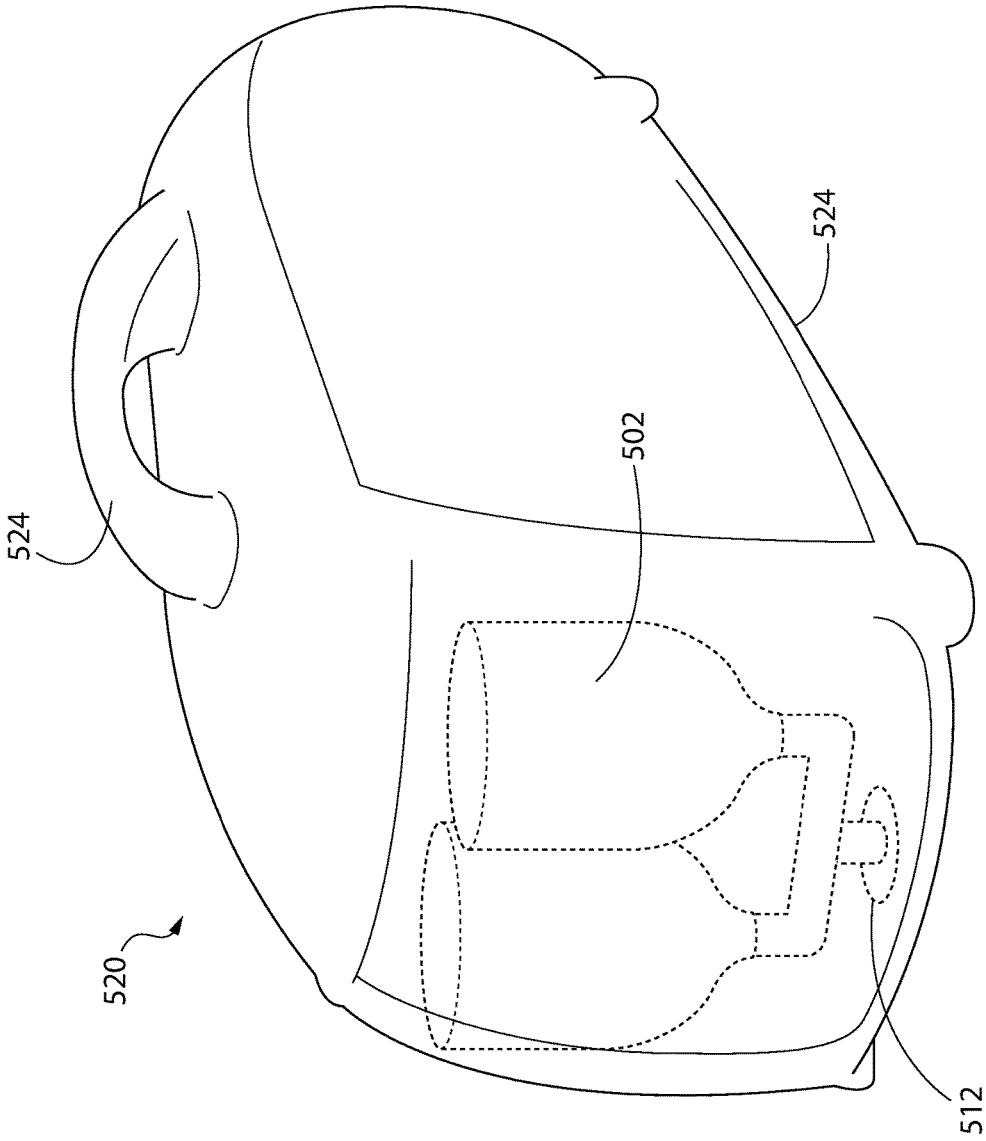


FIG. 108

PORTABLE GENERATORS

CROSS-REFERENCE TO RELATED APPLICATIONS

The present application is a continuation of U.S. patent application Ser. No. 14/169,739 filed Jan. 31, 2014, which claims the benefit of U.S. Provisional Patent Application Ser. No. US 61/760,054 filed Feb. 2, 2013, which are all hereby incorporated herein by reference in their entireties.

BACKGROUND OF THE INVENTION

The invention generally relates to engine-generator sets, for producing electric power, and more particularly to a highly customizable portable electric generators.

Portable electric generators produce a ready source of electricity for use in a variety of situations where electricity may not be available from the conventional power grid. Such situations may include power outages, remote locations without access to a nearby source of power from an electric utility, constructions sites, and instances where temporary power hookups from the utility are not practical or convenient for short periods of time.

A customizable portable generator system is desirable.

SUMMARY OF THE INVENTION

A portable generator system is disclosed that is configurable by a user and which may be assembled at the point of sale or later reconfigured afterwards to address changing needs. The portable generator system may include a versatile universal support frame that provides an opportunity for a user to select different sizes/types of engines and accessories which are conveniently interchangeable with a single frame. This may be accomplished through a generator frame that provides common accessory equipment interfaces adapted to accept a variety of accessories and/or readily alter the mounting arrangement of such accessories to vary the functionality of the portable generator system created by the user's preferences.

A universal frame generator system may include a universal engine mounting system that allows a plurality of engines having at least one different characteristic (e.g. KW electrical output capacity, fuel type, auxiliary equipment/accessories, etc.) to be interchangeably mounted to a single configuration and size generator frame via a common engine mounting interface.

A universal frame generator system may further include floor support system configured to mount different type floor supports including pedestal type legs and/or wheel assemblies at different locations which are detachably mounted to the frame via a common mounting system interface.

A universal frame generator system may further include a handlebar mounting system that is configurable to mount different sizes and types of handlebars in a plurality of different orientations.

A universal frame generator system may further include a power cord management system that is configurable to provide a variety of cord support arrangements.

In some example systems, the foregoing frame, engines, accessories, and other appurtenances disclosed herein may be assembled into a kit which presents custom options selectable by a user to equip the generator unit as desired. The kit can be shipped to a distributor or retailer which includes all the components necessary to assemble and create a fully functional generator unit that has been customized at the point of sale to meet the user's preferences. In one example, a generator unit assembly kit includes a universal frame, a plurality of engines having at least one different characteristic, a plurality of wheel assemblies, a plurality of frame support legs, and a plurality of handlebars.

An exemplary method for assembling a generator unit may include a user selecting an engine, at least two of a wheel assembly, pair of support legs, or combinations thereof from a generator kit, and mounting the foregoing components selected on the frame. The method may further include selecting at least one pair of handlebars and mounting the handlebars to the frame in at least one of two positions. In one example, the positions include a horizontal position and a vertical position.

The present application further discloses additional kits, accessories, and appurtenances which interface with and may be mountable on the generator and/or support frame disclosed herein.

The present application further discloses a portable generator with modular frame system. The frame system may generally include a plurality of corner members and connector plates which may be interconnected to collectively form an open space frame configured for supporting an internal combustion engine, related appurtenances, and accessories. The present application further discloses compact gaseous fuel power generators which may be operated from compressed liquid fuel canisters or cylinders. In one non-limiting example, the gaseous fuel may be propane.

BRIEF DESCRIPTION OF THE DRAWINGS

The features of some non-limiting examples will be described with reference to the following drawings, where like elements are labeled similarly, and in which:

FIGS. **1-7** show various views of one example of a portable generator and support frame;

FIGS. **8-14**A show various views of the generator and frame of FIGS. **1-7** with essentially all engine-related components removed except for the fuel tank to clearly show the frame;

FIG. **14**B is a detail of the frame with handlebar attachment taken from FIG. **14**A;

FIG. **15** is a cross-sectional view of a wheel assembly and the frame;

FIG. **16** A is a longitudinal cross-sectional view of a support leg assembly and the frame;

FIG. **16**B is a transverse cross-sectional view from FIG. **16**A;

FIGS. **17**, **18**A, and **18**B show a power cord management system and a detail thereof respectively;

FIG. **19**A shows a lifting member for transporting the generator;

FIG. **19**B is a detail of the frame with the handlebar assembly mounting arrangement taken from FIG. **19**A;

FIGS. **19**C-F show various examples of the handlebar and wheel assembly/support leg mounting options and positions;

FIG. **20**A is a perspective view of the handlebar assembly mountable to the frame;

FIG. **20**B is an exploded detail view from FIG. **20**A of the handlebar mounting clamp and position adjustment assembly and interface with the frame;

FIG. **21** is an exploded view of the handlebar assembly of FIG. **20**A;

FIG. **22** is an exploded view of the wheel assembly and mounting interface with the frame;

FIG. **23** is an exploded view of the support leg and mounting interface with the frame;

FIGS. **24** and **25** are rear and front perspective views respectively of an alternative configuration of the generator frame, showing a U-shaped pivotable/foldable front handlebar, alternative pivotable/foldable rear handlebars, and a laterally extending lifting bar attached to a top of the generator frame;

FIG. **25**A is a detail from FIG. **25** showing the pivotable front handlebar mounting assembly for a U-shaped handlebar;

FIG. **26** is a bottom rear perspective view of the alternative generator frame;

FIG. **26**A is a detail from FIG. **26** showing the pivotable front handlebar mounting assembly for the U-shaped handlebar;

FIG. **26**B is a detail from FIG. **26** showing the alternative pivotable rear handlebar mounting assembly;

FIG. **26**C is a detail from FIG. **26** showing the mounting assembly for the lifting bar attached to the top of the generator frame;

FIGS. **27** and **28** show rear and front elevation views of the alternative generator frame;

FIG. **29** is a side elevation view showing the U-shaped handlebar in the deployed operating position in dashed lines;

FIG. **30** is a top plan view showing the lifting bar and the U-shaped handlebar in the stowed position nested in the top of the generator frame;

FIG. **31** is an elevation view of an engine maintenance reminder system control panel usable with the generator;

FIG. **32** is an elevation view of an alternative configuration of a engine maintenance reminder system control panel;

FIG. **33** is a top perspective view of a portable generator with modular frame design constructed of corner members and connector plates;

FIG. **34** is a first side elevation view thereof;

FIG. **35** is a second side elevation view thereof of the side opposite the first side;

FIG. **36** is a third side elevation view thereof showing the control panel;

FIG. **37** is fourth side elevation view thereof of the side opposite the third side and showing the engine;

FIG. **38** is top plan view thereof;

FIG. **39** is a bottom plan view thereof;

FIG. **40** is a bottom perspective view thereof;

FIG. **41** is top perspective view of the modular frame design showing only the frame for clarity;

FIGS. **42-45** are side elevation views thereof;

FIG. **46** is a top plan view thereof;

FIG. **47** is a bottom plan view thereof;

FIG. **48** is a top corner perspective view thereof looking downwards through a top corner to a diagonal bottom corner;

FIG. **49** is a top corner perspective view thereof looking downwards through a different top corner to a diagonal bottom corner;

FIG. **50** is a bottom corner perspective view thereof looking upwards through a bottom corner to a diagonal top corner;

FIGS. **51-58** are various views showing a connector plate of the modular generator frame used to couple corner members together;

FIGS. **59-63** are various views showing the corner members;

FIG. **64** shows an assembled pair of connector plates positioned on a mounting tang of a corner member;

FIG. **65** is an exploded view of the corner member mounting tang and connector plate assembly including an accessory clip;

FIG. **66** is a partially assembled perspective view of two corner members being joined together by the mounting tangs via a pair of connector plates;

FIG. **67** is a top plan view of the permanent fuel tank of the portable generator with modular generator frame of FIG. **33**;

FIGS. **68-71** are side elevation views thereof;

FIG. **72** is a top plan view thereof;

FIG. **73** is a bottom plan view thereof;

FIGS. **74-81** show various views of the permanent fuel tank mounted in the modular generator frame;

FIG. **82** is an exploded view showing a removable portable fuel tank insertably mountable into the top of the permanent fuel tank;

FIG. **83** is a top perspective view of the portable fuel tank of FIG. **82**;

FIG. **84** is a bottom perspective view thereof showing the fluid outlet coupling for fluid connection to the permanent fuel tank;

FIG. **85** is a top plan view thereof;

FIG. **86** is a bottom plan view thereof;

FIGS. **87-88** are side views thereof;

FIGS. **89** and **90** are rear and front perspective views respectively of a fuel tank fastener;

FIGS. **91**, **92**, and **93** are rear plan, front plan, and side views respectively thereof;

FIG. **94** is a front (exterior) perspective view of a corner cap attachable to the corner members of the generator frame;

FIG. **95** is a rear (interior) perspective view thereof;

FIGS. **96** and **97** are front and rear plan views thereof;

FIGS. **98-101** are side views thereof from different perspectives;

FIG. **102** is an exploded view showing the corner cap and fuel tank fastener assembly;

FIG. **103** is a perspective view of a motor mount bracket;

FIG. **104** is a perspective view of a portable generator in the form of a backpack;

FIGS. **105-108** are perspective views of a portable generator fueled by compressed gas canisters or cylinders.

All drawings are schematic and not necessarily to scale.

DETAILED DESCRIPTION OF EXAMPLES

The features and benefits of apparatuses, systems, methods, and other innovations disclosed are illustrated and described herein by reference to non-limiting examples. This description of examples is intended to be read in connection with the accompanying drawings, which are to be considered part of the entire written description. Accordingly, the present disclosure expressly should not be limited to such examples illustrating some possible non-limiting combination of features that may exist alone or in other combinations of features; the scope of protection being defined by the claims appended hereto.

In the description of examples disclosed herein, any reference to direction or orientation is merely intended for convenience of description and is not intended in any way to limit the scope of the present invention. Relative terms such as "lower," "upper," "horizontal," "vertical,", "above," "below," "up," "down," "top" and "bottom" as well as derivative thereof (e.g., "horizontally," "downwardly," "upwardly," etc.) should be construed to refer to the orientation as then described or as shown in the drawing under discussion. These relative terms are for convenience of description only and do not require that the apparatus be constructed or operated in a particular orientation. Terms such as "attached," "coupled," "affixed," "connected,"

"interconnected," and the like refer to a relationship wherein structures are secured or attached to one another either directly or indirectly through intervening structures, as well as both movable or rigid attachments or relationships, unless expressly described otherwise.

FIGS. **1-7** illustrate a non-limiting example of a portable generator **20** according to the present disclosure. The generator **20** may include an engine **30**. The engine **30** may be an internal combustion engine, or various other engines or motors which may produce mechanical energy. The engine **30** may be powered by a fuel or resource, such as natural gas, diesel fuel, propane, gasoline, or various other fuels.

The generator **20** may additionally or alternatively include an alternator **32**. The alternator **32** may be an electromechanical device in mechanical communication and operably coupled with the engine **30**. The alternator **32** may include or use a rotating magnetic field with a stationary armature, a rotating armature with a stationary magnetic field, or a linear alternator. For example, the engine **30** may produce mechanical energy when operated which the alternator **32** may convert into electrical energy, such as without limitation alternating current.

Generator **20** may further include an onboard fuel tank **40** configured to store fuel which is supplied to the engine **30** by a fluid coupling such as without limitation a tube or hose. Generator **20** further includes a control panel **50** for operating and monitoring the generator, and to provide an electrical connection interface for power generated by the generating unit. The foregoing components of the generator **20** are mounted to a support structure such as frame **28**.

For convenience of further description, without limitation, generator **20** may be considered to have a front **21**, rear **22**, first and second lateral sides **23** and **24**, a top **25**, and bottom **26** (see, e.g. FIG. **1**). These designations are defined with generator **20** resting in a normal upright operating position on a horizontal support surface as shown. Generator **20** defines a longitudinal axis LA extending from front **21** to rear **22** along a centerline of the frame **28**.

Various examples of generator **20** may further include floor supports to raise and space frame **28** above the floor. The floor supports may comprise one or more sets of support legs **60** and/or wheel assemblies **80** for transporting the generator. In some frames, one or more pairs of handlebars **160** may further be provided to further facilitate transport of the generator **20**. These features will be further described herein.

Control panel **50** may be mounted on one of the lateral sides **23**, **24** of the frame **28** by brackets **54** (see, e.g. FIG. **13**) or other similar methods. The control panel may alternatively be located in other mounted positions. The control panel may include a plurality of controls **51** such as switches and indicators **51**, power outlets **52** which are electrically connected to alternator **32** and/or a power distribution panel, key start **53**, and other usual appurtenances.

FIGS. **8-14** show generator **20** with the fuel tank **40**. Engine **30** is not shown in these figures to more clearly show the frame construction.

The fuel tank **40** may include a top **44**, bottom **45**, and opposing lateral sides **43**. In one non-limiting example, top **44** of fuel tank **40** defines a forward sloping surface **41** and rearward sloping surface **42**. Both sloping surfaces **41**, **42** assist with shedding water to minimize accumulation and prevent ingress of water into the tank such as during refueling. The front and rear sloping surfaces **41**, **42** intersect linearly along a laterally extending peak **45** on the top of the fuel tank which defines the highest point on the tank. In some fuel tanks **40**, lateral sides **43** may also be sloped outwardly towards the sides **25** and **26** of the generator **20** to shed water.

In one example, a removable fuel cap **48** which operably covers a fill spout **47** may be disposed on one of the sloping surfaces **41** or **42** to facilitate filling the fuel tank **40**. In one example, the fuel cap may be located on rear sloping surface **42**. Locating the fuel cap **48** and spout **47** on a sloping surface creates better access and convenience for refueling to minimize spills as the user has to reach less far with a fuel can in contrast to flat topped fuel tanks with the fuel cap and fill spout located thereon. However, a flat topped fuel tank **40** may alternatively be provided in some configurations. The fuel cap **48** and fill spout **47** may be mutually threaded to provide a screwed arrangement, but is not limited to such configurations.

Fuel tank **40** may be supported independently by frame **28** via support flanges **49**. In other systems, the tank **40** may be supported by the engine-alternator assembly **30-32**.

The frame **28** may be configured and comprised of a first lateral side frame **100** and an opposing second lateral side frame **101** spaced laterally apart from side frame **100**. In one construction, without limitation, frame **28** may be a substantially generally tubular frame formed of tube or pipe having a circular cross section. Accordingly, in one example, lateral side frames **100**, **101** may be cylindrically tubular in construction and configuration as shown. In other possible constructions, frame **28** may be formed of square or rectangular tubes having a rectilinear cross sectional shape.

Lateral side frame **100** generally includes a front vertical member **102**, top horizontal member **105**, rear vertical member **103**, and bottom vertical member **104** which may be formed as a unitary structure or joined together by any suitable means to form a continuous structure. Similarly, lateral side frame **101** generally includes a front vertical member **107**, top horizontal member **109**, rear vertical member **106**, and bottom horizontal member **108** which may be formed as a unitary structure or joined together by any suitable means to form a continuous structure.

In some constructions, the horizontal and vertical members of lateral side frames **100**, **101** may be multiple separate pieces joined together by welding or brazing. In other constructions, the lateral side frames may be formed of a single tube having two ends and which is first bent to shape and then joined at the ends after forming a loop.

The vertical and horizontal members of lateral side frames **100**, **101** may intersect at four corner sections **110** which preferably are rounded or angled as shown in some examples to preclude catching the frame on various object when generator **20** is being transported. Each corner section **110** in one example may include a short straight section of tube **112** of various lengths disposed between two elbows **111**. In one example, the elbows **112** may be less than 90 degrees to avoid a squared corner section **110**. However, in other suitable but less preferred examples, some or all of the corners **110** may be square in configuration.

In one system, top horizontal members **105** and **109** may be angled between front vertical members **102**, **107** and rear vertical members **103**, **106** respectively to complement the shape of fuel tank **40**. This may be provided for not only aesthetic reasons, but also to help protect the fuel tank. Accordingly, when seen in side elevation view as in FIGS. **9** and **11**, lateral side frames **100** and **101** may have a generally asymmetric shape. In alternative configurations, top horizontal members **105**, **109** may have a straight or flat

horizontal shape between the front vertical members **102**, **107** and rear vertical members **103**, **106** as shown for example in FIGS. **24** and **25**.

The lateral side frames **100** and **101** may be structurally connected together by a front cross-piece **120** and a rear cross-piece **121** disposed proximate to the top **25** of generator **20**. Cross pieces **120**, **121** are longitudinally spaced apart along the longitudinal axis LA from front to rear, and extend laterally between the side frames **100**, **101** to assist with laterally stiffening the upper portion of frame **28**. In one example, the cross-pieces **120**, **121** may connected to corner sections **110** of each lateral side frame **100**, **101** by any suitable means such as without limitation welding or brazing. It will be appreciated that cross-pieces **120**, **121** may be coupled to other portions of each lateral side frame **100**, **101** in other examples. The cross-pieces **120**, **121** may be arranged substantially parallel to each other and perpendicular to lateral side frames **100**, **101**. In a certain construction, cross-pieces **120**, **121** are tubular in configuration having a round cross-section similar to the lateral side frames **100**, **101**. However, the cross-pieces **120**, **121** are not limited to circular tubular constructions and may have rectilinear tubular cross sectional shapes such as square or rectangular.

To help laterally stiffen the lower portion of frame **28**, a pair of longitudinally spaced apart cross-struts **130**, **132** may be provided that extend laterally between lateral side frames **100** and **101**. In one example, cross-struts **130** may be connected to bottom horizontal members **104** and **108** of the lateral side frames **100**, **101**. The cross-struts **130**, **132** may be arranged substantially parallel to each other and perpendicular to lateral side frames **100**, **101**. Cross-struts **130**, **132** may further be arranged substantially parallel to cross-pieces **120**, **121**. In one configuration cross-struts **130**, **132** may have a different cross-sectional configuration than the cross-pieces **120**, **121** such as without limitation rectilinear as further described herein for reasons which will become apparent. Cross-struts **130**, **132** may be connected to lateral side frames **100**, **101** by any suitable means including welding, brazing, mechanical fasteners, or other methods used in the art.

The frame **28** including lateral side frames **100**, **101**, cross-pieces **120**, **121**, cross-struts **130**, **132**, and some other structural brackets, accessories, or appurtenances attached or coupled to the frame may be made of any suitable material including metal or polymer. In one construction, the frame is made of steel and/or aluminum. Any suitable coating, finish, or topical treatment and texture may be provided.

In some systems, the frame **28** may be configured to provide a common interface configured and arranged to removably and interchangeably mount either legs **60** and/or wheel assemblies **80** to frame **28** for transport and raising the frame **28** above the floor. This provides a floor support system for generator frame **28** that is configurable and customizable to meet the preferences and needs of each user. For example, a user may elect four wheel assemblies **80** (see, e.g. FIG. **19**E), four legs **60** (see, e.g. FIG. **19**F), or a combination of legs and wheel assemblies mounted on either of the cross-struts **130**, **132**. As further described herein, both the legs and wheel assemblies have a common mounting arrangement and configuration advantageously allowing either wheel assemblies **80** or legs **60** to be interchangeably used at any of the four mounting locations on the cross-struts.

As shown in FIGS. **13**, **16**A-B, **22**, and **23**, cross-struts **130**, **132** in one arrangment are configured to provide four mounting locations on frame **28** for structurally supporting and attaching the wheel assemblies **80** or legs **60**. In addition, cross-struts **130**, **132** may further be configured for mounting and structurally supporting engine-alternator assembly **30-32** of the generator unit and associated appurtenances, as further described elsewhere herein.

Cross-struts **130**, **132** may have a non-tubular rectilinear cross-sectional shape. In one configuration, cross-struts **130**, **132** may each have the shape of a flanged structural C-channel including a U-shaped central section comprised of an axially extending web **134**, pair of legs **133** extending generally downwards and perpendicular from the web, and pair of lateral side flanges **131** extending generally outwards and perpendicular from the legs. Other suitable structural shapes, however, may be used.

FIG. **22** is a detailed exploded view of a wheel assembly **80** showing the mounting system interface with cross-struts **130**, **132**. Each wheel assembly **80** includes an axle **81** rigidly attached to a mounting plate **82** such as by welding, brazing, or other suitable means, a tire **83** having a hub **84** and wheel bearing **89** defining a through opening configured to receive a portion of the axle there through, washers **85**, and a cotter pin **86**. The tire **83** is mounted on one end of the axle **81** and mounting plate **82** is mounted on an opposing end. In one example, as shown, mounting plate **82** may have a central portion that is concavely shaped to complement the convex shape of axle **81**.

In one non-limiting exemplary construction, the axle **81** may terminate at or near the mounting plate **82** and does not extend across the lateral width of the frame **28** from side **24** to side **26**. Accordingly, each wheel may have its own separate axle **81** rather than an arrangement wherein two tires **83** may be mounted on opposing ends of a common axle. Advantageously, this provides two shorter and structurally more rigid axles **81** which helps prevent breakage rather than using a single long side-to-side axis. This may sometimes occur in instances where the generator **20** may be hoisted and transported via a forklift or other equipment.

With continuing reference to FIG. **22**, mounting plate in one configuration includes a pair of mounting holes **87** and a pair of angled mounting tabs **88**. Each pair of mounting holes **87** and tabs **88** are spaced apart across axle **81** as shown and arranged such that a mounting tab and hole are disposed on each side of the axle. Mounting tabs **88**, which may protrude upwards from the top of mounting plate **82**, have a somewhat L-shaped configuration with a substantially vertical leg **88***a* attached to mounting plate **82** and a substantially horizontal free leg **88***b* extending at an angle to the vertical leg. Mounting tabs **88** are configured to produce an interlocked relationship with cross-struts **130**, **132** when the wheel assemblies are mounted to the frame **28**. Accordingly, cross-struts **130**, **132** may each include a pair of spaced apart mounting holes **135** configured and arranged to be substantially aligned with mounting holes **87** and mounting tabs **88** when each wheel assembly **80** is aligned with and mounted on the cross-struts. Mounting holes **135** may be preferably located in flange **131** of cross-struts **130**, **132** Mounting holes **135** are configured and dimensioned to receive mounting tabs.

To mount a wheel assembly **80** onto frame **28**, the mounting plate **82** is first positioned below a cross-strut **130** or **132** with the mounting tabs **88** vertically aligned with their corresponding mounting holes **135** in the cross-strut on each side of the axle **81**. It should be noted that the distance between each tab **88** and its respective companion mounting hole **87** on each side of axle **81** in the mounting plate **82** is slightly larger than the distance between each pair of corresponding mounting holes **135** on each flange **131** of

cross-struts **130**, **132**. Accordingly, when the mounting tabs **88** are aligned with their corresponding mounting holes **135** in the cross-strut **130** or **132**, the companion mounting hole **87** (i.e. on the same side of axle **81**) for each tab will be intentionally slightly offset from and not perfectly concentrically aligned with the remaining mounting hole **135** on the cross-strut, for reasons which will become apparent.

Once the angled mounting tabs **88** are each aligned with their corresponding mounting holes **135**, the tabs are inserted upwards through the mounting holes (e.g. the inner mounting holes **135** on cross-strut **130** or **132**) to abuttingly contact the top of the mounting plate **82** with the bottom of the cross-struts **130**, **132**. The mounting plate **82** with attached axle **81** and tire **83** is then slid inwards in an axial direction along the cross-struts **130** or **132** to interlock the mounting tabs **88** with each opposed flange **131** on the cross-strut. This engages each mounting tab **88** with the cross-strut at the holes **135** such that the horizontal leg **88***b* of the tab slides over the top of flange **131** of the cross-strut thereby trapping the flange in the gap formed below the horizontal leg **88***b* and top of the mounting plate **82**. FIG. **16**A shows an analogous arrangement of an angled mounting tab **88** seated with cross-strut **130** or **132**.

Upon engaging the angled mounting tabs **88** with the cross-strut **130** or **132**, the remaining holes **135** in the cross-strut will now each become concentrically aligned with its corresponding mounting hole **87** in wheel assembly mounting plate **82**. A threaded fastener **136** is then inserted through each pair of aligned holes **135**, **87** and tightened with a threaded nut **137** to finish securing the wheel assembly **80** to the cross-struts **130** or **132** and frame **28**. Each tire **83** is mounted outboard of frame **28** when wheel assembly **80** is mounted on cross-struts **130**, **132**.

Since the wheel assembly **80** are detachably mounted to frame **28**, the foregoing assembly steps are simply reversed to remove the wheel assembly.

Legs **60** in one arrangement may have an identical mounting interface arrangement with frame **28** and cross-struts **130**, **132** as wheel assemblies **80** described above. Since the mounting holes **135** in cross-struts **130**, **132** are all identical in size, location, and arrangement as part of an interchangeable frame support system, a user may elect to interchangeably mount either wheel assemblies **80** or legs **60** at any of the four mounting locations on cross-struts **130**, **132** thereby providing an opportunity for customization.

Each support leg **60** may be formed of angled metal plate which is bent and/or welded or brazed together. In one construction, the leg may be steel or aluminum. Legs **60** may have any suitable configuration, but preferably should be shaped and constructed with sufficient strength and stiffness to support the generator **20**. In one configuration, leg **60** includes a substantially flat top mounting plate **61** which is oriented horizontally, two side plates **62** disposed at about 90 degrees to each other and attached to plate **61**, and a flat bottom plate **63** attached to side plates **62**. Plates **61** and **63** may be substantially parallel to each other. Side plates **62** may taper inwards from top to bottom to provide a wide support platform at top for engaging cross-struts **130**, **132**, and a narrower footprint at bottom for engaging the floor or other horizontal support surface.

In one construction, a pad **64** may be attached to leg **60** at bottom plate **63** and formed of a resilient material such as rubber or another elastomeric material to provide surface grip and vibration dampening. Holes **65** and **66** may be provided in bottom plate **63** and pad **64** respectively in some examples to receive a mounting fastener there through (not shown). Pad **64** engages the mounting floor or other surface on which the generator **20** will be located.

With continuing reference to the foregoing figures, legs **60** include angled mounting tabs **88** and mounting holes **87** which are configured, dimensioned, and arranged no mounting plate **61** to have the same spatial relationship as in mounting plate **82** of wheel assembly **80** as already described herein. This permits the legs **60** to be interchangeable with wheel assemblies **80**.

Legs **60** are mounted to cross-struts **130**, **132** in the same manner as wheel assemblies **80** described above; the mounting process not being repeated herein for sake of brevity. FIGS. **16**A and **16**B show legs **60** fully mounted and fastened to cross-struts **130**, **132**. Locking tab **88** appears protruding upwards through and above flange **131** of the cross-struts in the interlocked position.

In some systems, the frame **28** may be configured to provide a universal engine mount system that allows engines **30** having different characteristics to be interchangeably mounted to a single configuration and size generator frame. The different engine characteristics may include, without limitation, electrical output capacity (e.g. measure in Watts or KW), engine type (e.g. two-stroke, four-stroke), fuel type (e.g. gasoline, diesel), auxiliary equipment, and others. In one exemplary arrangement, without limitation, a common engine mount arrangement is provided which includes four frame engine mounts **140** fixedly attached to cross-struts **130**, **132**. A pair of laterally spaced apart engine mounts **140** are mounted on each of the cross-struts **130**, **132**. The engine mounts **140** are preferably each configured and arranged identically to provide a common interface with four mating commonly configured and arranged engine mount brackets **34** (see, e.g. FIGS. **1**-**7** and **16**A) that are provided on a plurality of engines **30** having at least one different engine characteristic. Preferably, without limitation, the spatial relationship between the engine mount brackets **34** are common to each of the plurality of different engines **30** to allow interchangeable use of the engines with a single frame **28** setup.

FIG. **16**A shows the frame engine mounts **140** and engine mount brackets **34** in greater detail. Each frame engine mount **140** includes a top plate such as angled top plate **142** configured for engaging an engine mount bracket **34** on the engine **30** and a bottom plate such as an angled bottom plate **143** configured for attachment to cross-struts **130**, **132**. Top plate **142** includes a substantially horizontal flat section **142***a* and a conjoined angled flat section **142***b* disposed at an angle to the horizontal section. The horizontal top section **142***a* may include an upward turned lip **142***c* to assist with placing and locating the engine mount bracket **34** on top plate **142**.

Bottom plate **143** of frame engine mount **140** may have a similar configuration as top plate **142** as shown in FIG. **16**A having a horizontal flat section **143***a* and angled flat section **143***b*. Bottom plate **143** may be spaced vertically below the top plate **142** in one arrangement. The horizontal section **143***a* of bottom plate **143** may be attached to cross-struts **130**, **132** while the angled section **143***b* is free and projects upwards from the cross-strut at an angle thereto as shown. In one construction, without limitation, top plate **142** and bottom plate **143** may each be made from a single metal plate that is bent to shape to produce the horizontal and angled sections.

With continuing reference to FIG. **16**A, the top plate **142** may be connected to bottom plate **143** by a connecting rod **144** extending between the angled sections of each plate **142** and **143**. In one arrangement, only the connecting rod **144**

supports the top plate **142** from the mounting plate **143** and cross-struts **130**, **132** in turn. This arrangement advantageously provides a flexible mount which assists with dampening engine vibrations transmitted between the engine and frame **28**. A motor mount bushing or spacer **145**, which in some constructions can be metal, may be mounted around rod **144** between top and bottom plates **142**, **143** for additional support.

Referring to FIG. **13** and particularly **16**A, frame engine mounts **140** may be pedestal-style engine mounts and include a threaded stud bolt **141** projecting upwards from top plate **142** of the engine mount. Stud bolt **141** is configured and dimensioned for insertion through a mating mounting hole **36** provided in engine mount bracket **34**. Stud bolt **141** may be vertically oriented in some examples as shown, or be disposed at an angle with respect to vertical. The orientation of stud bolt **141** will depend on the corresponding orientation of the mounting hole **36** in the engine mount brackets **34** and is not limited to any particular orientation or arrangement so long as the engine **30** may be securely mounted to the frame engine mounts **140**.

After the engine mount brackets **34** is slipped over the bolt **141**, a threaded nut is used to securely fasten the bracket to the top plate **142**. This process is repeated at all four motor mounting locations on frame **28** to complete installation of the engine **30** on the frame. In some constructions, to further provide additional engine vibration dampening and isolation, a resilient spring or bushing **145** (not shown) such as without limitation rubber or an elastomer may optionally be mounted between the top plate **142** and engine mount bracket **34** around rod **144**. Such engine vibration dampening devices are well known in the art without further elaboration.

It will be appreciated that engine mount bracket **34** may have numerous suitable configurations so long as the bracket structure includes a mounting hole **36** for receiving the engine mount stud bolts **141** on the frame **28**. Similarly, frame engine mounts **140** may have other suitable configurations. Accordingly, the systems are not limited by the exemplary engine mount brackets **34** or frame engine mounts **140** disclosed herein.

The frame **28** may be configured to provide a handlebar mounting system that is configurable and customizable to meet the preferences and needs of each user. For example, a user may elect a single handlebar **160** mounted horizontally to frame **28** in a wheelbarrow-type arrangement (see, e.g. FIG. **19**A), two horizontally mounted handlebars **160** (see, e.g. FIG. **19**D), a single handlebar **160** mounted vertically to frame **28** in a hand truck type arrangement (see, e.g. FIG. **19**C), or any combination thereof. As further described herein, the frame **28** and handlebar **160** are cooperative configured with an interface that provides both horizontal and vertical mounting options and multiple simultaneous handlebar mounting arrangements.

The handlebar **160** may include a first handle assembly **161** (e.g. left in referenced figures) and second handle assembly **162** (e.g. right in referenced figures) joined together by a laterally extending and elongated tie piece such as without limitation tie bar **166**. Tie bar **166** is preferably rigid in construction and have any suitable configuration such as a substantially flat bar as shown, tubular, and others. Preferably, the tie bar **166** is made metal such as without limitation steel or aluminum. Other suitable metals or other materials may be used to provide the intended structural functionality. Tie bar **166** is fixedly attached to the handle assemblies **161**, **162** such as by welding, brazing, or mechanical means to provide a stiff connection between the handle assemblies. In some configurations, tie bar **166** may be provided such as when each handle assembly **161** or **162** have sufficient lateral stiffness to avoid undue deflection and movement.

Each handle assembly **161**, **162** includes an elongated handle **163** having a proximal grasping end **164** and opposing distal mounting end **165** for connection to generator frame **28**. In one construction, handle **163** may be tubular having a generally round cross-section. However, other suitable configurations and cross-sectional shapes may be used such as handles having a rectilinear cross-sectional shape, solid round bars, etc. Preferably, handle **163** is constructed of a material having sufficient strength and thickness to allow a user to lift the generator **20** without bending or deforming the handlebar. In one example, handle **163** is preferably made of metal such as without limitation steel or aluminum. Other suitable metals or other materials may be used to provide the intended functionality.

In some configurations, handle **163** may include an angled section **168** to produce an axial offset between the grasping and mounting ends **164**, **165** as shown in FIGS. **20**A and **21**. This provides wider lateral spacing between the grasping ends **164** of the handle for a user. In other examples, handle **163** may be substantially straight between grasping and mounting ends **164**, **165**.

A grip **167** may be mounted on grasping end **164** of handle **163** to provide a comfortable grasping surface for a user. Grip **167** may have any suitable ergonomic configuration and be made of any type material including resilient polymeric materials to facilitate gripping. The surface of the grip **167** may be textured and/or include features such as ribbing, stippling, nubs, etc. to further facilitate comfort and grip.

Referring to FIGS. **14**B, **19**B, **20**A, **20**B, and **21**, mounting end **165** of handle **163** is configured to engage a clamp **170** mutually configured with and operable for attachment to generator frame **28**. In one example, clamp **170** may be horizontally spilt collar including a top half section **171** and bottom half section **172**. In one example, each half section has a half-tubular concave surface **173** with an arcuate curvature oriented transverse to longitudinal axis LA and a half-tubular concave surface **174** with an arcuate curvature oriented parallel to the longitudinal axis LA. The top and bottom half sections **171**, **172** together form a first split collar aligned with handle **163** which includes concave surfaces **173** and a second split collar aligned with a front or rear cross-piece **120**, **121** which includes concave surfaces **174**; the second collar being oriented perpendicular to the first split collar.

When the two half sections **171** and **172** are joined together, both concave surfaces **173** collectively define a tubular shape that defines a circular rear facing opening **175** or socket which is configured and dimensioned for receiving mounting end **165** of a handle **163**. When the two half sections **171** and **172** are joined together, both concave surfaces **174** collectively define a tubular shape that defines a circular laterally facing opening **174** or socket (e.g. left or right) which is configured and dimensioned for receiving a portion of front cross-piece **120** or a rear cross-piece **121** of frame **28**. The rear and lateral facing openings or sockets may have other configurations to match and receive handles or frames having other non-circular cross sectional shapes.

Referring again to FIGS. **14**B and **20**B, top and bottom half sections **171**, **172** of clamp **170** may be removably connected together in one example by any suitable means. A threaded fastener **169***a* (e.g. bolt/screw) and mating nut **169***b* may be used as a non-limiting example. To facilitate align-

ing the top and bottom half sections **171**, **172** for assembly, a tab **182** and slot **183** may be provided on each half section which fit together and interlock when the half sections are joined. It will be apparent that the tab **182** and slot **183** will alternate in relative position on the top and bottom half sections **171**, **172** for insertion of a tab in a slot and vice-versa. In one example, as shown in the referenced figures, the tab **182** and slot **183** may be formed in a peripheral edge portion of concave surfaces **174** closest to frame front cross-piece **120** or rear cross-piece **121**. Other suitable arrangements are possible.

Referring to FIGS. **14**B and **20**B, bottom half section **172** may include a positioning flange **190** including two arcuately spaced apart holes **191** and **192**. Holes **191** and **192** are preferably separated by an arc distance of 90 degrees apart. In one configuration, the positioning flange **190** forms a segment of or partial annularly shaped flange that extends around and is disposed at one end of concave surface **174** on the bottom half section **172** as shown. The flange **190** forms a substantially flat arcuate surface **193** which faces outwards towards on or the other of lateral side frames **100**, **101**. Flange **190** is rotatable in relation to and about front cross-piece **120** or a rear cross-piece **121** by rotating the handlebar **160**. It should be noted that in other configurations, positioning flange may alternatively be formed on top half section **171**. Either arrangement is suitable.

Frame **28** includes a locking flange **180** which cooperates with positioning flange **190** to lock the position of handlebar **160** in either a horizontal or upright vertical position (see, e.g. FIGS. **19**A and **19**C). In one example, a locking flange **180** may be positioned on frame **28** near each of four top corner sections **110** to provide flexibility for a user to mount handlebars **160** on either the front or rear end of the frame, or on both ends (see, e.g. FIG. **19**D). Locking flange **180** includes a single hole **181** that is fixed in position in relation to generator frame **28**. Hole **181** is selectively and concentrically alignable with either hole **191** or **192** of the positioning flange **190** by rotating the handlebar **160** between horizontal or vertical positions.

Referring to FIGS. **14**B, **19**B, and **20**B, a locking pin assembly **184** may be provided which cooperates with locking flange **180** and positioning flange **190** to selectively lock the handlebar **160** in the horizontal or vertical position. The locking pin assembly **184** includes an cylindrical body **185** and a locking pin **186** at one end. Pin **186** is configured and dimensioned to be insertable through hole **181** in flange **180** and hole **191** or **192** in flange **190**. Pin **186** is movable between a (1) locked position in which the pin projects inwards through holes **180** and **191**/**192**, or (2) an unlocked position in which the pin is retracted outwards from holes **180** and **191**/**192**.

An exemplary method for mounting a handlebar **160** on frame **28** will now be described with reference to **14**B, **20**A, **20**B, **21**, and **19**B. The top and bottom half sections **171**, **172** of two clamps **170** are first positioned over and under one of the front or rear cross-pieces **120**, **121**, respectively. The clamps **170** are selected and arranged so that the positioning flange **190** of each clamp **170** is located towards a lateral side frames **100**, **101** proximate to a locking flange **180**. The top and bottom half sections **171**, **172** are next brought together and engaged with the front or rear cross-piece **120**, **121** selected. The tabs **182** and slots **183** are used to guide proper alignment and joining of the top and bottom half sections **171**, **172**. The front or rear cross-piece is trapped between the concave surfaces **172** in the top and bottom half sections **171**, **172**. Threaded fastener **169***a* is then inserted through the top and bottom half sections **171**, **172** (see FIG. **20**B) and engaged with nut **169***b* which is partially tightened at this stage of assembly so that the clamp **170** will not slip off of the front or rear cross-piece **120**, **121**.

With continuing reference to the foregoing figures, the mounting ends **165** on each handle **163** are inserted into one of the clamps **170** through rear opening **175** and engaged with concave surfaces **173** on the top and bottom half sections **171**, **172**. Fastener **169***a* may then be further slightly tightened, but preferably not fully to lightly engage and hold handles **163** in the clamps **170** while still allowing the handlebar **160** to be rotated about the front or rear cross-piece **120**, **121** for the desired position of the handlebar.

The handle bar **160** with attached clamps **170** are then rotationally adjusted in position about the front or rear cross-piece **120**, **121** to concentrically align hole **181** in each locking flange **180** on the frame **28** (see FIG. **20**B) with one of the holes **191** (for horizontal handlebar mounting position) or **192** (for vertical handlebar mounting position) in each of the handlebar positioning flanges **190**. Once the holes are aligned, the locking pin **186** is inserted through the aligned holes to fix and lock the position of the handlebar **160**. The fastener **169***a* may then be fully tightened with nut **160***b* to firmly engage the front or rear cross-piece **120**, **121** and form a frictional fit between the clamps **170** and front or rear cross-piece.

It will be appreciated that the foregoing handlebar mounting process and sequence may be varied and still achieve the same result. Accordingly, equivalent mounting processes are possible and the systems are not limited to the foregoing sequence of steps described.

If the user desires to change the mounting position/orientation of the handlebar **160**, the clamps **170** may be slightly loosened by partially unscrewing the fasteners **169***a* in each clamp and retracting the locking pins **186** from the holes **180** and **191**/**192** in the locking flanges and positioning flanges. The handlebar **160** will now be rotatable about the front or rear cross-piece **120**, **121** to change the position of the handlebar from horizontal to vertical, or vice-versa.

FIGS. **24**-**30** illustrate an alternative configuration and features of a generator **20** and frame **28**. The generator **20**, frame **28**, support legs **60**, and wheel assemblies **80** may be essentially the same as already described herein for generator **20** and frame **28** with reference to FIGS. **1**-**24**. Instead of or in addition to handlebars **160** described herein, handlebars having an alternative configuration, construction, and functionality are provided.

The generator **28** includes a pivotably mounted front handlebar assembly comprising a handlebar **250** movably coupled to frame **28** near the front **21**. Handlebar **250** may have a generally U-shaped tubular body including two mounting portions **252** arranged substantially parallel to each other along the longitudinal axis LA and a grasping portion **253** arranged substantially perpendicular to and spanning between the mounting portions at one end **254** of the handlebar. Grasping portion **253** may be vertically offset from and positioned below the mounting portions **252**. In one arrangement, each mounting portion **252** may be mounted to one of the top horizontal frame members **105** or **109** about a pivot axis **251** oriented transversely to longitudinal axis LA. The mounting portions **252** may be mounted to the frame members about pivot axis **251** proximate to the free ends **255** of the handlebar **250**, and in one configuration is spaced inwards from the free ends. In one assembly, the pivot axis **251** may be formed by transversely mounted pivot members **256** such as pins or fasteners extending through mounting portions **252** of handlebar **250**

and top horizontal members **105** and **109**. In one configuration, the handlebar **250** may be disposed inside the top horizontal members **105** and **109**.

Handlebar **250** is pivotably movable with respect to frame **28** between a folded stowed position (see, e.g. FIGS. **24** and **25**) and an extended operating position (see, e.g. FIG. **29** shown by dashed lines). In the stowed position, handlebar **250** is nested in the top **25** of the frame between the longitudinally-extending top horizontal members **105** and **109**. The mounting portions **252** are oriented substantially horizontal in position. In the operating position, the grasping portion **253** of handlebar **250** is rotated forward about the pivot axis **251** to raise and locate the mounting portions **252** in a non-horizontal position. The handlebar **250** protrudes outwards from the frame for transporting generator **20**. In one configuration, the mounting portions **252** may be oriented in a generally upright position defined herein as being greater than 0 degrees (horizontal) and less than or equal to about and including 90 degrees (vertical). This arrangement allows the generator **20** to be transported in hand truck fashion with the handlebar **250** being upright and vertically or obliquely oriented with respect to the top horizontal members **105** and **109** of frame **28**.

To guide and limit the pivotable range of motion of the handlebar **250**, a flat guide plate **257** may be fixedly attached to one or both of each opposing top horizontal members **105** and **109**. Guide plate **257** includes an arcuately-shaped open guide slot **258** extending 90 degrees between a first and second closed end of the track. A spring-loaded position pin **259** is fixedly connected through the guide plate **257** and guide slot **258** to the handlebar **250** at or near one of the free ends **255** of the handlebar. The position pin **259** is arcuately movable in guide slot **258** with free ends **255** by rotating the handlebar **250**.

The ends **260** of the guide slot **258** may be diametrically enlarged and configured with the position pin **259** for mutual engagement to removably retain the pin **259** at either end location of the slot. The end **260** locations of guide slot **258** define the stowed and operating positions of handlebar **250**. To enable position pin **259** to move in the guide slot **258**, an enlarged operating end **261** of the pin may be moved and either pulled outwards (in the situation where pin **259** is biased inwards by the spring) or pushed inwards (in the situation where pin **259** is biased outwards). In one exemplary configuration, the pin **259** is inwards biased requiring an outward pull to release and move the pin. This releases the pin **259** from one of the ends **260** of the guide slot **258**, thereby allowing the pin to be moved through the slot to the other end location. Accordingly, an interlock formed between ends **260** and position pin **259** allow the handlebar **250** to be removably locked in either the stowed or operating positions.

Referring to FIGS. **24-30**, an alternative rear handlebar assembly including handlebars **270** is shown which is pivotably movable between a stowed position collapsed against the rear vertical members **103**, **106** of frame **28** and an operating position extending approximately horizontally outwards to the rear of the generator **20**. This allows generator **20** to be transported in wheelbarrow fashion. Each rear handlebar **270** includes a substantially U-shaped mounting clevis **271** on one end that may be movably mounted to rear vertical members **103**, **106** of frame **28** about a transversely-oriented pivot axis **274**. Each clevis **271** may include a pair of spaced apart ears **272** positioned on opposite sides of either vertical members **103** or **106**. The ears **272** are mounted to the rear vertical members **103**, **106** by a transversely mounted pivot member **273** such as a pin or fastener extending through the ears and vertical members as shown. In one configuration, the mounting clevis **271** and pivot members **273** are configured to retain the handlebars **270** in the stowed and operating positions via a frictional fit.

In one configuration shown in FIGS. **24-30**, generator **20** may include a lifting bar **280** transversely mounted to top horizontal members **105** and **109** of generator frame **28**. Each end **282** of the lifting bar **280** may be configured to engage a frame horizontal members **105** and **109**, which may be secured thereto with mounting fasteners **281** such as without limitation threaded fasteners and nuts. Other suitable means may be used to mount the lifting bar **280** to frame **28**, such as welding, soldering, rivets, etc. The lifting bar **270** extends under and is not operably connected to front handlebar **250** so as to not interfere with the pivotable movement of the handlebar. In one construction, lifting bar **280** may have a tubular metal construction for strength. Lifting bar **280** may be substantially V-shaped in one configuration forming an apex which is raised above than top horizontal members **105** and **109** of frame **28** for grasping manually or with lifting equipment.

In an alternative configuration shown in FIG. **19**A, the lifting member may be a lifting strap **210** which is attachable to a top portion of the frame **28** and spans between the lateral side frames **100**, **101**. The lifting strap may be formed of any suitable material including metal, polymeric high strength fibers, etc.

The generator frame **28** may be configurable to provide a power cord management system. As shown in FIGS. **17**, **18**A, **18**B, and **24-30**, two or more outwards projecting cord wrapping poles **200** may be provided which are mounted on generator frame **28** to allow a power cord **201** to be wrapped around the poles for convenient storage. In one arrangement, the poles **200** are preferably mounted on the top **24** of the frame **28** and project vertically upwards, and more preferably in some arrangements may be mounted on top horizontal members **105** and/or **109**. Other suitable mounting positions of wrapping poles **200** and orientations (e.g. horizontal or vertical) may be used. The poles **200** may be configured for attachment to frame **28** by any suitable means including separately or in combination clamping, fasteners, rivets, adhesives, etc. In various exemplary configurations, the poles **200** may be mounted to top horizontal members **105** and **109** of frame **28** and/or front handlebar **250**. A transversely oriented flanged end **202** may be provided having a width greater than the diameter of the poles **200** to prevent the power cord **201** from slipping off the poles. The poles **200** may be made of any suitable material including metal or polymers.

According to another aspect of the generator **20**, a maintenance reminder kit including a maintenance reminder module **290** may be provided to track and alert an operator in real time to the maintenance related engine operating parameters requiring attention. This module **290** essentially takes the guesswork out of separately tracking when routine maintenance needs to be performed. FIGS. **31** and **32** show two exemplary configurations of a maintenance reminder module **290**.

Maintenance reminder module **290** may comprise a microprocessor and appropriate electronic circuitry configured and operable to obtain, process, and display engine, electrical system, and/or auxiliary component maintenance-related operating information. In one configuration, maintenance reminder module **290** may communicate with the engine control system to download and process the desired operating parameter for extracting maintenance related data or information. In alternative or additional configurations,

maintenance reminders may be initiated and based on engine run time and preprogrammed thresholds operating times or maintenance intervals for initiating each maintenance operation (e.g. change oil, air filter, etc.).

Module **290** may include a faceplate **291** including a display **292**, a maintenance indicator array **293** comprised of one or more condition indicator lights, and an operating button **294**. The faceplate **291** may be mounted proximate to and/or on an electronics housing **295** housing the electronic components of the maintenance reminder module **290**, or the faceplate may alternatively be mounted on one part of the generator frame **28** and the electronics housing may be mounted on another different part of the frame distanced from and not mechanically coupled to the faceplate. Alternatively, the electronic components of the maintenance reminder module **290** may be incorporated and housed with the engine control system components and the faceplate **291** may be mounted on the frame at a suitable location, thereby eliminating a need for a separate electronics housing **295**. The faceplate **291** may be mounted to generator frame **28** at any suitable location. In one non-limiting arrangement, the maintenance reminder module **290** may be mounted on control panel **50**.

Referring to FIG. **31**, the indicator lights of the maintenance indicator array **293** may be illuminated buttons, bulbs, LEDs, etc. that include such maintenance-related alerts as without limitation "Change Oil," "Change Spark Plug," "Change Air Filter," and "Add Oil" in this non-limiting example. When an engine operating condition associated with one of the foregoing maintenance parameters reaches a predetermined set point or level based on run time or actual monitoring of engine operation, the indicator will become illuminated until cancelled by pressing the operating button **294**, which may function as a reset button. The display **292** may be an LCD, LED, or other suitable type illuminated display showing for example and providing a digital readout of the total running time of the generator **20** (e.g. service hours) from initial purchase and operation of the unit. The control circuitry may be configured so that the total running time cannot be reset with button **294**. It will be appreciated that additional and/or other engine maintenance indicators may be used to alert an operator to a required maintenance condition.

FIG. **32** shows a maintenance reminder module **290** in which the display **292** may provide a real time digital readout of a selected engine-related maintenance parameter. As opposed to alerting an operator when maintenance work may be immediately required, an operator can to track the status of maintenance related engine operating parameters to plan maintenance work in advance. The operating button **294** may be configured (in circuitry) to function as both a maintenance parameter select and a reset button. Repeatedly pressing button **294** sequentially cycles the display **292** through each maintenance parameter in a predetermined order. The indicator light associated with the maintenance parameter or information being displayed will be illuminated.

In one non-limiting exemplary configuration, the maintenance parameter tracked and displayed on display **292** may include "Power %" (available electric power), "Oil %" (remaining oil life), "Air Filter %" (remaining air filter life), "Spark Plug %" (remaining spark plug life, "Valve Adjustment %" (remaining valve adjustment life, and "Run hours" (total running time or service hours of the generator **20** from initial purchase and operation of the unit). Other or additional parameters may be tracked.

In one mode of operating the maintenance reminder module **290** of FIG. **32**, the maintenance parameter displayed may automatically change and loop through each parameter for predetermined period of time (e.g. 3 or more seconds). The maintenance parameter displayed may also be changed manually by repeatedly depressing the operating button **294**. When maintenance is required, the circuitry may be configured to cause the associated indicator light to flash or blink repeatedly. In some configurations, the circuitry may be configured to initiate the blinking mode when about 10% or less life is remaining for the blinking parameter providing advance notice. The associated remaining life may be concurrently displayed for the blinking parameter. In one configuration, preferably after maintenance has been performed, the operating button **294** may be held for a predetermined period of time (e.g. 3 seconds or other) to reset the blinking parameter to 100% life. The operating button circuitry may be configured so that holding the operating button **294** may not reset the total operating hours or run time.

It will be appreciated that numerous variations of the maintenance indicator array **293**, maintenance indicators provided, and programmed functionality are possible.

Modular Generator Frame System

FIGS. **33-103** depict various views of a portable generator **398** with modular generator frame system **300**. The frame system **300** may generally include a plurality of interconnected corner members **320** and connector plates **340** that collectively form an open space frame designated herein as generator frame **301**. The generator frame **301** is configured for mounting various components of a power generator, appurtenances, and accessories. In one configuration, generator frame **301** may support a permanent fuel tank **360**, a removable portable fuel tank **380**, engine **311**, handles **318** for lifting and maneuvering the generator, control panel **400**, and others appurtenances and accessories (e.g. tool holders, fire extinguisher holders, power cord wrapping poles, etc.) for forming a fully functional generator. Any suitable type of engine and fuel source may be used, including for example without limitation liquid fuels or gaseous fuels such as propane.

Generator frame **301** may include four lateral sides **302**, a top **303**, opposing bottom **304** and an open interior space **305** for housing the fuel tank **360**, engine **311**, and other accessories or appurtenances. The four lateral sides **302** may be joined together by four vertically extending corners **306** at the top **303** and bottom **304** of the frame. In one configuration, corners **306** may be disposed at a diagonal and oblique angle to the two adjoining lateral sides **302** as shown. Other corner configurations are possible.

Generator frame **301** may be considered to generally have the shape of a cube wherein the lateral sides **302** have a substantially similar height and width. The corners **306** may have a truncated shape in appearance created by the diagonal configuration of the corners. Accordingly, in one configuration, the generator frame **301** may appear to have an identical profile when viewed from either of the sides, the top, or the bottom. The diagonal corners **306** provide a generally octagonal shape (in top plan or side view) for generator frame **301** in one configuration.

The corner members **320** and connector plates **340** will now be described in greater detail with general reference to FIGS. **51-66**. Each corner member **320** may include a central body **321** and one or more outwardly and radially projecting mounting tangs **322** cooperatively configured to engage

locking elements on the connector plates **340** for assembling a complete frame **301**. Body **321** may be open in structure in one configuration defining a central opening **323** for weight reduction and to allow various appurtenances or accessories to be mounted and positioned therein and therethrough. The central opening **323** may have a complementary configuration to the perimeter shape of the central body **321**, or alternatively a different shape.

In one non-limiting example, body **321** may be generally triangular shaped with truncated corners forming six integral short and long segments as shown defining a three-sided corner structural element. The short segments **324** may be disposed adjacent a connector plate **340** at each corner region and the long segments **325** having a longer length span between two short segments **324**. The short and long segments **324**, **325** may have any configuration including polygonal and/or non-polygonal shapes. The short segments **324** define mounting portions **341** which include radially extending mounting tangs **322** for coupling to a connector plate assembly comprising a pair of coupled connector plates **340**, as further described herein. In one configuration, as shown, each corner member **320** includes three mounting portions **341** and mounting tangs **322** which may each be arranged approximately 120 degrees apart about a common center point **327** defined at the geometric center of the corner member body **321**. Each mounting tang **322** defines a mounting axis **330** which intersect at the common center point **327** of body **321**; the mounting tangs extending radially outwards from the center point along the mounting axis.

Mounting tangs **322** may have an elongated shape. The mounting tangs **322** include a free end **325** configured to lockingly engage a connector plate **340** and a fixed end **326** attached to or integrally formed with central body **321** of the corner member **320**. In one construction, without limitation, mounting tangs **322** may be integrally formed with body **321** as part of a single unitary structural unit that may be bent or otherwise formed into the shape shown. In one arrangement, mounting tangs **322** are obliquely disposed at an angle A**1** with respect to the corner member body **321**. This allows the creation of angled corners for the generator frame **301** in which the central body **321** of corner members **320** are disposed at an angle A**1** between 0 and 90 degrees to the connector plates **340** when mounted on the mounting tangs **322**. In some non-limiting examples, A**1** may be about 45 degrees.

For convenience of reference, the shape and features of generator frame **301** may be further described with respect to a three-dimensional X-Y-Z Cartesian coordinate reference system (see, e.g. FIGS. **33**, **36**, and **38**). The reference system includes X, Y, and Z axes and defines an X-Y, X-Z, and Y-Z reference planes.

The central body **331** of the corner members **320** may form four generally flat diagonal corners **306** of generator frame **301** each laying in a plane P**1** that may be disposed at angles A**2** and A**3** with respect to both the X-Y and Y-Z reference planes. In one non-limiting configuration, angles A**2** and A**3** may each be between 0 and 90 degrees. In some non-limiting examples, angles A**2** and A**3** may be about 45 degrees. In a similar manner, corner members **320** may further form four generally flat sloping top sides and corresponding bottom sides of generator frame **301** (extending in the X and Z directions) each laying in a plane P**2** that may be disposed at angles A**4** and A**5** with respect to both the X-Y and X-Z reference planes. In one non-limiting configuration, angles A**4** and A**5** may each be between 0 and 90 degrees. In some non-limiting examples, angles A**4** and A**5** may be about 45 degrees.

In one exemplary construction, corner members **320** including central body **321** and mounting tangs **322** may be formed from a single generally flat plate or sheet. Corner member **320** may therefore comprise a single unitary structural component. The corner members **320** may be made of suitable metal material having sufficient mechanical/structural strength and thickness, such as without limitation aluminum, steel, or other metallic materials. In one example, without limitation, corner members **320** may be made of aluminum sheet material having a thickness ranging from about and including 0.125 inches to 0.625 inches. In one example, without limitation, the thickness without limitation is about 0.125 inches. Corner members **320** may be made by any suitable fabrication process or combination of processes, including for example without limitation die-stamping, bending, casting, forging, machining, milling, drilling, etc.

Referring to FIGS. **51-58**, connector plates **340** may have an axially elongated and generally flat body including an axial centerline **348**, an outer surface **348**, inner surface **349**, opposing ends **350**, and opposing lateral sides **351**. Axial centerline **348** aligns with mounting axis **330** defined by mounting tangs **322** when the corner members **320** and connector plates **340** are assembled. When assembled, a pair of connector plates **340** may be coupled together with opposing inward facing inner surfaces **349** covering opposing outer and inner surfaces **331**, **332** of the mounting tangs of corner members **320**. Connector plates **340** may have a lateral width W**1** and axial length L**1** larger than the width. The lateral sides **351** may have any shape (in top plan view), including straight, convex, concave, angled, and others.

The corner members **320** and connector plates **340** may each include mutually configured and mating mechanical interlock features to form a strong, mechanically coupled generator frame **301**. The interlock features may include a lateral interlock feature and an axial interlock feature.

In one non-limiting example to form a lateral interlock feature, the connector plates **340** may include lateral locking tabs **343** configured to engage complementary configured lateral locking recesses **328** formed in lateral sides **333** of the mounting tangs **322** of each corner member **320**. The locking tabs **343** may project inwardly on the connector plates **340** towards and in a direction transverse to an axial centerline **348** defined by the connector. In one configuration, two locking tabs **343** may be provided. Locking tabs **343** may be offset from axial centerline **348**. In one configuration, locking tabs **343** may be integrally formed as unitary parts of opposing raised peripheral portions **347** formed on the inner surface **349** of connector plates **340**. Each raised portion **347** projects above the inner surface and is axially elongated extending for a distance parallel to centerline **348**. In one exemplary configuration, raised portions **347** may each extend axially from a point proximate to an end **350** of the connector plate **340** to a point proximate to the midline M**1** of the connector plate **340** (M**1** being defined as lying across a point midway between opposing ends **350** and dividing the connector into two halves). Accordingly, raised portions **347** may be arranged asymmetrically with respect to the midline M**1** of the connector plate **340** being on either one half or the other.

In some examples, locking tabs **343** may be formed alone on connector plates **340** without raised peripheral portions **347**. It will further be appreciated than in additional or alternative examples, the lateral locking tabs **343** may

instead be formed on the lateral sides of the mounting tangs **322** and locking recesses **328** may instead be formed in raised portions **347** on connector plates **340** providing a reverse construction. In either construction described herein, these lateral locking features assist with preventing axial withdrawal of the mounting tangs **322** from the connector plates **340** without disassembly of the inner and outer connector plates **340**.

The peripheral raised portions **347** and lateral locking tabs **343** collectively define a pair of inward facing and opposing bearing surfaces **342** engaged with mounting tangs **322**. The bearing surfaces **342** may have a complementary configuration to the peripheral shape of corner member mounting tangs **322**. This in turn further defines a mounting cavity **335** on the inner surface **349** of each connector plate **340** having a complementary configuration to the perimeter of mounting tangs **322**. The mounting tangs **322** substantially fill the cavity **335** when mounted therein, thereby forming a mating and interlocked keyed fit between the connector plates **340** and corner members **320**.

In one non-limiting example to form an axial interlock feature, the connector plates **340** may further include locking protrusions **345** arranged to engage complementary configured end locking recesses **329** formed on the mounting tangs **322**. The locking recesses **329** may be disposed and formed in the free ends **325** of the mounting tangs **322** and face outwards away from body **321** and center point **327**. In one arrangement, the locking protrusion **345** may be formed on inner surface **349** of connector plate **340** projecting outwards from and perpendicular to the inner surface in a direction transverse to the axial centerline **348**. The locking protrusions **345** may be aligned on the axial centerline **348** of the connector plates **340** in one exemplary arrangement as shown. In other possible arrangements, the locking protrusions may be offset from the centerline **348**.

Locking protrusions **345** may have a shape (in top plan view of inner surface **349** of connector plate **340**) that is configured to help resist axial withdrawal of the corner member mounting tang **322** in cooperation with the complementary shape of the end locking recesses **329**. In some examples, locking protrusion **345** may have a non-rectilinear polygonal shape. In one example, the locking protrusions **345** and locking recesses **329** may have a trapezoidal shape. In addition to preventing axial and sliding withdrawal of a mounting tang **322** from the connector plate **340**, the locking protrusions **345** help maintain the mounting tangs in proper position on a first connector plate until the second connector plate may be mounted thereto using fasteners **352** as described herein. This keeps mounting holes **346** and **334** in concentric alignment to permit insertion of the fasteners through the mounting tangs **322** and both connector plates.

The connector plates **340** may extend between and be operable to mechanically couple a spaced apart pair of corner members **320** together on the frame **301**. In one possible arrangement, a pair of connector plates **340** may be mounted back-to-back with the foregoing interlock features described herein on each connector facing inwards towards each other for linking the pair of corner members together. In one configuration, one of the connector plates **340** may be arranged opposite to and inverted 180 degrees in longitudinal orientation to its mating connector on the opposite side of corner member mounting tang **322**. In these foregoing positions, therefore, one each of the connector plates **340** may be placed on opposite sides of the mounting tang **322** of the corner member **320**, thereby trapping the mounting tang between the connector plates. It should be noted that the locking protrusion **345** on one connector plate **340** will actually be located between the raised portions **347** of the opposite connector plate **340**, and vice-versa. The mounting tangs **322** may be considered to form a male fastening element and the two assembled connector plates **340** a female fastening element configured to receive the male element at least partially therein.

To secure each connector plate **340** together on the mounting tang **322** of corner member **320**, one or more fasteners **352** may then be inserted through mounting holes **334** and **346** formed in the mounting tangs and connector plates, respectively. In one arrangement, each connector plate **340** includes two mounting holes **346** and each mounting tang **322** includes one mounting hole **334**. These holes **334**, **346** become concentrically aligned when the connector plates and tangs are positioned together on each side of the mounting tang **322**. Any suitable fasteners **352** may be used, including as an example without limitation threaded fasteners (e.g. bolts, screws, etc.) with mating nuts for detachable securement thereby allowing the frame to be easily assembled and disassembled. In one arrangement, two fasteners **352** may be provided for each corner member mounting tang-connector assembly **322/340**. One fastener **352** each may be located proximate to each end **350** of the connector plate **340**. In other possible examples, the connector plates **340** may be more permanently joined together with the mounting tang **322** trapped between them such as with rivets, welding, soldering, etc.

The connector plates **340** may be made of any suitable material having sufficient strength, stiffness, and thickness to form a substantially rigid structural member which resists bending and deflection without the application of undue force. In some non-limiting examples, the connector plates **340** may each be made of metal, such as without limitation aluminum, steel, or other. In one non-limiting exemplary construction, the connector plates **340** may be made of die-cast aluminum. In one example, each connector plate **340** may have a thickness ranging from about and including 0.125 inches to 0.625 inches. In one example, without limitation, the thickness may be about 0.25 inches. Connector plates **340** may be made by any suitable fabrication process or combination of processes, including for example without limitation die-stamping, bending, casting, forging, machining, milling, drilling, etc.

In one configuration, the outer surface **348** of the connector plates **340** may include an anti-rotation element **353** which cooperates with an accessory clip **354** attachable to the plate. Referring to FIG. **65**, the anti-rotation element **353** may be a rectilinear recess **359** in one configuration which engages a mating rectilinear mounting protrusion **355** formed on the clip **354** to prevent rotation of the clip with respect to the connector plate **340**. The clip protrusion **355** engages the walls formed around the recess to prevent twisting of the clip. The recess **359** may be disposed and formed around the mounting holes **346** on the connector plate **340** such that the holes extend through plate into the recess. In one exemplary configuration therefore, the accessory clips **354** may detachably retained on the connector plates **340** by the same fastener **352** used to secure the plates to the mounting tangs **322** of the corner members **320** via a through holes **357** formed in the clip.

In one possible configuration, the accessory clip **354** may include a pair of lateral extensions **358** each containing attachment apertures **356** of any suitable shape configured to engage a mounting feature of the accessory (e.g. straps, clips, fasteners, etc.). The lateral extensions **358** may be oriented 180 degrees apart and project laterally in opposing directions from the clip mounting protrusion **355** and fas-

tener **352** in one arrangement. The accessory clip **354** may be formed of metal such as aluminum, steel, titanium, etc. as some non-limiting examples for strength. It will be appreciated that the accessory clips **354** and attachment apertures **356** may have numerous possible configurations and shapes depending the design of the accessory to be attached.

The corner members **320** and connector plates **340** advantageously allow a light-weight generator frame to be shipped in an unassembled "knock down" condition to reduce packaging size requirements and shipping costs. The frame may then be assembled at the final destination which may be a point of sale, job site, or other. In addition, the modular frame system disclosed herein provides the opportunity to customize the frame and generator system in various ways to meet the end user's preferences.

An exemplary method for assembling a generator frame will now be described. The method may begin by providing a kit including a plurality of corner members **320** each having three mounting tangs **322**, connector plates **340**, and fasteners such as threaded fasteners **352**. Two corner members **320** and two connector plates **340** may first be selected from the kit, or alternatively may be selected separately later during the assembly process. A mounting tang **322** of a first corner member **320** may be positioned on and abuttingly engaged with the inner surface **349** of a first connector plate **340**. More particularly, the mounting tang **322** may be positioned and inserted into the complementary configured mounting cavity **335** of the selected first connector plate **340**. The first connector plate **340** may be located on a flat work surface to facilitate assembly.

A mounting tang **322** of a second corner member **320** may then be positioned on and abuttingly engaged with the inner surface **349** of the first connector plate **340** opposite the half having the mounting cavity **335** and mounting tang **322** of the first corner member **320** already in position. The locking protrusion **345** of the first connector plate **340** may be inserted into the end locking recess **329** in the mounting tang **322** of the second corner member **320** during placement of the tang.

With mounting tangs **322** of the first and second corner members **320** positioned on the first connector plate **340**, the second connector plate **340** is selected and positioned over the first connector with the inner surface **349** facing the inner surface **349** of the first connector. The second connector is rotated and inverted in orientation with respect to the first connector so that the mounting cavity **335** lies on an opposite half of the second connector than on the first connector. While maintaining this position, the second connector plate **340** is placed against the mounting tangs **322** of both the first and second corner members **320** already engaged with the first connector plate **340** as described above. The mounting tang **322** of the second corner member **320** may be positioned on and abuttingly engaged with the inner surface **349** of the second connector plate **340**. More particularly, the mounting tang **322** may be positioned and inserted into the complementary configured mounting cavity **335** of the second connector plate **340**.

The mounting tang **322** of a first corner member **320** (already inserted in the mounting cavity **335** of the first connector plate **340**) may be positioned on and abuttingly engaged with the inner surface **349** of the second connector plate **340** opposite the half having the mounting cavity **335**. The locking protrusion **345** of the second connector plate **340** (facing inwards) may be inserted into the end locking recess **329** in the mounting tang **322** of the first corner member **320** during placement of the tang. The mounting tangs **322** of the first and second corner members **320** are now clamped and trapped between the first and second connector plates **340** positioned on opposite sides or faces of the mounting tangs. To complete the connection, a fastener **352** is inserted through each of the two mounting holes **342** in the first and second connector plates **340** and through the mounting holes **334** of the first and second corner member mounting tangs **322** and secured with a threaded nut. The first and second corner members **320** are now mechanically coupled together.

To complete assembly of a full generator frame **301**, additional corner members **320** are then mechanically connector to each of the remaining two mounting tangs **322** of the first and second corner members in a similar manner to that described above. The foregoing process is repeated until a total of eight corner members **320** are interconnected using additional connector plates **340** and fasteners **352**. The completed and assembled frame is shown for example in FIG. **41-50** depicting the completed frame only (without engine and appurtenances) for clarity.

It will be appreciated that generator frame **301** may partially assembled during the assembly process to provide greater access for mounting individual components thereon. For example, the bottom half of the generator frame **301** may be assembled and then the motor mounted on the motor support. The top half of the generator frame **301** may be assembled separately and then the permanent fuel tank **360**, control panel **400**, etc. can be mounted. The bottom and top halves may then be joined together via the four vertical connector plates **340**. Numerous assembly variations are possible to maximize access and speed for assembling the complete generator. Advantageously, an assembled generator frame may be readily dismantled partially or completely to replace engine or other components, or to add accessories by simply removing fasteners **352** from the appropriate connector plates **340**.

Referring to FIGS. **67-82**, the permanent fuel tank **360** includes a top **361**, bottom **362**, lateral sides **363** extending between the top and bottom, and corners **367** extending between adjacent sides. The upper portions of lateral sides **363** may be downwardly sloping or angled in one configuration to complement the orientation of the connector plate **340** assemblies disposed laterally adjacent to the sides in the generator frame **301**. In some configurations, adjoining lower portions of lateral sides **363** below the sloped upper portion may be vertically oriented. Corners **367** extend between adjacent lateral sides **363** and may be configured to complement the orientation of corner members **320** disposed laterally adjacent to the fuel tank corners **367** in the generator frame **301**. Corners **367** may be may be arranged on a diagonal (i.e. angled between 0 and 90 degrees) with respect to the adjacent lateral sides **363**. In one configuration, fuel tank corners **367** may be downwardly sloping or angled defining an outward and upward facing flat diagonal surface **368**. Fuel tank **360** may have a shape selected to maximize volumetric storage capacity while conforming at least part to the angular or sloping shaped parts of the generator frame **301** where the fuel tank is mounted. In one non-limiting configuration, fuel tank **360** may have an octagonal shape in top plan view. Other suitable shapes may be used.

The permanent fuel tank **360** may further include a fill spout **373** disposed on one of the four corners **367** and a removable closure cap **374**. Cap **374** may be configured to threadably engage the spout **373** which may be externally threaded.

In one configuration, fuel tank **360** may include a recessed well **364** formed in the top **361** of the fuel tank for removable

storage of a portable fuel tank **380**. The well **362** may be complementary configured with the shape of the portable fuel tank **380** (in top plan view) to form a nested fuel tank arrangement. The portable fuel tank **380** and cooperative mounting with permanent fuel tank **360** is further described herein.

Fuel tank **360** may be made of any suitable material approved for the storage of liquid fuel. In example, the fuel tank **360** may be made of a fuel-grade molded plastic material. Other suitable materials including metal may be used.

Fuel tank **360** may be mounted to any suitable part of generator frame **310**. In one arrangement, the fuel tank **360** may be mounted near the top **303** of generator frame **301** and extends horizontally across the frame between all four lateral sides **302** substantially filling the upper third of the interior space **305** of the frame. The fuel tank **360** may extend from a location inside of and proximate to each of the four horizontal connector plates **340** that form the top half of the generator frame **301**. In one arrangement, the top **361** of fuel tank **360** may be flush with or slightly recessed below the top **303** of the generator frame **301** for protection. Accordingly, the fuel tank **360** may not protrude substantially above generator frame **301**, if at all in some configurations.

In one arrangement, permanent fuel tank **360** may be mounted to and suspended from generator frame **301** at the corner locations of the generator frame **301**, and more particularly near the top portions of the corners **306** at top corner members **320**. The flat diagonal surface **368** at corners **367** may include a mounting element **371** comprised of a threaded recess **369** and centrally-located cylindrical protrusion **370** disposed therein. Recess **369** and protrusion **370** may each be circular in one configuration. A circular depression **372** may be formed in corner **367** around recess **369** in some configurations, for reasons described elsewhere herein.

To mount the permanent fuel tank **360**, a fuel tank mounting assembly comprised of a corner cap **430** and fuel tank fastener **420** may be provided. Referring to FIGS. **89-102**, the corner cap **430** of the assembly may have a generally truncated triangular shaped body comprising three radially extending wings **440** to complement the shape of corner members **420** to which the cap may be attached. The wings **440** may further be slightly arcuately shaped in an inwards direction to complement the contour of the corner members **320** to which they are mounted. Corner cap **430** includes an outer surface **431**, opposing inner surface **432**, lateral sides **437**, raised plateau protuberance **434** disposed on the inner surface, and a centrally located through hole **433**. In one configuration, protuberance **434** has a peripheral shape selected to complement the peripheral shape of central opening **323** and is insertable into the opening for securing the corner cap **430** to the corner member **320**. Protuberance **434** and central opening **323** may therefore have a generally triangular shape. The protuberance **434** prevents the cap **430** from rotating with respect to corner member **320** via mutual engagement between the polygonal-shaped sides of the protuberance with mating surfaces surrounding the central opening **323** of the corner member. The hole **433** may be configured to insert the shank of fastener **420** therethrough for engaging a mounting element **371** on fuel tank **360**, as further described herein. A diametrically enlarged and inwardly protruding boss **433** may be formed around the hole **433** on the raised plateau-shaped protuberance **434** which may be configured for insertion into the depression **372** formed on the fuel tank corners **367**. The interaction and engagement of the boss **433** with depression **372** helps align and center the corner cap **430** on the corner member **320** to align hole **433** with threaded recess **369** to facilitate proper insertion of fastener **420**.

The fuel tank fastener **420** may include a head **421** and threaded shank **422**. In one configuration, shank **422** may be generally cylindrical in shape having one end **423** adjoining head **421** and a substantially flat terminal end **424** to preclude the possibility of puncturing fuel tank **360**. In one arrangement, terminal end **424** may include an open circular socket **425** configured for receiving cylindrical protrusion **370** of the fuel tank mounting element **371** when the tank is mounted to the generator frame **301**. It will be appreciated that in alternative examples, both socket **425** and protrusion **370** may be omitted wherein the threaded shank is solid and engages mating threaded recess **369** on the fuel tank **360**.

The exposed outer surface **426** of head **421** may include an operating feature **428** configured for engaging a mutually configured working end of a tool or key used to rotate the fuel tank fastener **420**. Any suitably shaped operating feature **428** and tool may be used.

The process for mounting the permanent fuel tank **360** to generator frame **301** will now be briefly described. In one example, without limitation, the fuel tank **360** may be mounted to the generator frame **301** using mounting element **371** at three corner members **320** to prevent twisting of the tank with respect to the generator frame. It will be noted that the remaining corner member **420** includes provisions for the fuel tank spout **373** and cap **374**. Fuel tank **360** is positioned inside the top **303** of frame **301**, as shown for example in FIG. **74**. At one corner, the corner cap **430** may be positioned on the corner member **320** to insert protuberance **434** into central opening **323** of the corner member. Boss **433** on the inside of cap **430** is inserted into depression **368** on the fuel tank **360**. Hole **433** in cap **430** is concentrically aligned with threaded recess **369** on fuel tank **360**. Fuel tank fastener **420** is inserted through corner cap **430** (i.e. hole **433**) and rotated to engage threaded shank **422** with mating threaded recess **369** on the fuel tank. The fastener **420** is rotated and tightened which draws the tank **360** and cap **430** together along the centerline axis of the fastener shank **422** with the corner member **320** being interspersed in between. The cap **430** is compressed against the outer surface of the corner member **420** between the head **421** of fastener **420** and corner member and the fuel tank corner **367** on the interior of the generator frame **301**. In one configuration, without limitation, the outer surface **431** of the corner cap **430** may include a recess **438** formed around hole **433** and the fastener head **421** may be seated in the recess and substantially flush with the outer surface **431**. In other configurations, the fastener head **421** may protrude above the outer surface **431** of the corner cap **430** either with or without a recess **438** if provided.

In one example, corner cap **430** and fuel tank fastener **420** may be formed of a suitable strength plastic material in addition to the fuel tank **360** and corner mounting elements **371**. Accordingly, a metal-free fuel tank mounting system may therefore be provided. This arrangement advantageously reduces weight and minimizes or eliminates interaction of the plastic fuel tank with metal edges to reduce wear or puncture of the tank.

Referring to FIGS. **41**, **42**, **46**, and **48**, the corner cap **430** associated with the fill spout **373** and closure cap **374** of the permanent fuel tank **360** may include a diametrically enlarged circular fill spout opening **439** configured to receive the fill spout. The fill spout **373** may project outwards through the opening **439** and the closure cap **374** is

attached to the fill spout **373** at the exterior of the corner cap **374**. In one configuration, the fill spout **373** may be configured to engage and retain the corner cap **430** in position on the generator frame **301**. In one non-limiting example, the fill spout **373** may include secondary external threading to attach the corner cap **430**. It will be appreciated that the fuel spout corner cap **430** may be configured similarly to the corner caps **430** described above associated with mounting the permanent fuel tank **360** with exception of the larger central fill spout opening **439** in lieu of the smaller fastener hole **433** and boss **435**.

According to another aspect of the generator **398**, a removable portable fuel tank may optionally be provided that is configured for detachable mounting in the modular generator frame system **300**. The portable fuel tank **380** advantageously extends the volumetric capacity and runtime of generator **398**. The fuel tank **380** may be configured for detachable coupling to the generator **398** so that the fuel tank **380** may be dismounted and removed without undoing fasteners or other mechanical coupling devices. This allows rapid change-out of portable fuel tanks in the modular generator frame system **300**. In one configuration, as described below, the portable fuel tank **380** may be configured for mounting directly to and is supported by the permanent fuel tank **360**.

Referring to FIGS. **83-88**, the portable fuel tank **380** includes a top **381**, bottom **382**, lateral sides **383** extending between the top and bottom, and corners **384** extending between adjacent sides. In one configuration, the sides **383** may be substantially flat. Corners **384** may be may be arranged on a diagonal (i.e. angled between 0 and 90 degrees) with respect to the adjacent lateral sides **383** defining a flat diagonal surface **392**. Fuel tank **380** may further include a compact handle **385** formed integrally with the body of the tank. In one example, a lateral portion of the handle may form one of the four lateral sides **383** of the fuel tank and an angled portion may form at least in part one of the corners **384**. This arrangement provides a handle without substantially altering the octagonal configuration or size of the fuel tank **380**.

The removable portable fuel tank **380** may further include a fill spout **386** for refilling the tank and a closure cap **387** threadably engaged with the spout. In one configuration, the fill spout **386** may be disposed on one of the four corners **384** of the fuel tank **380**. The corner **384** having the fill spout **386** may be configured and dimensioned such that the fill spout and cap **387** when in place do not extend laterally beyond either adjacent lateral side **383** to maintain a substantially uniform perimeter dimension and profile for insertion into the well **364** of the permanent fuel tank **360**.

Portable removable fuel tank **380** may further include a fluid coupling **391** allowing the tank to be fluidly connected to the permanent fuel tank **360**. Advantageously, the provides the permanent fuel tank **360** with two potential refueling connections and options. In one configuration, the fluid coupling may include an assembly comprising a tubular outlet nozzle **389** fluidly connected to a one-way valve **388**. Valve **388** may be configured to allow liquid fuel to flow outwards from the portable fuel tank **380** to permanent fuel tank **360**, but not to re-enter in a reverse direction back through the nozzle **389**. The nozzle **389** and valve **389** may be disposed in the bottom surface on the bottom **383** of the portable fuel tank **380**. In one arrangement, nozzle **389** may be located in the geometric center of the portable fuel tank **380**. To protect the nozzle **389** from damage during transport and refilling, the nozzle may be disposed in a recess **390** formed in the bottom **383** of the fuel tank **380**. Nozzle **389** may therefore be configured to not protrude above the bottom surface on the bottom **382** of the fuel tank **380**. The recess **390** may be circular in shape in one non-limiting configuration; however, other shapes may be used. Nozzle **389** and valve **388** may be formed of plastic in one construction or a combination of plastic and metal.

The outlet nozzle **389** of portable fuel tank **380** may be configured and operable to engage a mating fluid coupling **365** formed in the top **361** of the permanent fuel tank **360** to place the two tanks in fluid communication for transferring fuel from the portable tank to the permanent tank. In one configuration, fluid coupling **365** may comprise a raised tubular inlet coupling **366** which projects upwards from the recessed flat surface portion of the top **361** of fuel tank **360** inside the well **364**. Coupling **366** may be tubular shape. The coupling **366** and outlet nozzle **389** of the portable fuel tank **380** may be cooperatively configured and arranged so that the nozzle **389** may be inserted into the coupling. Accordingly, the inlet sleeve **366** may have an internal diameter sized slightly larger than the outer diameter of the outlet nozzle **389**. In one configuration, outlet nozzle **389** and inlet sleeve **366** are cooperatively configured to allow the nozzle to be axially and slidably inserted into the sleeve. In operation accordingly, the portable fuel tank **380** may be axially inserted downwards into the open top well **364** of the permanent fuel tank **360**. This motion would slidably insert outlet nozzle **389** into inlet sleeve **366** thereby fluidly coupling the portable and permanent fuel tanks **380** and **360** together. This allows the contents of portable fuel tank **380** to be transferred via gravity to the permanent fuel tank **360**. Advantageously, the total volumetric fuel capacity of generator **398** is therefore increased by providing one or more portable fuel tanks **380** which can be interchanged.

The portable fuel tank **380** may have a volumetric capacity of at least 10% of the permanent fuel tank **360**. For example, in one representative non-limiting example, the permanent fuel tank **360** may have a capacity of about 11 gallons and the removable portable fuel tank **380** may have a capacity of about 2 gallons. Any suitable permanent and portable fuel tank capacities may be used.

Referring to FIGS. **36** and **40**, the generator control panel **400** may include an emergency stop push button **401** which functions as a single readily accessible kill switch to stop the engine. In one example, the stop button **401** may be generally circular in shape. In one arrangement, the stop button **401** may be disposed in the center of a rotary operating dial **402** configured and operable to change the operating state of the motor (e.g. fuel, choke, run, start, etc.) to simplify the starting sequence. The push button **401** is actuated by depressing the button inwards towards the control panel **400**.

Referring to FIGS. **39**, **40** and **103**, the generator frame **301** may include a motor mount bracket **410** configured for attaching the motor mounts **414** of engine **311**. In one configuration, the motor mount bracket **410** may be an elongated and structurally stiff element formed of a suitable metal having two opposing mounting ends **411** configured for attachment to opposing connector plates **340** on the bottom of the generator frame. Bracket **410** may include a center section **412** extending between the ends **411** having a substantially horizontal position when attached to generator frame **301** for positioning and mounting the engine **311** thereon. The center section **412** may include spaced apart mounting holes **413** (e.g. two, four, etc.) configured for receiving threaded motor mounts of engine **311** therethrough having a matching layout pattern.

The mounting ends **411** of bracket **410** may be configured to engage the connector plates **340** and the same fasteners

352 used to connect each pair of connector plates together. Accordingly, in one arrangement the ends 411 may each include two attachment holes 415 for engaging both fasteners 352 of each opposing connector plate assembly. The ends 411 may have a substantially flattened shape and be upturned at an angle or slope complementary to the angular or sloped orientation of the connector plates 340 to which the bracket 410 is attached. The bracket 410 may bridge or span between the connector plates 340 without any interim supports. In one configuration, the center section 412 of the motor mount bracket 410 may have an elongated opening 416 for weight reduction. The bracket 410 may have any suitable cross-sectional structural shape which may be uniform or varied between the mounting ends 411. In non-limiting exemplary constructions, the motor mount bracket 410 may be made of aluminum or steel.

Gaseous Fuel Power Generators

FIGS. 104-108 illustrate some non-limiting examples of portable generators for producing electric power which may be fueled by one or more gas canisters or cylinders containing a compressed liquid fuel, which changes to gaseous phase when released from the containers. In one non-limiting example, the fuel source may be propane. The canisters or cylinders may be interchangeably/replaceable in design or permanent refillable type gas containers in various examples. The generators shown may include an engine configured to operate on a gaseous fuel source and include all related appurtenances and accessories necessary for a fully functional generator system (e.g. alternator, electrical system, pressure regulators, relief valves, controls, electrical outlets, etc.).

FIG. 104 depicts a non-limiting example of a portable generator configured in the form of a backpack. In one configuration, without limitation, the backpack generator 500 may be fueled by one or more gas replaceable gas canisters 502. The canisters 502 may be standard 16 ounce propane canisters in one example. The external housing 504 of the backpack generator may be formed at least in part from a soft cover material in one configuration made from NFPA rated fire resistant materials (e.g. fabric, etc.), and may have any suitable shape. In some constructions, the housing 504 may be formed from hard materials such as plastic or a combination of soft and hard materials. The backpack generator 500 may include two padded shoulder straps 506, one or more electrical outlets 508, engine 510 configured and designed to operate on gaseous fuel, alternator 511, pressure regulators 512, recoil manual start mechanism 514, and other appurtenances and accessories for a fully functional generator system.

FIGS. 105-108 depict various non-limiting examples of "suitcase" style compact portable generators 520 fueled by one or more gas canisters or cylinders containing a compressed liquid fuel such as propane. Multiple gas canisters 502 or larger cylinders 522 can be used to extend the run time of the generators. The gas canisters 502 may be standard 16 ounce replaceable propane canisters. In one non-limiting example shown in FIG. 107, a single larger replaceable or permanent refillable gas cylinder 522 (e.g. 5 pound or other) may be used. The gas canisters 502 or cylinder(s) 522 may be partially or completed contained in an outer case 524 which may include a carrying handle 526 in some examples. One or more pressure regulators 512 may be provided for each generator. The generator 520 shown in FIG. 106 may include a gas-fueled folding pivotable cooking range 521 which can be deployed to provide the ability to both cook with gas and produce electric power. The range 521 may be fueled by the gas canister(s) 502 or cylinder(s) 522.

The compact generators 520 each include an engine (not shown) configured to operate on gaseous fuel which may be mounted inside the outer case 524. The engine may be similar to the engine 510 shown in FIG. 104 or different.

In various configurations of the liquid or gaseous fueled generators disclosed herein, it will be appreciated that the power generation units may be configured as either a conventional generator (i.e. drawing AC electric power directly off the wire coil) or an inverter generator (i.e. transforming electric power off the coil into DC power and then back into AC).

While the foregoing description and drawings represent some example systems, it will be understood that various additions, modifications and substitutions may be made therein without departing from the spirit and scope and range of equivalents of the accompanying claims. In particular, it will be clear to those skilled in the art that the present invention may be embodied in other forms, structures, arrangements, proportions, sizes, and with other elements, materials, and components, without departing from the spirit or essential characteristics thereof. In addition, numerous variations in the methods/processes. One skilled in the art will further appreciate that the invention may be used with many modifications of structure, arrangement, proportions, sizes, materials, and components and otherwise, used in the practice of the invention, which are particularly adapted to specific environments and operative requirements without departing from the principles of the present invention. The presently disclosed examples are therefore to be considered in all respects as illustrative and not restrictive, the scope of the invention being defined by the appended claims and equivalents thereof, and not limited to the foregoing description or examples. Rather, the appended claims should be construed broadly, to include other variants and examples of the invention, which may be made by those skilled in the art without departing from the scope and range of equivalents of the invention.

What is claimed is:

1. A modular portable generator frame system, the system comprising:

an open space frame including a top, opposing bottom, a plurality of horizontally extending top lateral elements perimetrically circumscribing the top of the frame, a plurality of horizontally extending bottom lateral elements perimetrically circumscribing the bottom of the frame, and vertically extending corner elements each being coupled between a pair of the top and bottom lateral elements;

an open interior space defined by the frame;

a plurality of corner members each comprising a central body and plurality of elongated radially extending mounting tangs extending outwards from the central body and having a first set of locking features;

each corner member arranged at an intersection of either a pair of the top lateral elements or a pair of the bottom lateral elements;

a plurality of connector plates each comprising a second set of locking features complementary configured with the first set of locking features to form an interlocked keyed fit;

wherein each of the top and bottom lateral elements and vertically extending corner elements are formed by an interlocked assembly of a pair of the plurality of the

connector plates which lockingly receive therebetween one of the mounting tangs from each of two opposing corner members;

wherein each mounting tang includes a mounting hole concentrically aligned with a mating mounting hole in each connector plate of the assembly which receives a removable fastener therethrough which secures the assembly together.

2. The system according to claim **1**, wherein the central body of the corner members have a central opening having a polygonal shape defined by a short segment defined by the mounting tangs and a long segment connecting the mounting tangs together.

3. The system according to claim **2**, further comprising a corner cap including a raised plateau protuberance on an inner surface which has a complementary polygonal shape to the central opening and a mounting hole, the protuberance insertable through the central opening to secure the corner cap to the corner member.

4. The system according to claim **2**, further comprising an annular shaped fuel tank configured for mounting to the frame inside the top lateral elements.

5. The system according to claim **4**, wherein the fuel tank includes a plurality of corners having a threaded recess which receives a mating threaded fastener inserted through the mounting holes in the corner caps to secure the fuel tank to the corner members of the frame.

6. The system according to claim **4**, wherein the fuel tank includes an upwardly open well and a removable portable fuel tank inserted in the well.

7. The system according to claim **1**, wherein the mounting tangs from the two opposing corner members are arranged in proximate end-to-end relationship between the pair of connector plates in the locking assembly.

8. The system according to claim **1**, further comprising an engine mounted in the interior space of the frame.

9. A modular portable generator frame system, the system comprising:

an open space frame including a top, opposing bottom, a plurality of horizontally extending top and bottom lateral elements circumscribing the top and bottom of the frame respectively, and vertically extending corner elements coupled between the top and bottom lateral elements;

an open interior space defined by the frame;

a plurality of corner members each comprising a central body and plurality of elongated radially extending mounting tangs extending outwards from the central body and having a first set of locking features;

a plurality of connector plates each comprising a second set of locking features complementary configured with the first set of locking features to form an interlocked keyed fit;

wherein each of the top and bottom lateral elements and vertically extending corner elements are formed by an interlocked assembly of a pair of the plurality of the connector plates which lockingly receive therebetween one of the mounting tangs from each of two opposing corner members;

wherein the connector plates each include a pair of opposing locking tabs engaged with complementary configured lateral locking recesses formed in each mounting tang of the corner members to form an interlocked fit which cannot be axially separated when assembled.

10. The system according to claim **9**, wherein the locking tabs protrude inwardly towards each other and the lateral locking recesses each face outwardly to receive one of the locking tabs.

11. The system according to claim **9**, wherein the connector plates each further include a locking protrusion engaged with a complementary configured end locking recess formed on a distal free end of each mounting tang.

12. The system according to claim **11**, wherein the end locking recess opens radially outwards, the end locking recess and locking protrusion each having complementary mating non-rectilinear polygonal shapes to form an interlocked fit which cannot be axially separated when assembled.

13. The system according to claim **9**, wherein each mounting tang includes a mounting hole concentrically aligned with a mating mounting hole in each connector plate of the assembly which receives a removable fastener therethrough which secures the assembly together.

14. The system according to claim **9**, wherein the central body of the corner members have a central opening having a polygonal shape defined by a short segment defined by the mounting tangs and a long segment connecting the mounting tangs together.

15. The system according to claim **9**, wherein the corner members have a triad of radially extending mounting tangs spaced equally angularly apart.

16. A modular portable generator frame system, the system comprising:

an open space frame including a top, opposing bottom, a plurality of horizontally extending top and bottom lateral elements circumscribing the top and bottom of the frame respectively, and vertically extending corner elements coupled between the top and bottom lateral elements;

an open interior space defined by the frame;

a plurality of corner members each comprising a central body and plurality of elongated radially extending mounting tangs extending outwards from the central body and having a first set of locking features;

a plurality of connector plates each comprising a second set of locking features complementary configured with the first set of locking features to form an interlocked keyed fit;

wherein each of the top and bottom lateral elements and vertically extending corner elements are formed by an interlocked assembly of a pair of the plurality of the connector plates which lockingly receive therebetween one of the mounting tangs from each of two opposing corner members;

wherein the corner members have a triad of radially extending mounting tangs spaced equally angularly apart.

17. The system according to claim **16**, wherein the mounting tangs are each disposed at an oblique angle to the central body of the corner members.

18. The system according to claim **16**, wherein each mounting tang includes a mounting hole concentrically aligned with a mating mounting hole in each connector plate of the assembly which receives a removable fastener therethrough which secures the assembly together.

19. The system according to claim **16**, wherein the central body of the corner members have a central opening having a polygonal shape defined by a short segment defined by the mounting tangs and a long segment connecting the mounting tangs together.

* * * * *

EXHIBIT 1008

Oct. 24, 1933. G. L. HOLZAPFEL 1,931,698

FUEL SYSTEM FOR INTERNAL COMBUSTION ENGINES

Filed March 30, 1931

Fig. 1

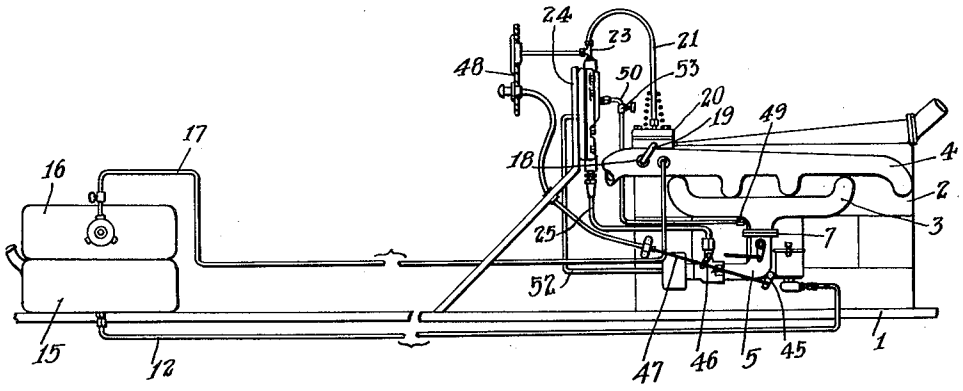


Fig. 2

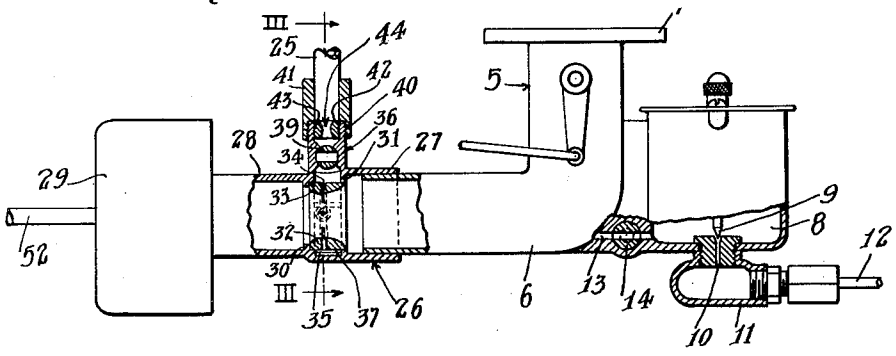
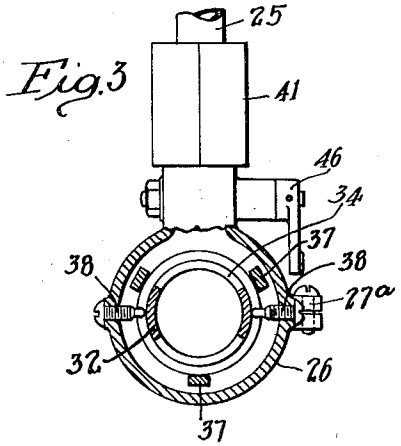


Fig. 3



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UNITED STATES PATENT OFFICE

1,931,698

FUEL SYSTEM FOR INTERNAL COMBUSTION ENGINES

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Application March 30, 1931. Serial No. 526,175

4 Claims. (Cl. 261—16)

My invention relates to fuel systems for internal combustion engines, and has particular reference to a fuel system embodying the supplying of liquid fuel and gaseous fuel to an engine in alternation at will.

In the copending joint application of David E. Day and myself, Serial No. 494,328, filed November 8, 1930, a system of employing gaseous fuel for an internal combustion engine was disclosed in which a relatively cheap low grade gaseous fuel stored under pressure was passed through a pressure reducing valve to be introduced into a carbureter connected with an internal combustion engine so that gaseous fuel could be supplied to the engine at the desired pressure and mixed with air in the carbureter in the desired quantities to produce a proper combustible mixture for forming a fuel for the engine.

However, it frequently occurs that it is undesirable to rely upon the gaseous fuel as the sole source of supply for an internal combustion engine, particularly when such engine is employed for the purpose of driving a motor vehicle. It is, therefore, an object of this invention to provide a fuel system in which a gaseous fuel supply shall be auxiliary to the liquid fuel supply now commonly employed in connection with automobiles and similar vehicles.

Another object of the invention is to provide a fuel system in which the normal liquid fuel supply and carbureter therefor is combined with a gaseous fuel supply and a mixer therefor, the two fuel supplies being used in alternation to each other.

Another object of the invention is to provide a fuel system as set forth in the preceding paragraph in which the liquid fuel supply may be cut off and the gaseous fuel supply turned on, or vice versa so that when the liquid fuel is employed the gaseous fuel is disconnected and while the gaseous fuel is employed the liquid fuel is disconnected.

Another object of the invention is to provide a fuel system for supplying the gaseous fuel to an internal combustion engine in alternation with the normal liquid fuel supply and in which the gaseous fuel system may be readily connected to the existing liquid fuel carbureters now used on automobiles and similar vehicles.

Other objects and advantages will be apparent from a study of the following specification, read in connection with the accompanying drawing, wherein

Figure 1 is a diagrammatic view indicating the various elements of both the liquid and gaseous fuel supplies in proper relation to an automobile engine;

Figure 2 is a detailed view of a carbureter such as generally employed for supplying liquid fuel to an engine, this view being partly in section to illustrate the attachment of my gaseous fuel mixing device thereto; and

Figure 3 is a detailed, sectional view taken along line III—III of Fig. 2, illustrating the interior mechanism of the gaseous fuel mixer employed in practicing my invention.

Referring to the drawing, I have illustrated a portion of an automobile chassis 1, constituting the mounting for an internal combustion engine 2, such as is commonly employed in automobiles and the like. The engine 2 is illustrated as being provided with an intake manifold 3 communicating with the intake ports of the various cylinders comprising the engine 2. An exhaust manifold 4 is illustrated as being connected to the inside ports of the cylinders constituting the engine.

Connected to the intake manifold 3 is a carbureter 5 of any suitable construction, such as those constructions now employed for the mixing of liquid fuel with air prior to injection of such mixture into the intake manifold of the engine.

The carbureter 5 is illustrated as comprising a bent tube 6, one end 7 of which is connected to the intake manifold 3 and the other end of which constitutes an inlet opening through which air may be sucked for mixing with the fuel injected into the carbureter. The carbureter is equipped with a float chamber or fuel reservoir 8 which may be of any suitable construction to provide a supply of fuel at atmospheric pressure ready for injection into the tube 6 of the carbureter 5, the float chamber being provided with a float operated needle valve 9 which obstructs the entrance of fuel into the chamber 8 by closing off the passage 10 communicating with an inlet connection 11 to which the fuel supply line 12 may be connected.

The float chamber 8 is illustrated as communicating with the bent tube portion of the carbureter as by means of a passage 13 communicating with both the float chamber 8 and the tube 6. A rotary valve 14 is preferably interposed in the passage 13 for the purpose of permitting this passage to be cut off, as will hereinafter be described.

A main liquid supply tank is illustrated at 15 having attached to the lower end thereof the fuel supply line 12 through which fuel from the tank

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15 may be passed to the float chamber 8 of the carburetor.

The foregoing description of the liquid fuel system is in all essential respects identical with the fuel supply systems now in common use in connection with automobiles, and similar vehicles.

I propose to add an additional fuel supply system employing a normally gaseous fuel, this system comprising a fuel supply tank 16 in which normally gaseous fuel is compressed and thereby maintained in liquefied condition. From tank 16 there extends a suitable gaseous fuel supply tube 17 extending to the bottom of the tank 16, so as to draw off the liquid and not gas. The gaseous fuel supply tube 17 is preferably threaded through the exhaust manifold 4 of the engine 2, as is indicated at 18, so that the fuel passing therethrough will be heated by contact of the tube 17 with the hot exhaust gases coming from the engine. The tube 17 continues from the exhaust manifold 4 as by way of the short section of tubing 19 into a combined filter and heat exchanger 20 which further assists in heating the gaseous fuel. From the filter 20 the fuel inlet tube continues, as is indicated at 21, to the inlet connection 23 to a pressure reducing valve 24.

The pressure reducing valve 24 is illustrated and described in detail in my copending application Serial No. 491,468, filed October 27, 1930 which has since matured into Patent 1,912,479 dated June 6, 1933 and comprises a valve constructed to receive a gas under relatively high pressure and to permit gases to flow therefrom at a pressure equal to that of the air supply. In order to equalize these pressures, a pipe 52 connects the air cleaner 29 with the atmosphere side of the pressure equalizing valve 24.

The gaseous fuel supply through outlet tube 25 may be mixed with air and passed into the tube 6 of the carbureter 5 so that such air and gas mixture may pass into the intake manifold 3 of the engine 2 and thus constitute the combustible fuel for operating the engine 2.

The mixing device is illustrated as comprising an attachment which may be connected to the inlet end of the carbureter tube 6, this mixing device comprising a tubular member 26, one end 27 of which is provided with an internal diameter of such size as to telescopically engage the air inlet end of the carbureter tube 6 and may be clamped thereon as by means of clamping screw 27*a*. The opposite end 28 of the mixer 26 is illustrated as being of slightly reduced diameter so that its outside diameter corresponds substantially with the outside diameter of the inlet end of the tubular carbureter 6. By reason of this reduced diameter the air cleaner 29, which normally is telescopically engaged upon the inlet end of the carbureter 6, may be inserted in place upon the end 28 of my mixer.

Within the tubular mixer 26 is a pair of radial flanges 30 and 31 which extend inwardly to form a seat for a venturi 32. The flange 30 is preferably formed with a shoulder 33 thereon to act as a limit stop for the venturi 32 when the same is inserted into engagement with the flanges 30 and 31.

The venturi 32 comprises an annular ring which has one or more slots 34 extending radially therethrough. Between the flanges 30 and 31 is an annular space 35 surrounding the venturi 32. This annular space 35 communicates with a gas inlet stem 36 through which gaseous fuel may be passed into the annular space 35 and thence through the slots 34 to the interior of the venturi 32. The annular space 35 between the flanges 30 and 31 may be bridged at a plurality of points by means of cross members 37 to assist in guiding the venturi 32 into its proper seating relation relative to the flanges when the same is to be inserted within the tubular mixer 26. The venturi 32 may be held in place when inserted as by means of one or more set screws 38 threaded radially into the mixer 26.

The gas inlet stem 36 is preferably provided with a cut off valve 39 which may be operated to either open the inlet passage 36 to the passage of gas therethrough, or to close the same so that gas cannot pass therethrough.

The upper end of the stem 36 is provided with external threads 40 for engagement with a suitable union nut 41 to secure the gas inlet pipe 25 thereto while the stem 36 may be additionally provided with internal threads 42 to permit a metering jet 43 to be threaded thereinto to provide the desired size orifice 44 for metering or limiting the quantity of gas permitted to flow into the mixer 26.

Referring, particularly, to Figure 1, it will be observed that the operating levers 45 and 46 for the valves 14 and 39, respectively, are connected to a rod or wire 47 which extends to a point of convenient access to the operator of the vehicle such as, for example, the dashboard 48 so that the rod or wire 47 may be manipulated to simultaneously control both valves 14 and 39.

The operation of my device is as follows: When the engine 2 is to be started the valves 14 and 39 may be operated by the rod or wire 47 to the position opposite to that illustrated in Fig. 2. That is, the valve 14 is closed and the valve 39 is open so that under these conditions gaseous fuel will be supplied to the carburetor 5 and thus the engine may be started upon the gaseous fuel. For small loads, or low speeds, the engine may be operated continuously upon such gaseous fuel though as soon as the engine is started it may be switched over to the liquid fuel by manipulating the rod or wire 47. Operating the rod or wire 47 will reverse the valves 14 and 39 so that the gaseous fuel supply is cut off and the liquid fuel supply connected to feed the carburetor.

Either the liquid fuel contained in tank 16 which changes to gas on release of pressure or the higher boiling point gasoline fuel in tank 15 may be used for starting the motor, the object of applicant's invention being to obtain flexibility in the use of either type of fuel as conditions or supply may warrant. For example, on a cold morning, or with a cold motor, it would be preferable to use the low boiling point fuel in tank 16 to start the motor and thereafter when the motor is warmed up the fuel supply coming from tank 16 may, if desired, be shut off and the gasoline supply in tank 15 turned on, although the operation of the motor can be maintained on the fuel from tank 16 if desired. There is an advantage in starting a cold motor on the low boiling point fuel from tank 16 in that liquid fuel is not deposited on the cylinder walls to wash away the lubricating oil. Furthermore, the low boiling point fuel from tank 16 has higher anti-knock value than ordinary gasoline so that where a long run on full throttle is to be made, the motor may be supplied with fuel from the tank 16 and detonation or knocking prevented.

Upon opening the valve 39 gaseous fuel passing through the metering jet 43 will pass to the annular chamber 35 surrounding the venturi 32 and

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will thence pass inwardly through the leads 34 to be mixed with the incoming air stream drawn into the inlet end of the carbureter 5 through the air cleaner 29.

By properly selecting the orifice 44 of the metering jet 43 and the slotted orifices 34 of the venturi 32, the desired quantity of gaseous fuel to be supplied relative to the quantity of air drawn into the carburetor may be adjusted to provide a properly combustible mixture.

It will be observed that as the air is drawn through the venturi into the carbureter 5, the quantity of air so drawn will act on the venturi 32 to increase or decrease the quantity of gaseous fuel supplied in proportion to the quantity of air drawn through so that the proper combustible mixture will be maintained at various speeds of operation of the engine.

For idling purposes, when operating on the gaseous fuel, an idler connection 49 may be connected into the intake manifold 3 of the engine 2, this connection being made as by means of a tube or pipe 50 to the low pressure side of the pressure reducing valve 24. Thus at idling speeds a desirable quantity of fuel for idling may be furnished to the engine. The amount of fuel bypassed to the engine is regulated by the valve 53.

It will be observed that I have disclosed herein a system in which liquid fuel or gaseous fuel may be used in alternation with each other for supplying the required fuel to the engine. Hence the engine 2 may be supplied with high grade gaseous fuel for the greater portion of its operation and the liquid fuel may be used only when a supply of gaseous fuel is undesirable.

If desired, the valves 14 and 39 may be operated so as to cut off partly both the liquid and the gaseous fuel, so that a progressive proportioning can be utilized whereby more or less liquid or gaseous fuel can be utilized without disturbing the air fuel ratio of the mixture delivered to the engine.

While I have illustrated and described the preferred embodiment of my invention, I do not desire to be limited to any of the details shown or described herein, except as defined in the appended claims.

I claim:

1. In a fuel system for internal combustion engines, a liquid fuel supply, a gaseous fuel supply, a carbureter comprising an air passage through which air may be passed into the intake of the engine, a liquid fuel inlet communicating with the air passage of said carbureter for supplying liquid fuel to the air stream drawn through the carbureter, a cut off valve for cutting off the supply of liquid fuel to said carbureter, a gaseous fuel inlet connected to and communicating with the air passage of said carbureter, said gaseous fuel inlet including a venturi through which said air stream may pass and having openings therein through which gaseous fuel may be drawn through the venturi into the air stream, a valve interposed between said gaseous fuel inlet and said venturi, and means for simultaneously operating said valves to open one of said valves and close the other, and vice versa, to supply either liquid or gaseous fuel to said carbureter.

2. In a fuel system for internal combustion engines having a carbureter, a source of gaseous fuel, means for introducing said gaseous fuel into said carbureter comprising a tube connectible to the air inlet passage of said carbureter, a venturi in said tube having openings therein through which gaseous fuel may be passed into said tube, a gaseous fuel inlet communicating with the openings in said venturi, and a metering jet in said gaseous fuel inlet for metering the quantity of fuel supplied therethrough.

3. An attachment for a standard liquid fuel carbureter having an air supply opening, which attachment comprises a tube connectible to the air inlet passage of said carbureter, said tube having a Venturi passage therein for air drawn through said tube into the carbureter, with openings in the throat of the venturi through which gaseous fuel may flow into said tube and a gaseous fuel inlet passage communicating with the openings in the throat of the venturi.

4. An attachment for a liquid fuel carbureter having an air inlet nipple with an air cleaner slidably engageable therewith, which attachment comprises a tube dimensioned at one end to engage with the air inlet of the carbureter normally engaging with the air cleaner, and dimensioned at the opposite end to engage with the end of the air cleaner normally engaging with the carbureter inlet, whereby said tube may be inserted as a unit between the air inlet passage of the carbureter and the air cleaner, said tube being provided with gaseous fuel inlet openings and an inlet port connected thereto for connection to a supply of gaseous fuel.

GEORGE L. HOLZAPFEL.

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EXHIBIT 1009

*Meaning of **apparatus** in English*

apparatus

noun

UK /ˌæp.əˈreɪ.təs/ **US** /ˌæp.əˈræţ.əs/

plural **apparatuses**

apparatus *noun* (EQUIPMENT)

Add to word list

[U or C]

a set of equipment or tools or a machine that is used for a particular purpose:

- *a **piece of** apparatus*
- *The divers checked their breathing apparatus.*

− SMART Vocabulary: related words and phrases

Things needed for a task

accoutrements

armoury

could use something *idiom*

equipment

gadgetry

gear

kit

material

matériel

necessaries

tackle

thing

tool

tool (something/someone) up *phrasal verb*

use

apparatus ***noun*** **(ORGANIZATION)**

[C usually singular]

an organization or system, especially a political one:

- *The whole apparatus of communism was already falling apart.*

+ SMART Vocabulary: related words and phrases

*(Definition of **apparatus** from the **Cambridge Advanced Learner's Dictionary & Thesaurus** © Cambridge University Press)*

apparatus | AMERICAN DICTIONARY

apparatus

noun

US /ˌæp·əˈræţ·əs/

plural **apparatuses** or **apparatus**

apparatus ***noun*** **(EQUIPMENT)**

Add to word list

[C/U]

a set of equipment, tools, or a machine that is used for a particular purpose:

- [C] *The garage had an apparatus to lift cars up.*

apparatus ***noun*** **(SYSTEM)**

[C]

a system or organization:

(Definition of **apparatus** from the **Cambridge Academic Content Dictionary** © Cambridge University Press)

EXAMPLES of **apparatus**

apparatus

The source of statehood remained extraverted, limiting the impact of the colonial state and its apparatuses.

From the Cambridge English Corpus

Indeed, the healthcare system seems to be primarily devoted to increasing the use of a variety of artefacts: synthetic drugs, diagnostics, a whole range of biomedical apparatuses.

From the Cambridge English Corpus

288 development of the outer and middle ear apparatuses.

From the Cambridge English Corpus

The book's strength lies in its detailed discussion of military and security apparatuses and capabilities, which will be of significant interest to defense and policy analysts.

From the Cambridge English Corpus

Cruz does not directly address this concern, for the book says little about differences among the countries that concern their socioeconomic conditions and the organisation of their state apparatuses.

From the Cambridge English Corpus

The 1930s proved to be a decade of relative prosperity for most chicleros, partly because state apparatuses initially compensated for the fall in price on the world market.

From the Cambridge English Corpus

One such option taken by the new leaders combined constitutional amendments, the multiplication of restrictive political statutes (for example, censorship laws), and the creation of elaborate security apparatuses.

From the Cambridge English Corpus

Of course, these apparatuses are being provided in the maternity hospitals as quickly as they can be made available and midwives trained to use them.

From the Hansard archive

From the Hansard archive

He was responsible for dealing with these diathermic apparatuses because they were, in effect, transmitting stations.

From the Hansard archive

The apparatuses—so long as they are used within those prescribed limits, the police themselves are protected.

From the Hansard archive

The limits of these two apparatuses in many respects appear to be the same.

From the Hansard archive

Are you going to forbid the pubs and clubs and hotels in a certain area to turn on their apparatuses on a certain occasion or between certain hours?

From the Hansard archive

The *apparatus* is shown in figure 7(a).

From the Cambridge English Corpus

Two kinds of *apparatus* were used in this study.

From the Cambridge English Corpus

See all examples of apparatus

These examples are from corpora and from sources on the web. Any opinions in the examples do not represent the opinion of the Cambridge Dictionary editors or of Cambridge University Press or its licensors.

COLLOCATIONS with **apparatus**

apparatus

These are words often used in combination with *apparatus*.

Click on a collocation to see more examples of it.

administrative apparatus

- First, union participation in the administrative apparatus of the welfare state provides them with a potentially important voice in relevant decision-making.

From the Cambridge English Corpus

and considered as lying elsewhere.

From the Cambridge English Corpus

complex apparatus

- In oligodendrocytes, recent studies provided evidence of a complex apparatus of molecules, which might coordinate trafficking of proteins and lipids.

From the Cambridge English Corpus

These examples are from corpora and from sources on the web. Any opinions in the examples do not represent the opinion of the Cambridge Dictionary editors or of Cambridge University Press or its licensors.

See all collocations with apparatus

What is the pronunciation of *apparatus*?

Translations of **apparatus**

in Chinese (Traditional)

設備, 儀器, 裝置…

See more

in Chinese (Simplified)

设备, 仪器, 装置…

See more

in Spanish

equipo, aparato, aparato [masculine]…

See more

in Portuguese

aparelhagem, aparato, instrumento [masculine]…

See more

in more languages

Need a translator?

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Browse

appalled

appalling

appallingly

apparatchik

apparatus

apparatuses

apparel

apparent

apparent damage

*Meaning of **couple** in English*

couple

noun

UK /ˈkʌp.ᵊl/ **US** /ˈkʌp.ᵊl/

couple *noun* (SOME)

Add to word list

B1 [S]

two or a few things that are similar or the same, or two or a few people who are in some way connected:

- **a couple of** *The doctor said my leg should be better in a couple of days.*
- *A couple of people objected to the proposal, but the vast majority approved of it.*
- **another couple of** *We'll have to wait another couple of hours for the paint to dry.*
- **a couple more** *She'll be retiring in a couple more years.*
- **the last couple of** *The weather's been terrible for the last couple of days.*
- **the next couple of** *Many economists expect unemployment to fall over the next couple of months.*
- **the past couple of** *I'm sorry I didn't call you, but I've been very busy over the past couple of weeks.*

− Fewer examples

- *Take a couple of steps backwards.*
- *She lived in Rome for a couple of years, where she taught English.*
- *Take a couple of weeks off - you need a break.*

- *There's not much in the way of entertainment in this town - just the cinema and a couple of pubs.*

+ SMART Vocabulary: related words and phrases

couple *noun* (TWO PEOPLE)

B1 [C, + sing/pl verb]

two people who are married or in a romantic or sexual relationship, or two people who are together for a particular purpose:

- **married couple** *We sit in our armchairs watching TV like an old married couple.*
- **elderly couple** *An elderly couple live next door.*
- **young couple** *Should the government do more to help young couples buy their own homes?*
- *The couple skated spectacularly throughout the competition.*

+ More examples

+ SMART Vocabulary: related words and phrases

couple

verb

UK /ˈkʌp.ᵊl/ US /ˈkʌp.ᵊl/

couple *verb* (JOIN)

[T usually passive, usually + adv/prep]

to join or combine:

- **be coupled together** *The sleeping car and restaurant car were coupled together.*

- **couple something with something** *The government has promoted the use of public transport, but has not coupled it with adequate investment in the network.*

+ SMART Vocabulary: related words and phrases

couple *verb* **(HAVE SEX)**

[I] formal

When two people or two animals couple, they have sex.

(Definition of ***couple*** *from the* ***Cambridge Advanced Learner's Dictionary & Thesaurus*** *© Cambridge University Press)*

couple | AMERICAN DICTIONARY

couple

noun

US /ˈkʌp·əl/

couple *noun* **(SOME)**

Add to word list

[U]

two or a few things that are similar or the same, or two or a few people who are in some way connected:

- *I'm packing a couple of sweaters in case it gets cold.*

couple noun (TWO PEOPLE)

[C]

two people who are married or who spend a lot of time together esp. in a romantic relationship:

- *We're having two couples over for dinner.*

*(Definition of **couple** from the **Cambridge Academic Content Dictionary** © Cambridge University Press)*

What is the pronunciation of *couple*?

Translations of **couple**

in Chinese (Traditional)

一些，（相似或相同事物的）一對，一雙，幾個，（有某種聯繫之人的）一對，幾個…

See more

in Chinese (Simplified)

一些，（相似或相同事物的）一对，一双，几个，（有某种联系之人的）一对，几个…

See more

in Spanish

par, pareja, pareja [feminine]…

See more

in Portuguese

alguns, casal, casal [masculine]…

See more

in more languages

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Translator tool

coup de grace

coup de grâce to something

coup de théâtre

coupe

couple

couple something together

couple something with something

coupled

couplet

*Meaning of **operate** in English*

operate

verb

UK /ˈɒp.ᵊr.eɪt/ **US** /ˈɑː.pə.reɪt/

operate *verb* (WORK)

Add to word list

B2 [I or T]

to (cause to) work, be in action or have an effect:

- *How do you operate the remote control unit?*
- *Does the company operate a pension scheme?*
- *For several years she operated a dating agency from her basement.*
- *Changes are being introduced to make the department operate more efficiently.*
- *Specially equipped troops are operating in the hills.*
- *We have representatives operating in most countries.*
- *Exchange rates are currently operating to the advantage of exporters.*

− Fewer examples

- *The video provides instruction on how to operate the computer.*
- *This sewing machine is operated by a foot pedal.*
- *The legal system operates very differently in the US and Britain.*
- *Terrorists have been operating covertly in England for several years.*

+ SMART Vocabulary: related words and phrases

operate *verb* **(MEDICAL PROCESS)**

B1 [I]

to cut a body open for medical reasons in order to repair, remove, or replace an unhealthy or damaged part:

- *If the growth gets any bigger they'll have to operate.*
- *Are they going to operate **on** him?*

+ SMART Vocabulary: related words and phrases

*(Definition of **operate** from the **Cambridge Advanced Learner's Dictionary & Thesaurus** © Cambridge University Press)*

operate | AMERICAN DICTIONARY

operate

verb

US /ˈɑp·əˌreɪt/

operate *verb* **(WORK)**

Add to word list

[I/T]

to work or cause something to work, be in action, or have an effect:

- [T] *How do you operate the remote control unit?*
- [I] *Changes are being introduced to make the department operate more efficiently.*

operate *verb* **(USE MEDICAL PROCESS)**

[I]

to cut a body open in order to repair, remove, or replace an unhealthy or damaged part:

operate

verb

UK /ˈɒpəreɪt/ **US**

Add to word list

[T]

to use or control equipment or a machine or to make something work:

- *Older people are returning to classrooms for instruction in everything from operating a mouse to mastering spreadsheets.*

[I]

if a business, system, etc. operates in a particular way, it works or is used in that way:

- *Changes are being introduced to make the department operate more profitably.*
- *Exchange rates are currently operating to the advantage of exporters.*
- **operate as sth** *The company operates as an international delivery service for produce trading companies.*
- **operate through sb/sth** *Centralized lenders operate through mortgage brokers and insurance companies.*
- **operate at sth** *The plant is operating at barely 30% of capacity.*

[I]

to do business in or from a particular place:

- **operate across/in/from sth** *We operate in 50 countries around the world.*
- *We have representatives operating in most countries.*
- **operate out of sth** *The organization operates out of a small office in the centre of town.*

[T]

to manage a business, service, etc. and make it work:

- *The company operated 156 fewer stores this year.*
- *We began operating a pilot service earlier this year.*
- *Does the company operate a pension plan?*

*(Definition of **operate** from the **Cambridge Business English Dictionary** © Cambridge University Press)*

EXAMPLES of **operate**

operate

In the procedure of calculation above, it can be easily seen that the next step is being operated with the remainder of the former step.

From the Cambridge English Corpus

They have a vested interest in teaching it just as surgeons have a vested interest in operating on people.

From the Cambridge English Corpus

I observed and took notes of each interaction from a distance and operated the tape-recorder myself.

From the Cambridge English Corpus

But if both processes are operating, what explanation provides a synthesis?

These systems *operate* actively over many repeated hearings.

From the Cambridge English Corpus

In section 3.1 we discuss how to represent tables and how to *operate* on tables.

From the Cambridge English Corpus

Developmentalists show that these schemas *operate* between language and perception, thereby facilitating semantic development.

From the Cambridge English Corpus

Further, structural conditions may affect how the processes identified here *operate*.

From the Cambridge English Corpus

They favored the creation of strong central bureaucratic authorities to ensure that the new agglomerations of private power operated ultimately in the public interest.

From the Cambridge English Corpus

It is also the case that historic preservation virtually never operates this way either.

From the Cambridge English Corpus

This suggests that any scaffolding provided by previous utterances operates at a highly lexically-specific level at this stage in development.

From the Cambridge English Corpus

A heavy-duty mobile manipulator while carrying a load, experiencing a force, or operating on uneven terrain needs to maintain its balance.

From the Cambridge English Corpus

This was in part because it wanted to keep on good terms with the government that was operating in its interests.

From the Cambridge English Corpus

In fact, a change of pressure is the practical way to optimize a given machine (fixed geometry) to *operate* in a certain regime.

From the Cambridge English Corpus

Modulation of the oil flow in the main valves is presently controlled by the pilot oil pressure through manually operated pilot control valves.

From the Cambridge English Corpus

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What is the pronunciation of *operate*?

Translations of **operate**

in Chinese (Traditional)

工作, (使)工作, (使)運行, 運作...

See more

in Chinese (Simplified)

工作, (使)工作, (使)运行, 运作...

See more

in Spanish

operar, funcionar, manejar...

See more

in Portuguese

operar, acionar, funcionar...

See more

in more languages

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Translator tool

Browse

operagoer

operand

operant

operant conditioning BETA

operatic

operatically

operating

*Meaning of **together** in English*

together

adverb

UK /təˈgeð.əʳ/ **US** /təˈgeð.ɚ/

together *adverb* (WITH EACH OTHER)

Add to word list

A1

with each other:

- *We used to go to yoga together.*
- *We worked together on a project a couple of years back.*
- *Could you add these numbers together for me?*
- *You mix all the dry ingredients together before you add the milk.*
- *I like both flavours separately but I don't like them together.*
- *You could stick that back together (= join the separate parts to each other) with some glue.*
- **close together** *She said, "Never trust a man whose eyes are so close together!"*
- **all together** *The waiter asked if we were all together, so I explained that we were two separate parties.*
- **get together** *We should get together (= meet each other socially) some time and have a drink.*

See also

altogether *adverb*

- *Mira and Ellis have been together now for almost five years.*

get together informal (UK also **get it together**)

If two people get together or get it together, they start a sexual relationship with each other:

- *We'd seen each other a few times as part of a group, but we didn't really get it together till Rachel's party.*

See more

− Fewer examples

- *I cherish the memories of the time we spent together.*
- *When you've finished your work sheets, clip them together and hand them in to me.*
- *It was when we started living together that we found we just weren't compatible.*
- *If you two don't talk out the differences between you, it'll be very difficult for you to continue working together.*
- *The bride and groom walked down the aisle together.*

+ SMART Vocabulary: related words and phrases

together *adverb* (AT THE SAME TIME)

B1

at the same time:

- *The packages were sent separately, but they all came together.*
- *We can deal with the next two items on the list together.*

+ More examples

+ SMART Vocabulary: related words and phrases

B2

combined:

- *Together they must earn over $300,000 a year.*
- **put together** *She's got more sense than the rest of you put together.*

+ More examples

+ SMART Vocabulary: related words and phrases

together *adverb* (IN ONE PLACE)

B2

in one place:

- *I'll just gather my stuff together, and then we can go.*

+ More examples

+ SMART Vocabulary: related words and phrases

together *adverb* (AND ALSO)

B2 mainly UK

in addition to; and also:

- *The money that I owe you for the phone together with the rent equals £300.*

See more

Grammar

Altogether or all together?

Altogether means 'completely' or 'in total': …

Altogether

Altogether means 'completely' or 'in total': …

All together

All together means 'with each other', and is similar to together: …

together

adjective • informal approving

UK /təˈgeð.əʳ/ US /təˈgeð.ɚ/

organized, confident of your abilities, and able to use them to achieve what you want:

- *For a sixteen-year-old, he seems pretty together.*

get it together

to get something organized:

- *We were going to go skiing over Christmas but we never got it together.*

See more

+ SMART Vocabulary: related words and phrases

*(Definition of **together** from the **Cambridge Advanced Learner's Dictionary & Thesaurus** © Cambridge University Press)*

What is the pronunciation of *together*?

Translations of **together**

in Chinese (Traditional)

一起, 共同, 在一起，關係親密…

See more

in Chinese (Simplified)

一起, 共同, 在一起，关系亲密…

See more

in Spanish

juntos, a la vez, juntos/as…

See more

in Portuguese

juntos, ao mesmo tempo, juntos/-tas…

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Translator tool

Browse

toffee-nosed

tofu

tog

toga

together

together with *phrase*

togetherness

toggle

toggled

EXHIBIT 1010

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(54) **FUEL DELIVERY SYSTEM**

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(73) Assignee: **Marine Green, LLC**

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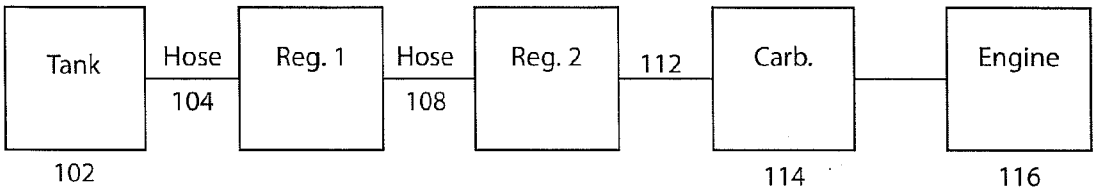
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(57) **ABSTRACT**

A method and system for fuel delivery. The method and system may include one or more fuel storage tanks housing propane, one or more regulators through which fuel may flow and one or more fuel metering devices or systems. The flow of fuel from a fuel storage tank to an engine may be such that the engine runs in a clean and efficient manner.



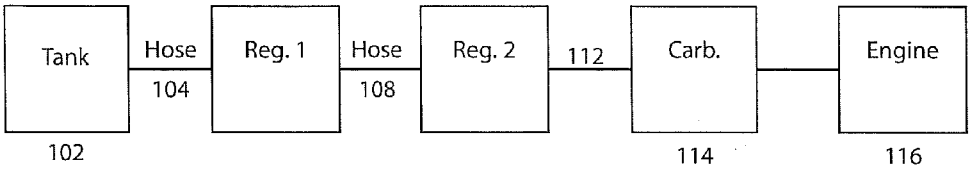


Fig. 1

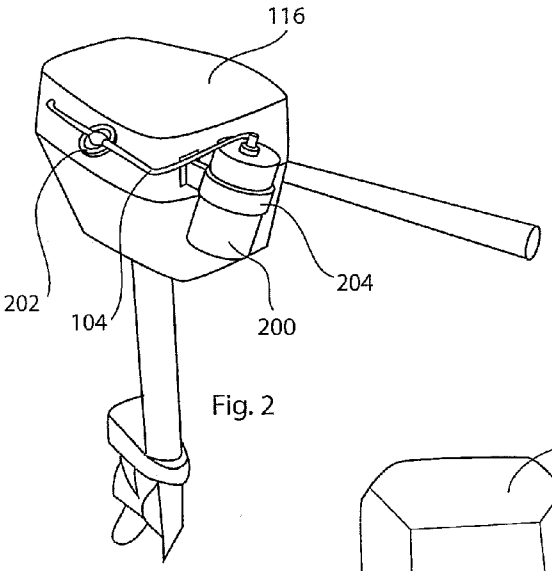


Fig. 2

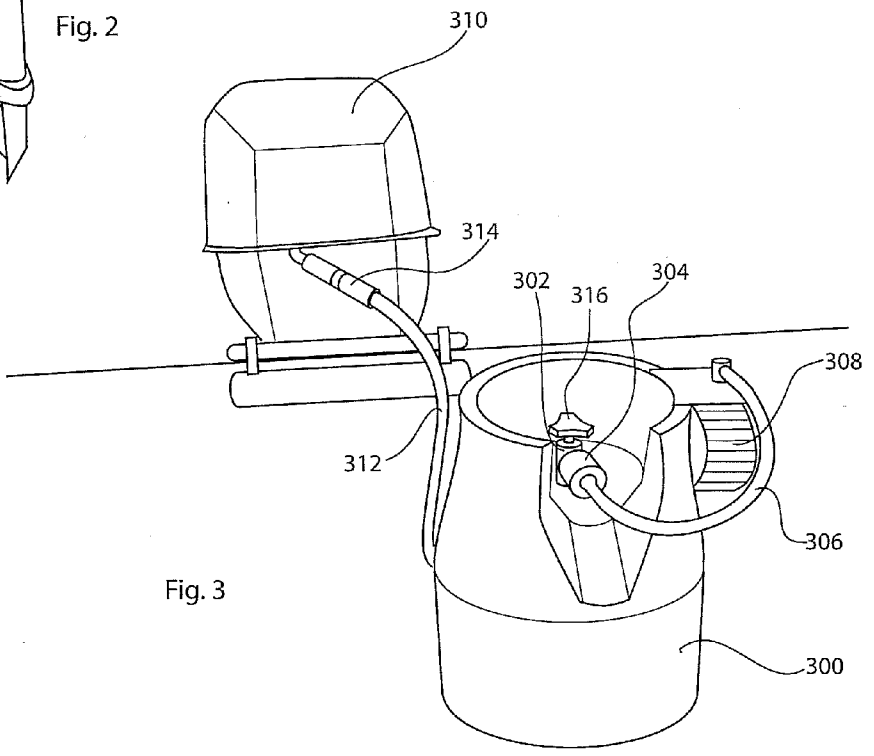


Fig. 3

FUEL DELIVERY SYSTEM

BACKGROUND

[0001] Engines used in any application are known to utilize fuel. In many applications, such as in two and four stroke marine outboard engines, fossil fuels been used as the primary source of fuel. However modern engines are typically criticized for their use of fossil fuels as they are claimed to not be environmentally friendly. Two stroke engines are further criticized due to their running and use characteristics. Thus, in many applications where two stroke engines used to be prevalent, such as in marine outboard engines, designers and manufacturers are now switching to four stroke outboard engines.

[0002] However, as is well known and publicized, four stroke engines remain somewhat inefficient and suffer from a variety of potential drawbacks, specifically with regard to the environment. A number of solutions have been proposed to increase both the efficiency and environmental friendliness of such engines. For example, some marine outboard engines now utilize alternative fuels, such as ethanol-infused gasoline, or bio-gas. These alternatives cause a variety of problems in many applications, however. The use of ethanol in gasoline can absorb water which can cause onboard fuel supplies to eventually contain water as a result of condensation or other means, causing significant problems in engines that are used less frequently. Additionally, for users who wish to safeguard their outboard engines from these problems, fuel disposal is often viewed as a solution but environmentally-unfriendly manners of disposing of unused fuel are often implemented.

[0003] Additionally the decreased effective shelf life of alternative fuels, such as bio-gas can pose problems in engines that are used less frequently as the fuel may not properly combust if it is not used within a limited time period.

[0004] Other attempted solutions, for example the use of liquid propane delivery systems, have not proved successful. In such uses of propane, the vaporization of propane from its liquid state causes a significant amount of heat to be absorbed, ultimately freezing the regulator through which the propane flows from a tank to the engine and rendering the engine inoperable.

SUMMARY

[0005] One exemplary embodiment describes a fuel delivery system. The fuel delivery system may include a fuel tank that houses propane; a first regulator coupled to the fuel tank; a second regulator coupled to the first regulator; and a fuel metering system coupled to the second regulator and delivering propane to an engine.

[0006] Another exemplary embodiment can include a method for providing fuel to an engine. The method for providing fuel to an engine can include storing liquid propane in a tank; flowing the liquid propane to a first regulator; converting the liquid propane to a propane vapor by the first regulator; flowing the propane vapor to a second regulator; metering the flow of the propane vapor from the second regulator to an engine; and maintaining the temperature of the propane at about an ambient temperature.

BRIEF DESCRIPTION OF THE FIGURES

[0007] Advantages of embodiments of the present invention will be apparent from the following detailed description of the exemplary embodiments thereof, which description should be considered in conjunction with the accompanying drawings in which:

[0008] FIG. **1** shows an exemplary block diagram of a fuel delivery system.

[0009] FIG. **2** shows an exemplary diagram of components of a fuel delivery system and engine.

[0010] FIG. **3** shows another exemplary diagram of components of a fuel delivery system and engine.

DETAILED DESCRIPTION

[0011] Aspects of the invention are disclosed in the following description and related drawings directed to specific embodiments of the invention. Alternate embodiments may be devised without departing from the spirit or the scope of the invention. Additionally, well-known elements of exemplary embodiments of the invention will not be described in detail or will be omitted so as not to obscure the relevant details of the invention. Further, to facilitate an understanding of the description discussion of several terms used herein follows.

[0012] The word "exemplary" is used herein to mean "serving as an example, instance, or illustration." Any embodiment described herein as "exemplary" is not necessarily to be construed as preferred or advantageous over other embodiments. Likewise, the term "embodiments of the invention" does not require that all embodiments of the invention include the discussed feature, advantage or mode of operation.

[0013] Generally referring to FIGS. **1-3**, a propane vapor delivery system may be described and shown. The fuel vapor delivery system may be such that clean, economical use of an engine may be achieved with minimal complexity.

[0014] In one exemplary embodiment and as shown in the exemplary diagram in FIG. **1**, a gas, such liquefied petroleum gas or propane, may be supplied to an engine as fuel. The gas may be stored in a bottle or tank, such as LPG tank **102**. In a further exemplary embodiment, the gaseous form of liquefied propane may be used as a fuel for marine outboard four-stroke propulsion engines, as well as two stroke engines or any other type of engine with a pressurized lubricating injection system. An exemplary fuel system may use liquefied petroleum gas (LPG or propane) which may be stored in a vapor gaseous state in LPG tank **102** and which may not utilize engine coolant to keep regulators from freezing or otherwise malfunctioning.

[0015] LPG, as implemented or utilized in the exemplary embodiments described herein, may provide any of a number of advantages over other fuels. For example, LPG may act as a higher octane fuel source as well as decrease potentially harmful emissions while also offering substantially the same or better performance over other known fuels. An increase in fuel economy and a decrease in potentially harmful emissions may be up to 20% or more. Additionally, LPG may reduce wear on an engine and some components, for example extending the life of spark plugs, reducing varnish build-up on a carburetor and increasing the amount of time needed between oil changes. Further, LPG may be stored in any number of environments without the use of additives or unique storage conditions without affecting its usefulness or performance.

[0016] In a further exemplary embodiment, the LPG tank **102** may be connected to a regulator, such as a primary regulator **106**, converter or vaporizer. This connection may be made using a hose **104**, such as a high pressure hose, or any

form of pipe, for example a rigid pipe. The primary regulator **106** may step down the pressure of any stored propane vapor energy in tank **102**, and may further output this lowered pressure to a secondary regulator **110**. For example, the primary regulator **106** may be such that it reduces pressure of the propane from a high pressure state while the propane is in the LPG tank **102**, which may be approximately 180-250 psi, to a sub-atmospheric pressure, approximately 10 psi. The secondary regulator **110** can act as a vacuum-controlled flow rate regulator. Further, the secondary regulator **110** can meter propane gas to an engine **116**, for example, through a negative pressure signal, such as a vacuum signal, from the air-fuel mixer on a carburetor **114** designed or modified to use propane as a fuel source.

[0017] Still to exemplary FIG. **1**, and in a further example, the carburetor **114** may include both an air-fuel mixer and a throttle body. When the engine **116** is being cranked or otherwise running, a partial vacuum may be generated in the throttle body of carburetor **114**. This partial vacuum may be passed on from the throttle body to the fuel vapor hose to the secondary regulator **110**. The secondary regulator **110** may sense the vacuum and open, letting fuel vapor flow into the carburetor **114**. As the throttle is opened, a butterfly valve inside the carburetor **114** may open further, thus allowing for more fuel to be drawn in and allowing the engine **116** to accelerate or increase revolutions. Further, the air-valve implementation of the carburetor **114** may allow for a safe and accurate method of supplying propane gas from LPG tank **102** while also allowing for the fuel metering in the mixer to be self-contained. Additionally, it may be envisioned that a mechanical or electrical fuel injection system could be used in lieu of carburetor **114** while achieving similar results.

[0018] In further exemplary embodiments, and referring to FIG. **2**, the regulator arrangement of the primary regulator **106** and the secondary regulator **110** may be such that they are combined as a single, two-stage, regulator **202**. In this exemplary embodiment the combined regulator **202** would then feed the vaporized propane into carburetor **114** through hose **112**. Additionally, as shown in exemplary FIG. **2**, LPG tank **200**, in some examples, may be coupled directly to engine **116**, using any desired coupling, such as mounting bracket **204**.

[0019] The exemplary embodiment shown in FIG. **2** may be utilized in any desired situation, such on a small or single-cylinder outboard engine. The mounting of LPG tank **200** on the engine may provide many advantages. For example, this embodiment may be beneficial in some desired situations, for example when utilizing compact fuel supplies on a general yacht tender service or for any other similar-sized application, such as generators or personal watercraft where storage space or space of a fuel tank may be limited.

[0020] In still further exemplary embodiments, and referring to FIG. **3**, the LPG tank **300** may be located separate, but coupled to, an engine **310**. The LPG tank **300** may be connected to the primary regulator **304** as described previously, for example by connector **302**. Additionally, a low pressure output of the secondary regulator **308** may be attached to a propane carburetor (not pictured) on the engine **310** in any of a variety of manners, for example through hose **312** and coupling **314**. Further, a low pressure output of the secondary regulator **308** may be attached to the propane carburetor either permanently or releasably.

[0021] Still referring to exemplary FIG. **3**, the primary regulator **304** and the secondary regulator **308** may be two separate regulators that can be based on the design and size of an engine. The primary regulator **304** in this exemplary embodiment may be connected to LPG tank **300** as described above. Additionally, the primary regulator **304** may be connected to the secondary regulator **308** by a hose **306**, pipe, tube or the like, such as that described above.

[0022] In each of the exemplary embodiments described herein the arrangement of the regulators **106** and **110**, **202** and **304** and **308**, as shown in FIGS. **1-3**, respectively, can maintain about an ambient temperature from the surrounding air temperature rather than utilizing an interface with a cooling circuit of the engine **116** or other heat source to function as desired. In some other exemplary embodiments, the temperature maintained by any regulators described herein may be such that the flow of propane is allowed as desired. This may be accomplished as a result of the vapor delivery nature of the system as well as a design match of the system to fit specific air-fuel requirements of an engine size, for example for a marine engine. The maintenance of the regulator at about an ambient temperature may be facilitated by both the design of the regulator or regulators which may be sized to a specific engine, for example a marine engine having a known or desired horsepower rating. Additionally, the maintenance of the about ambient temperature may be such that the regulator or regulators are not affected, for example, by freezing as a result of the conversion of the LPG from tank **102** to a gaseous state and the flow of the fuel is not negatively affected or impacted.

[0023] In still further examples, any of the embodiments described herein may include a fuel shut-off valve, such as a safety fuel shut-off valve **316** or switch. For example, a standard propane tank shut-off valve may be utilized in some embodiments. Additionally, in other embodiments, such as embodiments where a larger, electrically actuated engine is utilized, an electric solenoid shut-off valve may be used.

[0024] In still other exemplary embodiments, for example engines as shown in FIG. **3**, which may be large or powerful engines, or any situation where more fuel may be desired, an LPG tank may be mounted or located in any desired position or location. Whereas the LPG tank shown in exemplary FIG. **2** may be mounted on the side of an engine, here other locations for mounting an LPG tank may be used as desired. As shown in FIG. **3**, a mounting bracket (not pictured) may be secured to LPG tank **300** and any other desired member of a vessel or craft associated with engine **310**. Also the LPG tank **300** may be formed out of any desired material, for example aluminum or a composite construction. Additionally, the LPG tank **300** may be coupled with any conduit described herein, such as a high pressure hoses, rigid tubes, pipes and the like in any desired manner, such as a screw fitting. Additionally, should it be desired, any number of fuel tanks may be coupled together to provide any desired amount of available fuel. Any coupling of additional tanks may be done in a manner known in the art.

[0025] Further, as described herein, the LPG applications may be used on any desired form of engine, such as marine outboard engines, and may further operate independently of an engine's cooling circuit. This may be accomplished through the use of vapor fuel delivery rather than the delivery of liquid propane gas. Additionally, on some engine types, such as engines of smaller displacement, smaller amounts of propane may be utilized to achieve a desired operation, thus

further issues pertaining to cooling and freezing, such as those known to affect the conversion of propane from liquid to gas, may be avoided.

[0026] The foregoing description and accompanying drawings illustrate the principles, preferred embodiments and modes of operation of the invention. However, the invention should not be construed as being limited to the particular embodiments discussed above. Additional variations of the embodiments discussed above will be appreciated by those skilled in the art.

[0027] Therefore, the above-described embodiments should be regarded as illustrative rather than restrictive. Accordingly, it should be appreciated that variations to those embodiments can be made by those skilled in the art without departing from the scope of the invention as defined by the following claims.

1. A fuel delivery system, comprising:
a fuel tank that houses propane;
a first regulator coupled to the fuel tank;
a second regulator coupled to the first regulator; and
a fuel metering system coupled to the second regulator to deliver propane vapor to an engine.

2. The fuel delivery system of claim **1**, wherein the engine is a marine outboard engine.

3. The fuel delivery system of claim **1**, wherein the fuel is reduced in pressure by the first regulator.

4. The fuel delivery system of claim **1**, wherein the second regulator receives a signal from the fuel metering system and delivers fuel to the fuel metering system.

5. The fuel delivery system of claim **4**, wherein the signal is a vacuum signal.

6. The fuel delivery system of claim **1**, wherein the fuel metering system is a carburetor.

7. The fuel delivery system of claim **1**, wherein the fuel is propane.

8. The fuel delivery system of claim **1**, wherein the first regulator and the second regulator are combined as a two stage regulator.

9. The fuel delivery system of claim **1**, wherein the coupling of the first regulator with the second regulator is such that fuel flowing between the first regulator and the second regulator is kept at an about ambient temperature.

10. The fuel delivery system of claim **1**, wherein the second regulator is coupled to the fuel metering system with a releasable coupling.

11. The fuel delivery system of claim **1**, further comprising at least one shut-off valve.

12. A method for providing fuel to an engine, comprising:
storing liquid propane in a tank;
flowing propane to a first regulator;
decreasing the pressure of the propane by the first regulator;
flowing the propane to a second regulator;
metering the flow of the propane from the second regulator to an engine; and
maintaining the temperature of the propane at an about ambient temperature.

13. The method of claim **12**, wherein the flow of the propane vapor from the second regulator to the engine is metered by a carburetor.

14. The method of claim **12**, further comprising reducing the pressure of the liquid propane in the tank from about 180-250 psi to about 4-12 psi by the first regulator.

15. The method of claim **12**, further comprising one of releasably coupling the second regulator to the engine and releasably coupling the first regulator to the second regulator.

16. The method of claim **12**, wherein the flow of the propane vapor from the second regulator to the engine is metered by a vacuum signal.

17. The method of claim **12**, wherein the flow of the propane vapor from the second regulator to the engine is metered by a carburetor.

18. The method of claim **12**, wherein the flow of the propane vapor from the second regulator to the engine is metered by a fuel injection system.

19. The method of claim **12**, further comprising securing the tank to the engine.

20. The method of claim **12**, wherein the engine is a marine outboard engine.

* * * * *

EXHIBIT 1011

Gasoline-engine management
Basics and components

BOSCH

Automotive Technology

- EGAS electronic throttle control
- Gasoline direct injection
- NO_x accumulator-type catalytic converter



Exhibit 1003, Page 766
Petitioner MWE Investments, LLC, et al.

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Robert Bosch GmbH

Gasoline-engine management
Basics and components

Bosch

Exhibit 1003, Page 768
Petitioner MWE Investments, LLC, et al.

Robert Bosch GmbH

Variable valve timing

Apart from using the throttle-valve to throttle the flow of incoming fresh gas drawn in by the engine, there are several other possibilities for influencing the cylinder charge. The proportion of fresh gas and of residual gas can also be influenced by applying variable valve timing.

Of great importance for valve timing is the fact that the behaviour of the gas columns flowing into and out of the cylinders varies considerably as a function of engine speed or throttle-valve opening. With invariable valve timing, therefore, this means that the exhaust and refill cycle can only be ideal for one single engine operating range. Variable valve timing, on the other hand, permits adaptation to a variety of different engine speeds and cylinder charges. This has the following advantages:

- Higher engine outputs,
- Favorable torque curve throughout a wide engine-speed range,
- Reduction of toxic emissions,
- Reduced fuel consumption,
- Reduction of engine noise.

1 Camshaft adjustment

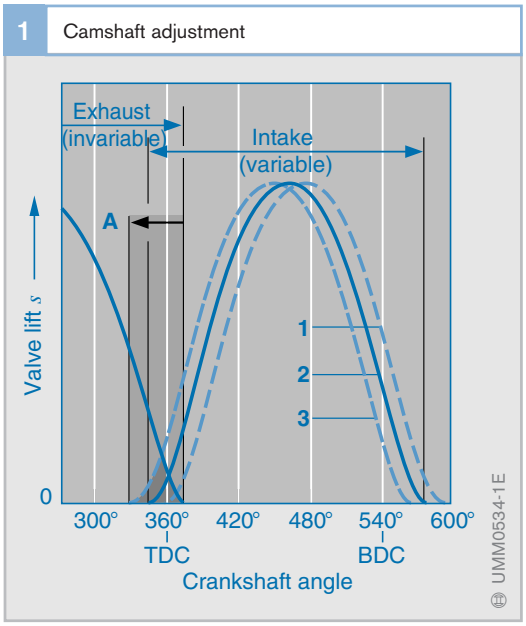


Figure 1
1 Camshaft retarded
2 Camshaft normal
3 Camshaft advanced
A Valve overlap

Camshaft phase adjustment

In conventional IC engines, camshaft and crankshaft are mechanically coupled to each other through toothed belt or chain. This coupling is invariable.

On engines with camshaft adjustment, at least the intake camshaft, but to an increasing degree the exhaust camshaft as well, can be rotated referred to the crankshaft so that valve overlap changes. The valve opening period and lift are not affected by camshaft phase adjustment, which means that "intake opens" and "intake closes" remain invariably coupled with each other.

The camshaft is adjusted by means of electrical or electro-hydraulic actuators. On less sophisticated systems provision is only made for two camshaft settings. Variable camshaft adjustment on the other hand permits, within a given range, infinitely variable adjustment of the camshaft referred to the crankshaft.

Fig. 1 shows how the "position", or lift, of the open intake-valve changes (referred to TDC) when the intake camshaft is adjusted.

Retard adjustment of the intake camshaft

Retarding the intake camshaft leads to the intake valve opening later so that valve overlap is reduced, or there is no valve overlap at all. At low engine speeds (<2000 min^{-1}), this results in only very little burnt exhaust gas flowing past the intake valve and into the intake manifold. At low engine speeds, the low residual exhaust-gas content in the intake of A/F mixture which then follows leads to a more efficient combustion process and a smoother idle. This means that the idle speed can be reduced, a step which is particularly favorable with respect to fuel consumption.

Robert Bosch GmbH

Ignition coil

Assignment

The ignition coil stores the required ignition energy and generates the high voltage for the spark flashover at the ignition point.

Design

Today's state-of-the-art ignition coils are comprised of two magnetically-coupled copper windings (primary and secondary windings), an iron core assembled from sheet-metal laminations, and a plastic case. Depending upon design, the core can be of either the closed type (compact coil), or of the rod type (rod-type coil). The arrangement and location of the primary and secondary windings depends upon the coil's shape. In order to increase the insulation resistance, the secondary winding can be designed as a disc or chamber winding.

So as to ensure efficient insulation between primary and secondary winding, and between the windings and the case, the case is filled with epoxy resin. The design and construction of the ignition coil are adapted to the application in question.

Operating concept

The ignition coil functions according to Faraday's Law. The energy stored in the primary winding's magnetic field is transferred to the secondary winding by magnetic induction. Depending upon the turns ratio, voltage and current are transferred from the primary to the secondary winding (Fig. 2).

On the single-ended ignition coils for systems with rotating high-voltage distribution, one of the primary-winding terminals is connected to one of the secondary-winding terminals and then to Terminal 15 of the driving switch (economy connection). The other end of the secondary winding is connected to the ignition driver stage (Terminal 1). The secondary winding's other connection goes to the ignition distributor (Terminal 4). On the double-ended and dual-spark ignition coils used on ignition systems with static voltage distribution, primary and secondary windings are not connected. On the double-ended ignition coil, one end of the secondary winding (Term. 4a) is connected to ground, while the other end is directly connected to the spark plug. On the dual-spark ignition coil, each secondary-winding terminal is connected to a spark plug.

High-voltage generation

On modern ignition systems, the engine-management ECU switches on the ignition driver stage for the calculated dwell period, during which the coil's primary current increases to its desired value and in the process generates a magnetic field.

The magnitude of the primary current, together with the ignition coil's primary inductance, are decisive for the energy stored in this magnetic field.

2 Ignition coils: Schematic representations

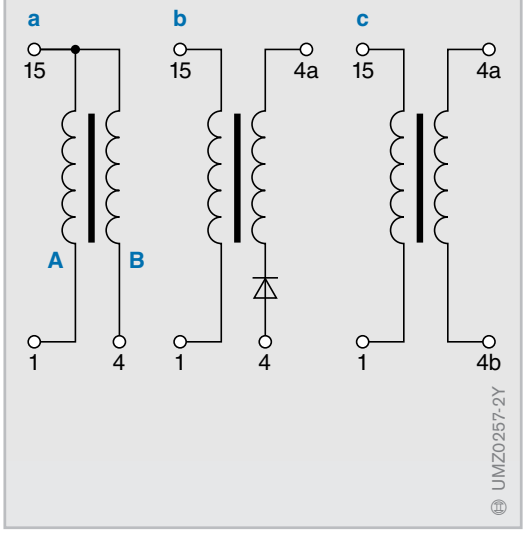


Figure 2
For rotating high-voltage distribution:
a Single-ended ignition coil

For static high-voltage distribution
b Double-ended ignition coil
c Dual-spark ignition coil

A Primary winding
B Secondary winding

EXHIBIT 1012

THIRD EDITION

ENGINE TESTING



A. J. Martyr • M. A. Plint

Exhibit 1003, Page 772
Petitioner MAE Investments, LLC, et al.

Engine Testing
Theory and Practice

Third edition

A.J. Martyr
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The essential role of damping will be clear from Fig. 3.6: it limits the potentially damaging amplitude of vibration at resonance. The ordinate in Fig. 3.6 is often described as the transmissibility of the mounting system: it is a measure of the extent to which the disturbing force f is reduced by the action of the flexible mounts. It is considered good practice to design the system so that the minimum speed at which the machine is to run is not less than three times the natural frequency, corresponding to a transmissibility of about 0.15. It should be noticed that once the frequency ratio exceeds about 2 the presence of damping actually has an adverse effect on the isolation of disturbing forces.

Practical considerations in the design of engine and test bed mountings

In the above simple treatment we have only considered oscillations in the vertical direction. In practice, as has already been pointed out, an engine carried on flexible mountings has six degrees of freedom (Fig. 3.1). While in many cases a simple analysis of vibrations in the vertical direction will give a satisfactory result, under test cell conditions a more complete computer analysis of the various modes of vibration and the coupling between them may be advisable. This is particularly the case with tall engines with mounting points at a low level, when cyclic variations in torque may induce transverse rolling of the engine.

Reference 6 lists the design factors to be considered in planning a system for the isolation and control of vibration and transmitted noise:

- specification of force isolation
  - as attenuation, dB
  - as transmissibility
  - as efficiency
  - as noise level in adjacent rooms
- natural frequency range to achieve the level of isolation required
- load distribution of the machine
  - is it equal on each mounting?
  - is the centre of gravity low enough for stability?
  - exposure to forces arising from connecting services, exhaust system, etc.
- vibration amplitudes – low frequency
  - normal operation
  - fault conditions
  - starting and stopping
  - is a seismic block or sub-base needed?

- higher-frequency structure-borne noise (100 Hz+)
  - is there a specification?
  - details of building structure
  - sufficient data on engine and associated plant
- transient forces
  - shocks, earthquakes, machine failures
- environment
  - temperature
  - humidity
  - fuel and oil spills.

Detailed design of engine mountings for test bed installation is a highly specialized matter, see Ker-Wilson[1] for guidance on standard practice. In general, the aim is to avoid 'coupled' vibrations, e.g. the generation of pitching forces due to unbalanced forces in the vertical direction, or the generation of rolling moments due to the torque reaction forces exerted by the engine. These can give rise to resonances at much higher frequencies than the simple frequency of vertical oscillation calculated in the following section and to consequent trouble, particularly with the engine-to-brake connecting shaft.

Massive foundations and supported bedplates

The analysis and prediction of the extent of transmitted vibration to the surroundings is a highly specialized field. The theory is dealt with in Ref. 1, the starting point being the observation that a heavy block embedded in the earth has a natural frequency of vibration that generally lies within the range 1000 to 2000 c.p.m. There is thus a possibility of vibration being transmitted to the surroundings if exciting forces, generally associated with the reciprocating masses in the engine, lie within this frequency range. An example would be a four cylinder four-stroke engine running at 750 rev/min. We see from Table 3.1 that such an engine generates substantial second-order forces at twice engine speed or 1500 c.p.m. Figure 3.7, redrawn from Ref. 1, gives an indication of acceptable levels of transmitted vibration from the point of view of physical comfort.

Figure 3.8 is a sketch of a typical seismic block. Reinforced concrete weighs roughly 2500 kg/m^3 and this block would weigh about 4500 kg. Note that the surrounding tread plates must be isolated from the block, also that it is essential to electrically earth (ground) the mounting rails. The block is shown carried on four combined steel spring and rubber isolators, each having a stiffness of 100 kg/mm (Fig. 3.9). From eq. 2a, the natural frequency of vertical oscillation of the bare block would be 4.70 Hz or 282 c.p.m., so the block would be a suitable base for an engine running at about 900 rev/min or faster. If the engine weight were, say, 500 kg

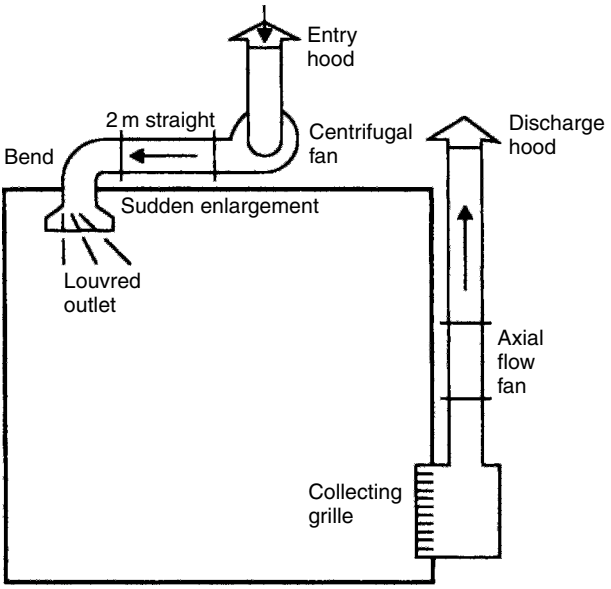


Figure 5.9 *Simplified cell ventilation system layout used in example calculation. Purge air is taken out in combined extract duct situated low in cell*

giving, from eqs (13) and (14)

$$\text{shaft power} = \frac{7.0 \times 454}{0.65} = 4900\,\text{W}$$

$$\text{motor power} = 5\,\text{kW}$$

For the extraction fan, an axial flow unit has the following specification:

Diameter	1 m
Speed	960 rev/min
Velocity pressure	51 Pa
Fan total efficiency	80%

$$\text{shaft power} = \frac{7.0 \times (112 + 51)}{0.80} = 1400\,\text{W}$$

In this case, the manufacturer recommends a motor rated at 2.2 kW.

Ventilation of the control room

This is in general a much less demanding exercise for the test installation designer. Heating loads are moderate, primarily associated with lights and heat generated by

Table 5.6 *Calculation of system pressure losses*

Item	*Size* (*mm*)	*Area* (m^2)	*Volume flow rate* (m^3/s)	*Velocity* (*m/s*)	*Velocity pressure* (*Pa*)	*Fitting loss factor* K_e	*Pressure drop* K_e/m	*Length* (*m*)	*Pressure loss* (*Pa*)	*Cumulative loss* (*Pa*)
Forced draught										
Entry hood	800 dia	0.5	7.0	14	118	1.1			129	129
Straight	800 dia	0.5	7.0	14	118		0.02	2	5	134
Bend	800 dia	0.5	7.0	14	118	0.23			27	161
Fan										
Straight	600 × 600	0.36	7.0	19	217		0.025	2	11	172
Bend	600 × 600	0.36	7.0	19	217	0.23			50	222
Sudden enlargement	600 × 600	0.36	7.0	19	217	1.0			217	439
Louvred outlet	1000 × 1000	1.00	7.0	7	29	0.5			15	454
									Total	454
Extraction										
Collecting hood	1000 dia*	0.79	7.0	9	49	1.25			61	61
Straights	1000 dia	0.79	7.0	9	49		0.02	2	2	63
Fan										
Discharge hood	1000 dia	0.79	7.0	9	49	1.0			49	112
									Total	112

* Diameter of branch connection.

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electronic apparatus located in the room. Regulations regarding air flow rate per occupant and general conditions of temperature and humidity are laid down in various codes of practice and should be equivalent to those considered appropriate for offices.

Air conditioning

Most people associate this topic with comfort levels under various conditions: sitting, office work, manual work, etc. Levels of air temperature and humidity, as well as air change rates, are laid down by statute (see BS 5720[7] for particulars). Such regulations must be observed with regard to the control room. However, the conditions in an engine test cell are far removed from the normal, and justify special treatment, which follows.

Two properties of the ventilating air entering the cell (and, more particularly, of the induction air entering the engine) are of importance: the temperature and the moisture content. Air conditioning involves four main processes:

- heating the air;
- cooling the air;
- reducing the moisture content (dehumidifying);
- increasing the moisture content (humidifying).

Fundamentals of psychometry

The study of the properties of moist air is known as psychometry. It is treated in many standard texts[1,2] and only a very brief summary will be given here. Air conditioning processes are represented on the psychometric chart (Fig. 5.10).[9]

This relates the following properties of moist air:

- the *moisture content* or *specific humidity*, ω kg moisture/kg dry air. Note that even under fairly extreme conditions (saturated air at 30°C), the moisture content does not exceed 3 per cent by weight
- the *percentage saturation* or *relative humidity*, φ. This is the ratio of the mass of water vapour present to the mass that would be present if the air were saturated at the same conditions of temperature and pressure. The mass of vapour under saturated conditions is very sensitive to temperature:

Temperature °C	10	15	20	25	30
Moisture content ω (kg/kg)	0.0076	0.0106	0.0147	0.0201	0.0273

A consequence of this relationship is the possibility of drying air by cooling. As the temperature is lowered the percentage saturation increases until at the dew point temperature the air is fully saturated and any further cooling results in the deposition of moisture.

It will be observed that the energy requirements, particularly for the chiller either internal or from separate chilled water supply, are quite substantial. In the case of large engines and gas turbines any kind of combustion air conditioning is not really practicable and reliance must be placed on correction factors.

Any combustion air supply system must be integrated with the fire alarm and fire extinguishing system and must include the same provisions for isolating individual cells as the main ventilation system.

Important integration points

Combustion air units of all types will require condensate drain lines with free gravity drainage to a building system; back-up or overspill of this fluid can cause significant problems. Those fitted with steam generators will require both condensate and possibility steam 'blow-down' lines piped to safe drainage.

If steam generators are fitted, the condition of feed water will be critical and suitable water treatment plant needs to be installed, particularly where supply water is hard.

Pressure controlled units

A close coupled air supply duct attached to the engine is necessary in special cases, such as anechoic cells, where air intake noise must be eliminated, at sites running legislative tests that are situated at altitudes above those where standard conditions can be achieved and where a precisely controlled conditioned supply is needed.

Since engine air consumption varies more or less directly with speed and that parameters can vary more rapidly than any air supply system can respond, and since it is essential not to impose pressure changes on the engine air, some form of excess air spill or engine bypass strategy has to be adopted to obtain pressure control with any degree of temperature and humidity stability.

For testing at steady state conditions, the pressure control system can operate with a slow and well-damped characteristic; typically having a stabilization time, following an engine speed set-point change, of 30 seconds or more. This is well suited to systems based on regulating the pressure of an intake plenum fitted with a pressure controlling spill valve/flap.

The inlet duct must be of sufficient size to avoid an appreciable pressure drop during engine acceleration and it needs to be attached to the engine inlet in such a way as to provide a good seal without imposing forces on the engine.

It is sensible to encapsulate or connect to the normal engine air inlet filter, if it can sustain the imposed internal pressure imposed by the supply system. If the filter has to be encapsulated it often requires a bulky rig item that has to be suspended from a frame above the engine.

Various strategies have been adopted for fast pressure control, the best of which are able to stabilize and maintain pressure at set point within <2 seconds of an engine step change of 500 r.p.m.

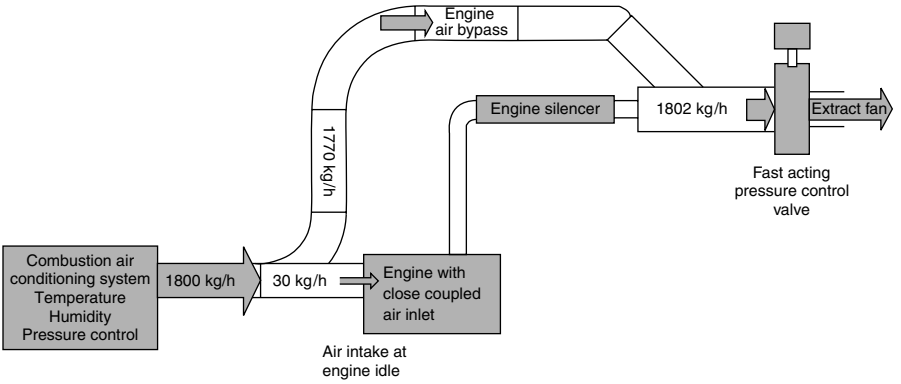


Figure 5.12 *AVL dynamic combustion air pressure control system. Example flows shown at engine idle condition*

Control of inlet pressure only does not simulate true altitude; this requires that both inlet and exhaust are pressurized at the same pressure, a condition that can be simulated using a dynamic pressure control system patented by AVL List GmbH as shown in Fig. 5.12.

Such systems find use in motor sport and other specialist applications, but where the 'ram effect' of the vehicle's forward motion is required, as is the case in F1 and motor cycle engine testing, then considerable energy is required to move the air and the test cell begins to resemble a wind tunnel, which is outside the remit of this book.

Charge (boost) air temperature control

The turbocharging process compresses the combustion air and raises its temperature, the effect of which is discussed in Chapter 12. Charge air cooling frequently has to be accommodated in an engine test cell and there are various strategies that may be used; one is to use the ex-vehicle charge air cooler as a rig item, the other is to use a 'fluid/charge air' cooler as a cell fixture.

Using forced air from a spot fan to cool the vehicle charge air cooler will introduce a great deal of additional heat energy into the cell; it is therefore good practice to use a liquid cooled charge air cooler having very similar flow resistance on the air side to the vehicle unit. A common strategy is to encase the original vehicle air/air intercooler in a special casing and use a water or water spray as the cooling medium. Temperature control is achieved by a flow control valve on the fluid circuit.

Summary

Design of the ventilation system for a test cell is a major undertaking and a careful and thorough analysis of expected heat loads from engine, exhaust system, dynamometer,

- Soot deposits and staining condensate in the fan and ducting are unsightly and make maintenance difficult.
- Other difficulties can arise, such as noise, variability in exhaust back pressure, etc.

If 'tail pipe' testers of the type used in vehicle servicing and inspection are to be employed, then easy access to the tail pipe end is necessary.

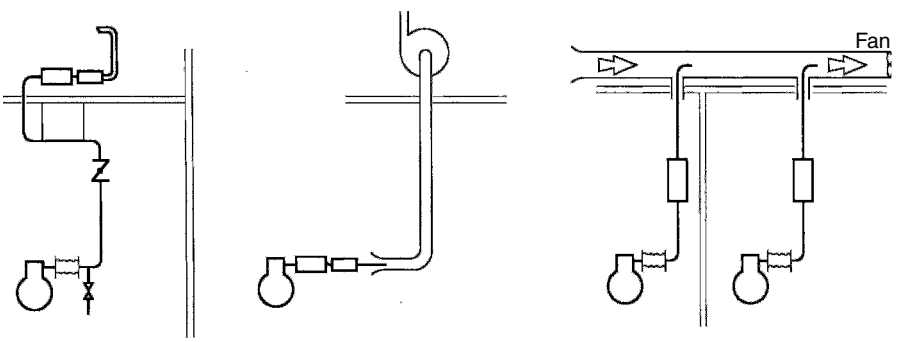
Test cell exhaust layouts may be classified as follows:

- individual cell, close coupled;
- individual cell, vehicle exhaust system, scavenged duct;
- multiple cells with common scavenged duct;
- specially designed emission cells (see Chapter 16).

Individual cell, close coupled

Such an arrangement is shown schematically in Fig. 6.5a, it may be regarded as the 'standard' arrangement for a general purpose test bed, and is also commonly used for production testing. The exhaust manifold is coupled to a flexible stainless steel pipe, of fairly large diameter to minimize pressure waves, and led by way of a back pressure regulating valve to a pipe system suspended from the cell roof. Condensate, which is highly corrosive, tends to collect in these pipes, which should be laid to a fall with suitable drainage arrangements.

In cells designed to occasionally use smoke or particulate analysis, it is usually a requirement that the sample is taken from the exhaust pipe at particular points; some devices have quite specific requirements regarding size, position and angle of probe insertion and it is desirable, in order to ensure a representative sample, that



(a) Individual cell, close coupled

(b) Individual cell, engine system and scavenge duct

(c) Multiple cells with common scavenge duct

Figure 6.5 *Exhaust systems*

9 Coupling the engine to the dynamometer

Introduction

The selection of suitable couplings and shaft for the connection of the engine to the dynamometer is by no means a simple matter. Incorrect choice or faulty system design may give rise to a number of problems:

- torsional oscillations;
- vibration of engine or dynamometer;
- whirling of coupling shaft;
- damage to engine or dynamometer bearings;
- excessive wear of shaft line components;
- catastrophic failure of coupling shafts;
- engine starting problems.

This whole subject, the coupling of engine and dynamometer, can give rise to more trouble than any other routine aspect of engine testing, and a clear understanding of the many factors involved is desirable. The following chapter covers all the main factors, but in some areas more extensive analysis may be necessary and appropriate references are given.

The nature of the problem

The special feature of the problem is that it must be considered afresh each time an engine not previously encountered is installed. It must also be recognized that unsatisfactory torsional behaviour is associated with the whole system – engine, coupling shaft and dynamometer – rather than with the individual components, all of which may be quite satisfactory in themselves.

Problems arise partly because the dynamometer is seldom equivalent dynamically to the system driven by the engine in service. This is particularly the case with vehicle engines. In the case of a vehicle with rear axle drive, the driveline consists of a clutch, which may itself act as a torsional damper, followed by a gearbox, the characteristics of which are low inertia and some damping capacity. This is followed

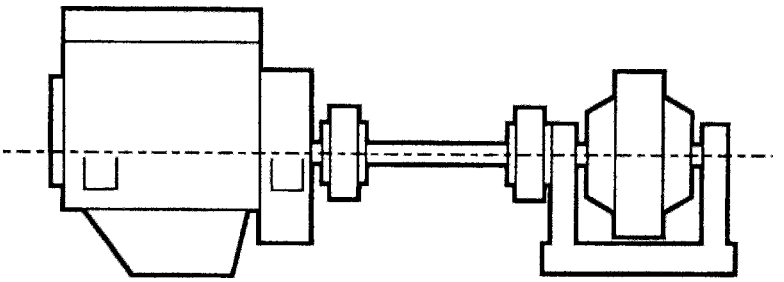


Figure 9.1 *Simple form of dynamometer/engine drive line*

by a drive shaft and differential, itself having appreciable damping, two half shafts and two wheels, both with substantial damping capacity and running at much slower speed than the engine, thus reducing their effective inertia.

When coupled to a dynamometer this system, Fig. 9.1, with its built-in damping and moderate inertia, is replaced by a single drive shaft connected to a single rotating mass, the dynamometer, running at the same speed as the engine. The clutch may or may not be retained.

Particular care is necessary where the moment of inertia of the dynamometer is more than about twice that of the engine. A further consideration that must be taken seriously concerns the effect of the difference between the engine mounting arrangements in the vehicle and on the testbed. This can lead to vibrations of the whole engine that can have a disastrous effect on the drive shaft.

Overhung mass on engine and dynamometer bearings

Care must be taken when designing and assembling a shaft system that the loads imposed by the mass and unbalanced forces do not exceed the overhung weight limits of the engine bearing at one end and the dynamometer at the other. Steel adaptor plates required to adapt the bolt holes of the shaft to the dynamometer flange or engine flywheel can increase the load on bearings significantly and compromise the operation of the system. Dynamometer manufacturers produce tables showing the maximum permissible mass at a given distance from the coupling face of their machines; the equivalent details for most engines is more difficult to obtain, but the danger of overload should be kept in mind by all concerned.

Background reading

The mathematics of the subject is complex and not readily accessible. Den Hartog[1] gives what is possibly the clearest exposition of fundamentals. Ker Wilson's classical treatment in five volumes[2] is probably still the best source of comprehensive information; his abbreviated version[3] is sufficient for most purposes. Mechanical

Engineering Publications have published a useful practical handbook[4] while Lloyd's Register[5] gives rules for the design of marine drives that are also useful in the present context. A listing of the notation used is to be found at the end of this chapter.

Torsional oscillations and critical speeds

In its simplest form, the engine–dynamometer system may be regarded as equivalent to two rotating masses connected by a flexible shaft, Fig. 9.2. Such a system has an inherent tendency to develop torsional oscillations. The two masses can vibrate 180° out of phase about a node located at some point along the shaft between them. The oscillatory movement is superimposed on any steady rotation of the shaft. The resonant or critical frequency of torsional oscillation of this system is given by:

$$n_c = \frac{60}{2\pi}\sqrt{\frac{C_c(I_e + I_b)}{I_e I_b}} \tag{1}$$

If an undamped system of this kind is subjected to an exciting torque of constant amplitude T_{ex} and frequency n, the relation between the amplitude of the induced oscillation θ and the ratio n/n_c is as shown in Fig. 9.3.

At low frequencies, the combined amplitude of the two masses is equal to the static deflection of the shaft under the influence of the exciting torque, $\theta_0 = T_{ex}/C_s$. As the frequency increases, the amplitude rises and at $n = n_c$ it becomes theoretically infinite: the shaft may fracture or non-linearities and internal damping may prevent actual failure. With further increases in frequency the amplitude falls and at $n - \sqrt{2n_c}$ it is down to the level of the static deflection. Amplitude continues to fall with increasing frequency.

The shaft connecting engine and dynamometer must be designed with a suitable stiffness C_s to ensure that the critical frequency lies outside the normal operating

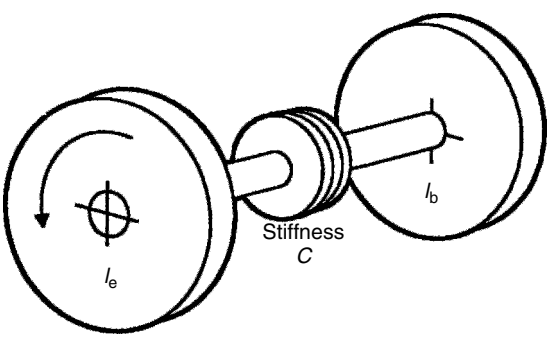


Figure 9.2 *Two mass system (compare with Fig. 9.1)*

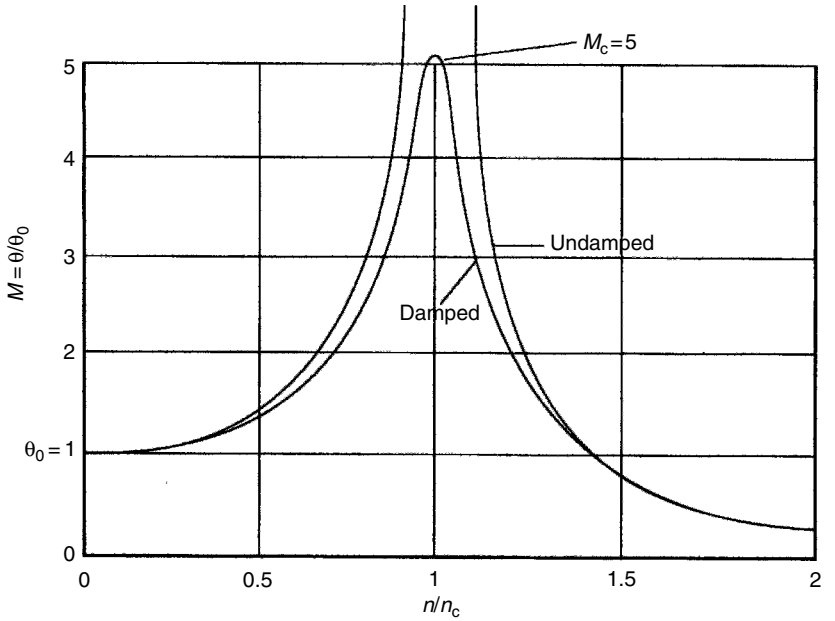


Figure 9.3 *Relationship between frequency ratio, amplitude and dynamic amplifier M*

range of the engine, and also with a suitable degree of damping to ensure that the unit may be run through the critical speed without the development of a dangerous level of torsional oscillation. Figure 9.3 also shows the behaviour of a damped system. The ratio θ/θ_0 is known as the dynamic magnifier M. Of particular importance is the value of the dynamic magnifier at the critical frequency, M_c. The curve of Fig. 9.3 corresponds to a value $M_c = 5$.

Torsional oscillations are excited by the variations in engine torque associated with the pressure cycles in the individual cylinders (also, though usually of less importance, by the variations associated with the movement of the reciprocating components).

Figure 9.4 shows the variation in the case of a typical single cylinder four-stroke diesel engine. It is well known that any periodic curve of this kind may be synthesized from a series of *harmonic components*, each a pure sine wave of a different amplitude having a frequency corresponding to a multiple or submultiple of the engine speed and Fig. 9.4 shows the first six components.

The *order* of the harmonic defines this multiple. Thus a component of order $N_0 = 1/2$ occupies two revolutions of the engine, $N_0 = 1$ one revolution and so on. In the case of a four cylinder four-stroke engine, there are two firing strokes per revolution of the crankshaft and the turning moment curve of Fig. 9.4 is repeated at intervals of 180°. In a multicylinder engine, the harmonic components of a given

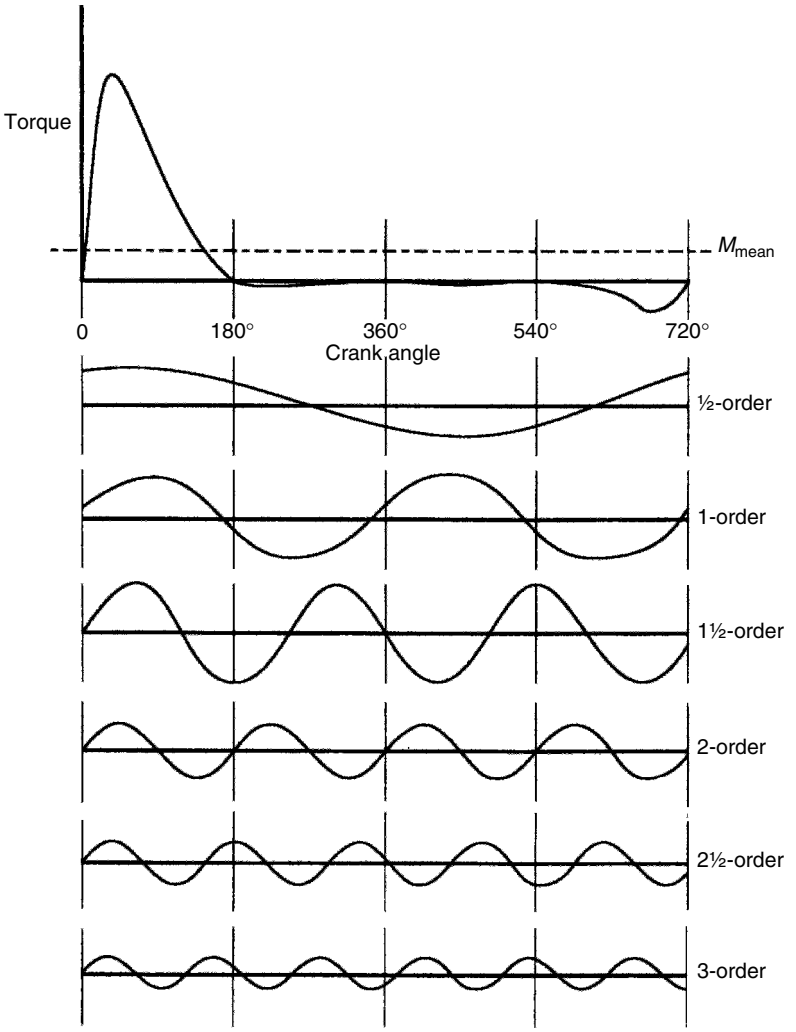


Figure 9.4 *Harmonic components of turning moment, single cylinder four-stoke gasoline engine*

order for the individual cylinders are combined by representing each component by a vector, in the manner illustrated in Chapter 3, for the inertia forces. A complete treatment of this process is beyond the scope of this book, but the most significant results may be summarized as follows.

The first major critical speed for a multicylinder, in-line engine is of order:

$$N_0 = N_{CYL}/2 \quad \text{for a four-stroke engine} \tag{2a}$$

$$N_0 = N_{CYL} \quad \text{for a two-stroke engine} \tag{2b}$$

Thus, in the case of a four cylinder, four-stroke engine the major critical speeds are of order 2, 4, 6, etc. In the case of a six cylinder engine, they are of order 3, 6, 9, etc.

The distinction between a major and a minor critical speed is that in the case of an engine having an infinitely rigid crankshaft it is only at the major critical speeds that torsional oscillations can be induced. This, however, by no means implies that in large engines having a large number of cylinders, the minor critical speeds may be ignored.

At the major critical speeds the exciting torques T_{ex} of all the individual cylinders in one line act in phase and are thus additive (special rules apply governing the calculation of the combined excitation torques for Vee engines).

The first harmonic is generally of most significance in the excitation of torsional oscillations, and for engines of moderate size, such as passenger vehicle engines, it is generally sufficient to calculate the critical frequency from eq. (1), then to calculate the corresponding engine speed from:

$$N_c = n_c/N_0 \tag{3}$$

The stiffness of the connecting shaft between engine and dynamometer should be chosen so that this speed does not lie within the range in which the engine is required to develop power.

In the case of large multicylinder engines, the 'wind-up' of the crankshaft as a result of torsional oscillations can be very significant and the two-mass approximation is inadequate; in particular, the critical speed may be overestimated by as much as 20 per cent and more elaborate analysis is necessary. The subject is dealt with in several different ways in the literature; perhaps the easiest to follow is that of Den Hartog.[1] The starting point is the value of the mean turning moment developed by the cylinder, M_{mean} (Fig. 9.4). Values are given for a so-called 'p factor', by which M_{mean} is multiplied to give the amplitude of the various harmonic excitation forces. Table 9.1, reproduced from Den Hartog, shows typical figures for a four-stroke medium speed diesel engine.

Exciting torque:

$$T_{ex} = p.\ M_{mean} \tag{4}$$

Table 9.1 *p factors*

Order	$\frac{1}{2}$	1	$1\frac{1}{2}$	2	$2\frac{1}{2}$	3. . .	8
p factor	2.16	2.32	2.23	1.91	1.57	1.28	0.08

The relation between M_{mean} and imep (**i**ndicated **m**ean **e**ffective **p**ressure) is given by:

$$\text{for a four-stroke engine} \quad M_{mean} = p_i.\frac{B^2 S}{16}.10^{-4} \tag{5a}$$

$$\text{for a two-stroke engine} \quad M_{mean} = p_i.\frac{B^2 S}{8}.10^{-4} \tag{5b}$$

Lloyd's *Rulebook*,[5] the main source of data on this subject, expresses the amplitude of the harmonic components rather differently, in terms of a 'component of tangential effort', T_m. This is a pressure that is assumed to act upon the piston at the crank radius $S/2$. Then exciting torque per cylinder:

$$T_{ex} = T_m \frac{\pi B^2}{4} \frac{S}{2} \times 10^{-4} \tag{6}$$

Lloyd's give curves of T_m in terms of the indicated mean effective pressure p_i and it may be shown that the values so obtained agree closely with those derived from Table 9.1.

The amplitude of the vibratory torque T_v induced in the connecting shaft by the vector sum of the exciting torques for all the cylinders, $\sum T_{ex}$ is given by:

$$T_v = \frac{\sum T_{ex} M_c}{(1 + I_e/I_b)} \tag{7}$$

The complete analysis of the torsional behaviour of a multicylinder engine is a substantial task, though computer programmes are available which reduce the effort required. As a typical example, Fig. 9.5 shows the 'normal elastic curves' for the first and second modes of torsional oscillation of a 16 cylinder Vee engine coupled to a hydraulic dynamometer. These curves show the amplitude of the torsional oscillations of the various components, relative to that at the dynamometer which is taken as unity. The natural frequencies are respectively $n_c = 4820$ c.p.m. and $n_c = 6320$ c.p.m. The curves form the basis for further calculations of the energy input giving rise to the oscillation. In the case of the engine under consideration, these showed a very severe fourth-order oscillation, $N_0 = 4$, in the first mode. (For an engine having eight cylinders in line the first major critical speed, from eq. (2a), is of order $N_0 = 4$.) The engine speed corresponding to the critical frequency of torsional oscillation is given by:

$$N_c = n_c / N_0 \tag{8}$$

giving, in the present case, $N_c = 1205$ rev/min, well within the operating speed range of the engine. Further calculations showed a large input of oscillatory energy at $N_0 = 4½$, a minor critical speed, in the second mode, corresponding to a critical engine speed of $6320/4¼ = 1404$ rev/min, again within the operating range. Several

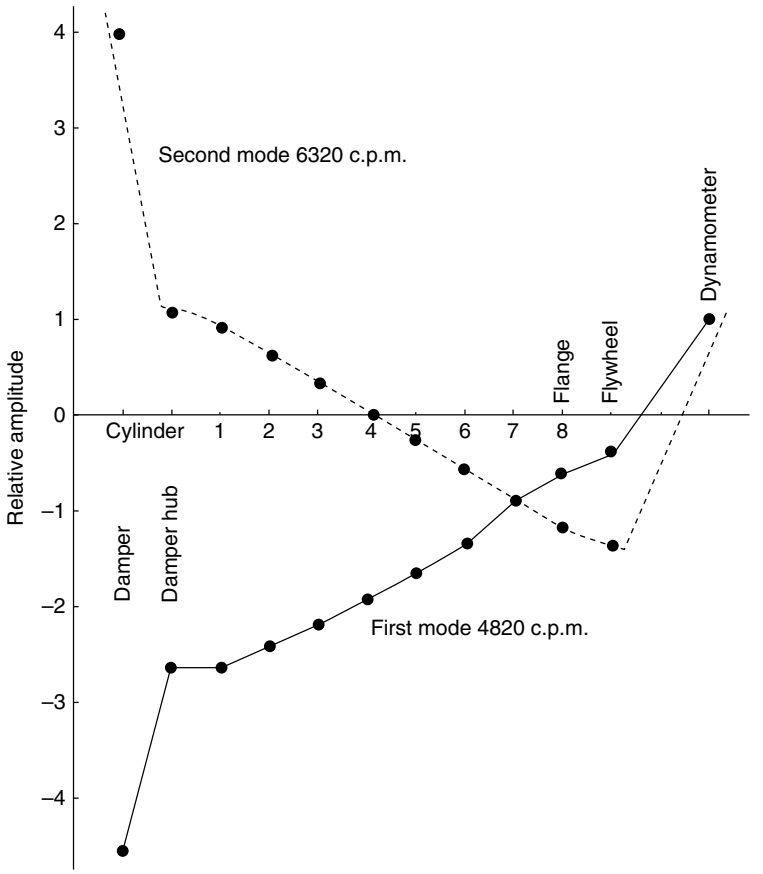


Figure 9.5 *Normal elastic curves for a particular 16 cylinder V engine coupled to a hydraulic dynamometer (taken from an actual investigation)*

failures of the shaft connecting engine and dynamometer occurred before a safe solution was arrived at.

This example illustrates the need for caution and for full investigation in setting up large engines on the test bed.

It is not always possible to avoid running close to or at critical speeds and this situation is usually dealt with by the provision of torsional vibration dampers, in which the energy fed into the system by the exciting forces is absorbed by viscous shearing. Such dampers are commonly fitted at the non-flywheel end of engine crankshafts. In some cases it may also be necessary to consider their use as a component of engine test cell drive lines, when they are located either as close as possible to the engine flywheel, or at the dynamometer. The damper must be 'tuned' to be most effective at the critical frequency and the selection of a suitable damper involves equating the

energy fed into the system per cycle with the energy absorbed by viscous shear in the damper. This leads to an estimate of the magnitude of the oscillatory stresses at the critical speed. For a clear treatment of the theory, see Den Hartog.[1]

Points to remember:

- As a general rule, it is good practice to avoid running the engine under power at speeds between 0.8 and 1.2 times critical speed. If it is necessary to take the engine through the critical speed, this should be done off load and as quickly as possible. With high inertia dynamometers the transient vibratory torque may well exceed the mechanical capacity of the drive line and the margin of safety of the drive line components may need to be increased.
- Problems frequently arise when the inertia of the dynamometer much exceeds that of the engine: a detailed torsional analysis is desirable when this factor exceeds 2. This situation usually occurs when it is found necessary to run an engine of much smaller output than the rated capacity of the dynamometer.
- the simple 'two mass' approximation of the engine-dynamometer system is inadequate for large engines and may lead to overestimation of the critical speed.

Design of coupling shafts

The maximum shear stress induced in a shaft, diameter D, by a torque T Nm is given by:

$$\tau = \frac{16T}{\pi D^3} \text{ Pa} \tag{9a}$$

In the case of a tubular shaft, bore diameter d, this becomes:

$$\tau = \frac{16TD}{\pi(D^4 - d^4)} \text{ Pa} \tag{9b}$$

For steels, the shear yield stress is usually taken as equal to $0.75 \times$ yield stress in tension. A typical choice of material would be a nickel–chromium–molybdenum steel, to specification BS 817M40 (previously En 24) heat-treated to the 'T' condition.

The various stress levels for this steel are roughly as follows:

ultimate tensile strength	not less than 850 MPa (55 t.s.i.)
0.1% proof stress in tension	550 MPa
ultimate shear strength	500 MPa
0.1% proof stress in shear	300 MPa
shear fatigue limit in reversed stress	±200 MPa

It is clear that the permissible level of stress in the shaft will be a small fraction of the ultimate tensile strength of the material.

The choice of designed stress level at the maximum rated steady torque is influenced by two principal factors.

Stress concentrations, keyways and keyless hub connection

For a full treatment of the very important subject of stress concentration see Ref. 6. There are two particularly critical locations to be considered:

- At a shoulder fillet, such as is provided at the junction with an integral flange. For a ratio fillet radius/shaft diameter = 0.1 the stress concentration factor is about 1.4, falling to 1.2 for $r/d = 0.2$.
- At the semicircular end of a typical rectangular keyway, the stress concentration factor reaches a maximum of about 2.5× nominal shear stress at an angle of about 50° from the shaft axis. The authors have seen a number of failures at this location and angle.

Cyclic stresses associated with torsional oscillations is an important consideration and as, even in the most carefully designed installation involving an internal combustion engine, some torsional oscillation will be present, it is wise to select a conservative value for the nominal (steady state) shear stress in the shaft.

In view of the stress concentration inherent in shaft keyways and the backlash present that can develop in splined hubs, the use of keyless hub connection systems of the type produced by the Ringfeder Corporation are now widely used. These devices are supported by comprehensive design documentation; however, the actual installation process must be exactly followed for the design performance to be ensured.

Stress concentration factors apply to the cyclic stresses rather than to the steady state stresses. Figure 9.6 shows diagrammatically the Goodman diagram for a steel having the properties specified above. This diagram indicates approximately the

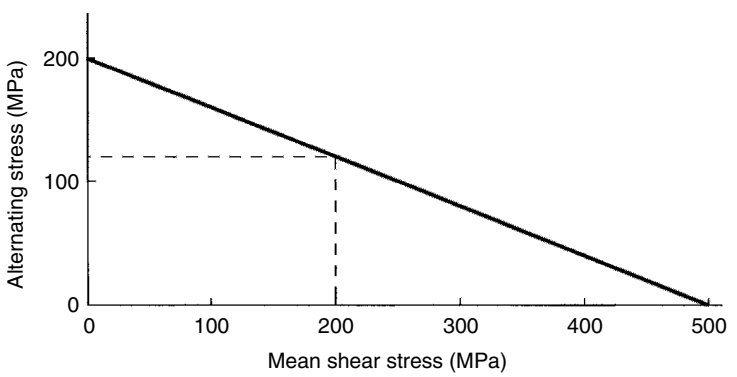


Figure 9.6 *Goodman diagram, steel shaft in shear*

relation between the steady shear stress and the permissible oscillatory stress. The example shown indicates that, for a steady torsional stress of 200 MPa, the accompanying oscillatory stress (calculated after taking into account any stress concentration factors) should not exceed ±120 MPa. In the absence of detailed design data, it is good practice to design shafts for use in engine test beds very conservatively, since the consequences of shaft failure can be so serious. A shear stress calculated in accordance with eq. (9) of about 100 MPa for a steel with the properties listed should be safe under all but the most unfavourable conditions. To put this in perspective, a shaft 100 mm diameter designed on this basis would imply a torque of 19 600 Nm, or a power of 3100 kW at 1500 rev/min.

The torsional stiffness of a solid shaft of diameter D and length L is given by:

$$C_s = \frac{\pi D^4 G}{32L} \tag{10a}$$

for a tubular shaft, bore d:

$$C_s = \frac{\pi (D^4 - d^4)}{32L} \tag{10b}$$

Shaft whirl

The coupling shaft is usually supported at each end by a universal joint or flexible coupling. Such a shaft will 'whirl' at a rotational speed N_w (also at certain higher speeds in the ratio $2^2\ N_w$, $3^2\ N_w$, etc.).

The whirling speed of a solid shaft of length L is given by:

$$N_w = \frac{30\pi}{L^2}\sqrt{\frac{E\pi D^4}{64 W_s}} \tag{11}$$

It is desirable to limit the maximum engine speed to about $0.8\,N_w$. *When using rubber flexible couplings it is essential to allow for the radial flexibility of these couplings, since this can drastically reduce the whirling speed.* It is sometimes the practice to fit self-aligning rigid steady bearings at the centre of flexible couplings in high-speed applications, but these are liable to give fretting problems and are not universally favoured.

As is well known, the whirling speed of a shaft is identical with its natural frequency of transverse oscillation. To allow for the effect of transverse coupling flexibility the simplest procedure is to calculate the transverse critical frequency of the shaft plus two half couplings from the equation:

$$N_t = \frac{30}{\pi}\sqrt{\frac{k}{W}} \tag{12a}$$

where W = mass of shaft + half couplings and k = combined radial stiffness of the two couplings.

Then whirling speed N taking this effect into account will be given by:

$$\left(\frac{1}{N}\right)^2 = \left(\frac{1}{N_w}\right)^2 + \left(\frac{1}{N_t}\right)^2 \tag{12b}$$

Couplings

The choice of the appropriate coupling for a given application is not easy: the majority of drive line problems probably have their origin in an incorrect choice of components for the drive line, and are often cured by changes in this region. A complete discussion would much exceed the scope of this book, but the reader concerned with drive line design should obtain a copy of Ref. 4, which gives a comprehensive treatment together with a valuable procedure for selecting the best type of coupling for a given application. A very brief summary of the main types of coupling follows.

Quill shaft with integral flanges and rigid couplings

This type of connection is best suited to the situation where a driven machine is permanently coupled to the source of power, when it can prove to be a simple and reliable solution. It is not well suited to test bed use, since it is intolerant of relative vibration and misalignment.

Quill shaft with toothed or gear type couplings

Gear couplings are very suitable for high powers and speeds, and can deal with relative vibration and some degree of misalignment, but this must be very carefully controlled to avoid problems of wear and lubrication. Lubrication is particularly important as once local tooth to tooth seizure takes place deterioration may be rapid and catastrophic. Such shafts are inherently stiff in torsion.

Conventional 'cardan shaft' with universal joints

These shafts are readily available from a number of suppliers, and are probably the preferred solution in the majority of cases. However, standard automotive type shafts can give trouble when run at speeds in excess of those encountered in vehicle applications. A correct 'built-in' degree of misalignment is necessary to avoid fretting of the needle rollers.

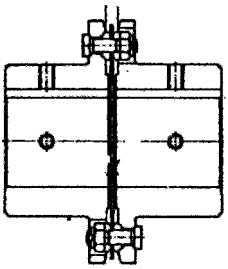


Figure 9.7 *Multiple steel disc type flexible coupling*

Multiple membrane couplings

These couplings, Fig. 9.7, are stiff in torsion but tolerant of a moderate degree of misalignment and relative axial displacement. They can be used for very high speeds.

Elastomeric element couplings

There is a vast number of different designs on the market and selection is not easy. Ref. 8 may be helpful. The great advantage of these couplings is that their torsional stiffness may be varied widely by changing the elastic elements and problems associated with torsional vibrations and critical speeds dealt with (see the next section).

Damping: the role of the flexible coupling

The earlier discussion leads to two main conclusions: the engine–dynamometer system is susceptible to torsional oscillations and the internal combustion engine is a powerful source of forces calculated to excite such oscillations. The magnitude of these undesirable disturbances in any given system is a function of the damping capacities of the various elements: the shaft, the couplings, the dynamometer and the engine itself.

The couplings are the only element of the system the damping capacity of which may readily be changed, and in many cases, for example with engines of automotive size, the damping capacity of the remainder of the system may be neglected, at least in an elementary treatment of the problem, such as will be given here.

The dynamic magnifier M (Fig. 9.3) has already been mentioned as a measure of the susceptibility of the engine–dynamometer system to torsional oscillation. Now referring to Fig. 9.1, let us assume that there are two identical flexible couplings, of stiffness C_c, one at each end of the shaft, and that these are the only sources of damping. Figure 9.8 shows a typical torsionally resilient coupling in which torque is transmitted by way of a number of shaped rubber blocks or bushes which provide torsional flexibility, damping and a capacity to take up misalignment. The torsional characteristics of such a coupling are shown in Fig. 9.9.

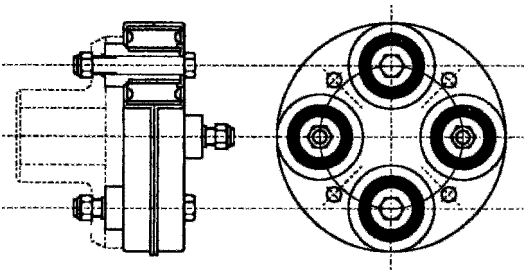


Figure 9.8 *Rubber bush type torsionally resilient coupling*

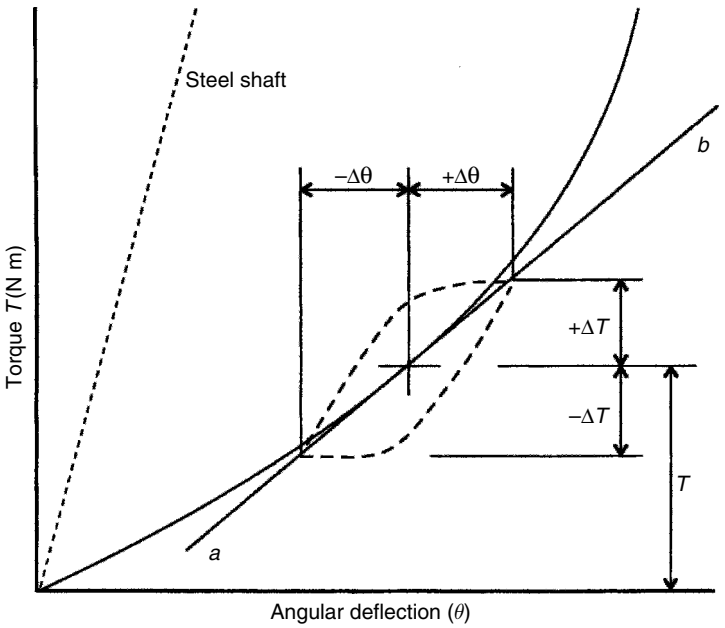


Figure 9.9 *Dynamic torsional characteristic of multiple bush type coupling*

These differ in three important respects from those of, say, a steel shaft in torsion:

1. The coupling does not obey Hooke's law: the stiffness or coupling rate $C_c = \Delta T/\Delta\theta$ increases with torque. This is partly an inherent property of the rubber and partly a consequence of the way it is constrained.
2. The shape of the torque–deflection curve is not independent of frequency. Dynamic torsional characteristics are usually given for a cyclic frequency of 10 Hz. If the load is applied slowly the stiffness is found to be substantially less. The following values of the ratio dynamic stiffness (at 10 Hz) to static stiffness of natural rubber of varying hardness are taken from Ref. 4.

Shore (IHRD) hardness	40	50	60	70
$\frac{\text{Dynamic stiffness}}{\text{Static stiffness}}$	1.5	1.8	2.1	2.4

Since the value of C_c varies with the deflection, manufacturers usually quote a single figure which corresponds to the slope of the tangent *ab* to the torque–deflection curve at the rated torque, typically one third of the maximum permitted torque.

3. If a cyclic torque $\pm\Delta T$, such as that corresponding to a torsional vibration, is superimposed on a steady torque T, Fig. 9.9, the deflection follows a path similar to that shown dotted. It is this feature, the hysteresis loop, which results in the dissipation of energy, by an amount ΔW proportional to the area of the loop that is responsible for the damping characteristics of the coupling.

Damping energy dissipated in this way appears as heat in the rubber and can, under adverse circumstances, lead to overheating and rapid destruction of the elements. The appearance of rubber dust inside coupling guards is a warning sign.

The damping capacity of a component such as a rubber coupling is described by the *damping energy ratio*:

$$\psi = \frac{\Delta W}{W}$$

This may be regarded as the ratio of the energy dissipated by hysteresis in a single cycle to the elastic energy corresponding to the wind-up of the coupling at mean deflection:

$$W = \frac{1}{2}T\theta = \frac{1}{2}T^2/C_c$$

The damping energy ratio is a property of the rubber. Some typical values are given in Table 9.2. The dynamic magnifier is a function of the damping energy ratio: as would be expected a high damping energy ratio corresponds to a low dynamic magnifier. Some authorities give the relation:

$$M = 2\pi/\psi$$

Table 9.2 *Damping energy ratio* ψ

Shore (IHRD) hardness	50/55	60/65	70/75	75/80
Natural rubber	0.45	0.52	0.70	0.90
Neoprene		0.79		
Styrene-butadiene (SBR)		0.90		

Table 9.3 *Dynamic magnifier M*

Shore (IHRD) hardness	50/55	60/65	70/75	75/80
Natural rubber	10.5	8.6	5.2	2.7
Neoprene		4.0		
Styrene-butadiene (SBR)		2.7		

However, it is pointed out in Ref. 2 that for damping energy ratios typical of rubber the exact relation:

$$\psi = \left(1 - e^{-2\pi/M}\right)$$

is preferable. This leads to values of M shown in Table 9.3, which correspond to the values of ψ given in Table 9.2.

It should be noted that when several components, e.g. two identical rubber couplings, are used in series the dynamic magnifier of the combination is given by:

$$\left(\frac{1}{M}\right)^2 = \left(\frac{1}{M_1}\right)^2 + \left(\frac{1}{M_2}\right)^2 + \left(\frac{1}{M_3}\right)^2 + \cdots \tag{13}$$

(this is an empirical rule, recommended in Ref. 5).

An example of drive shaft design

The application of these principles is best illustrated by a worked example. Figure 9.1 represents an engine coupled by way of twin multiple-bush type rubber couplings and an intermediate steel shaft to an eddy current dynamometer, with dynamometer starting.

Engine specification is as follows:

Four cylinder four-stroke gasoline engine	
Swept volume 2.0 litre, bore 86 mm, stroke 86 mm	
Maximum torque	110 Nm at 4000 rev/min
Maximum speed	6000 rev/min
Maximum power output	65 kW
Maximum bmep	10.5 bar
Moments of inertia	$I_e = 0.25\,\text{kgm}^2$
.	$I_d = 0.30\,\text{kgm}^2$

Table 9.4 indicates a service factor of 4.8, giving a design torque of $110 \times 4.8 = 530$ Nm.

Table 9.4 *Service factors for dynamometer/engine combinations*

Dynamometer type	*Number of cylinders*									
	Diesel					*Gasolene*				
	1/2	3/4/5	6	8	10+	1/2	3/4/5	6	8	10+
Hydraulic	4.5	4.0	3.7	3.3	3.0	3.7	3.3	3.0	2.7	2.4
Hyd. + dyno. start	6.0	5.0	4.3	3.7	3.0	5.2	4.3	3.6	3.1	2.4
Eddy current (EC)	5.0	4.5	4.0	3.5	3.0	4.2	3.8	3.3	2.9	2.4
EC + dyno. start	6.5	5.5	4.5	4.0	3.0	5.7	4.8	3.8	3.4	2.4
d.c. + dyno. start	8.0	6.5	5.0	4.0	3.0	7.2	5.8	4.3	3.4	2.4

It is proposed to connect the two couplings by a steel shaft of the following dimensions:

Diameter $D = 40\,\text{mm}$
Length $L = 500\,\text{mm}$
Modulus of rigidity $G = 80 \times 10^9\,\text{Pa}$

From eq. (9a), torsional stress $\tau = 42\,\text{Mpa}$, very conservative.
From eq. (10a)

$$C_s = \frac{\pi \times 0.04^4 \times 80 \times 10^9}{32 \times 0.5} = 40\,200\,\text{Nm/rad}$$

Consider first the case when rigid couplings are employed:

$$n_c = \frac{60}{2\pi}\sqrt{\frac{40\,200 \times 0.55}{0.25 \times 0.30}} = 5185\,\text{c.p.m.}$$

For a four cylinder, four-stroke engine, we have seen that the first major critical occurs at order $N_0 = 2$, corresponding to an engine speed of 2592 rev/min. This falls right in the middle of the engine speed range and is clearly unacceptable. This is a typical result to be expected if an attempt is made to dispense with flexible couplings.

The resonant speed needs to be reduced and it is a common practice to arrange for this to lie between either the cranking and idling speeds or between the idling and minimum full load speeds. In the present case these latter speeds are 500 and

1000 rev/min, respectively. This suggests a critical speed N_c of 750 rev/min and a corresponding resonant frequency $n_c = 1500$ cycles/min.

This calls for a reduction in the torsional stiffness in the ratio:

$$\left(\frac{1500}{5185}\right)^2$$

i.e. to 3364 Nm/rad.

The combined torsional stiffness of several elements in series is given by:

$$\frac{1}{C} = \frac{1}{C_1} + \frac{1}{C_2} + \frac{1}{C_3} + \cdots \tag{14}$$

This equation indicates that the desired stiffness could be achieved by the use of two flexible couplings each of stiffness 7480 Nm/rad. A manufacturer's catalogue shows a multi-bush coupling having the following characteristics:

Maximum torque	814 Nm (adequate)
Rated torque	170 Nm
Maximum continuous vibratory torque	±136 Nm
Shore (IHRD) hardness	50/55
Dynamic torsional stiffness	8400 Nm/rad

Substituting this value in eq. (14) indicates a combined stiffness of 3800 Nm/rad. Substituting in eq. (1) gives $n_c = 1573$, corresponding to an engine speed of 786 rev/min, which is acceptable.

It remains to check on the probable amplitude of any torsional oscillation at the critical speed. Under no-load conditions, the imep of the engine is likely to be in the region of 2 bar, indicating, from eq. (5a), a mean turning moment $M_{mean} = 8$ Nm.

From Table 9.1, p factor = 1.91, giving $T_{ex} = 15$ Nm per cylinder:

$$\sum T_{ex} = 4 \times 15 = 60 \text{ Nm}$$

Table 9.3 indicates a dynamic magnifier $M = 10.5$, the combined dynamic magnifier from eq. (13) = 7.4.

The corresponding value of the vibratory torque, from eq. (7), is then:

$$T_v = \frac{60 \times 7.4}{(1 + 0.25/0.30)} = \pm 242 \text{ Nm}$$

This is in fact outside the coupling continuous rating of ±136 Nm, but multiple bush couplings are tolerant of brief periods of quite severe overload and this solution should be acceptable provided the engine is run fairly quickly through the critical speed. An alternative would be to choose a coupling of similar stiffness using SBR bushes of 60/65 hardness. Table 9.3 shows that the dynamic magnifier is reduced from 10.5 to 2.7, with a corresponding reduction in T_v.

If in place of an eddy current dynamometer we were to employ a d.c. machine, the inertia I_b would be of the order of 1 kgm^2, four times greater.

This has two adverse effects:

1. Service factor, from Table 9.4 increased from 4.8 to 5.8;
2. The denominator in eq. (7) is reduced from $(1 + 0.25/0.30) = 1.83$ to $(1 + 0.25/1.0) = 1.25$, corresponding to an increase in the vibratory torque for a given exciting torque of nearly 50 per cent.

This is a general rule: the greater the inertia of the dynamometer the more severe the torsional stresses generated by a given exciting torque.

An application of eq. (1) shows that for the same critical frequency the combined stiffness must be increased from 3364 Nm/rad to 5400 Nm/rad. We can meet this requirement by changing the bushes from Shore Hardness 50/55 to Shore Hardness 60/65, increasing the dynamic torsional stiffness of each coupling from 8400 Nm/rad to 14000 Nm/rad (in general, the usual range of hardness numbers, from 50/55 to 75/80, corresponds to a stiffness range of about 3:1, a useful degree of flexibility for the designer).

Eq. (1) shows that with this revised coupling stiffness n_c changes from 1573 cycles/min to 1614 cycles/min, and this should be acceptable. The oscillatory torque generated at the critical speed is increased by the two factors mentioned above, but reduced to some extent by the lower dynamic magnifier for the harder rubber, $M = 8.6$ against $M = 10.5$. As before, prolonged running at the critical speed should be avoided.

For completeness, we should check the whirling speed from eq. (11). The mass of the shaft per unit length is: $W_s = 9.80$ kg/m.

$$N_w = \frac{30\pi}{0.50^2}\sqrt{\frac{200 \times 10^9 \times \pi \times 0.04^4}{64 \times 9.80}} = 19\,100 \text{ r.p.m.}$$

The mass of the shaft + half couplings is found to be 12 kg and the combined radial stiffness 33.6 MN/m. From eq. (12a):

$$N_t = \frac{30}{\pi}\sqrt{\frac{33.6 \times 10^6}{12}} = 16\,000 \text{ r.p.m.}$$

then from eq. (12b), whirling speed = 12 300 rev/min, which is satisfactory.

Note, however, that, if shaft length were increased from 500 to 750 mm, whirling speed would be reduced to about 7300 rev/min, which is barely acceptable. This is a common problem, usually dealt with by the use of tubular shafts, which have much greater transverse stiffness for a given mass.

There is no safe alternative, when confronted with an engine of which the characteristics differ significantly from any run previously on a given test bed, to following through this design procedure.

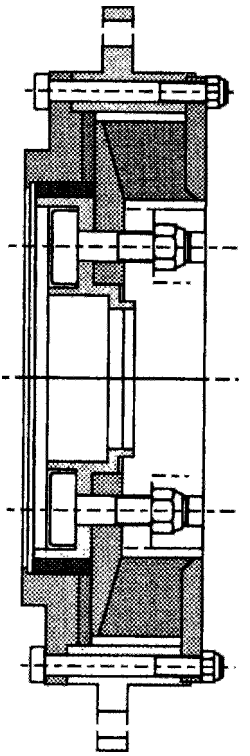


Figure 9.10 *Annular type rubber coupling*

An alternative solution

The above worked example makes use of two multiple-bush type rubber couplings with a solid intermediate shaft. An alternative is to make use of a conventional propeller shaft with two universal joints, as used in road vehicles, with the addition of a coupling incorporating an annular rubber element in shear to give the necessary torsional flexibility. These couplings, Fig. 9.10, are generally softer than the multiple bush type for a given torque capacity, but are less tolerant of operation near a critical speed. If it is decided to use a conventional universal joint shaft, the supplier should be informed of the maximum speed at which it is to run. This will probably be much higher than is usual in the vehicle and may call for tighter than usual limits on balance of the shaft.

Shock loading of couplings due to cranking, irregular running and torque reversal

Systems for starting and cranking engines are described in Chapter 8, where it is emphasized that during engine starting severe transient torques can arise. These have

been known to result in the failure of flexible couplings of apparently adequate torque capacity. The maximum torque that can be necessary to get a green engine over t.d.c. or that can be generated at first firing should be estimated and checked against maximum coupling capacity.

Irregular running or imbalance between the powers generated by individual cylinders can give rise to exciting torque harmonics of lower order than expected in a multicylinder engine and should be borne in mind as a possible source of rough running. Finally, there is the possibility of momentary torque reversal when the engine comes to rest on shutdown.

However, the most serious problems associated with the starting process arise when the engine first fires. Particularly when, as is common practice, the engine is motored to prime the fuel injection pump, the first firing impulses can give rise to severe shocks. Annular type rubber couplings, Fig. 9.10, can fail by shearing under these conditions. In some cases, it is necessary to fit a torque limiter or slipping clutch to deal with this problem.

Axial shock loading

Engine test systems that incorporate automatic shaft docking systems have to provide for the axial loads on both the engine and dynamometer imposed by such a system. In some high volume production facilities, an intermediate pedestal bearing isolates the dynamometer from both the axial loads of normal docking operation and for cases when the docking splines jam 'nose to nose'; in these cases the docking control system should be programmed to back off the engine, spin the dynamometer and retry the docking.

Selection of coupling torque capacity

Initial selection is based on the maximum rated torque with consideration given to the type of engine and number of cylinders, dynamometer characteristics and inertia. Table 9.4, reproduced by courtesy of Twiflex Ltd, shows recommended service factors for a range of engine and dynamometer combinations. The rated torque multiplied by the service factor must not exceed the permitted maximum torque of the coupling.

Other manufacturers may adopt different rating conventions, but Table 9.4 gives valuable guidance as to the degree of severity of operation for different situations. Thus, for example, a single cylinder diesel engine coupled to a d.c. machine with dynamometer start calls for a margin of capacity three times as great as an eight cylinder gasoline engine coupled to a hydraulic dynamometer.

Figure 9.11 shows the approximate range of torsional stiffness associated with three types of flexible coupling: the annular type as illustrated in Fig. 9.10, the multiple bush design of Fig. 9.8 and a development of the multiple bush design which permits a greater degree of misalignment and makes use of double-ended bushed links between the two halves of the coupling. The stiffness figures of Fig. 9.11 refer to a single coupling.

Exhibit 1003, Page 803
Petitioner MWE Investments, LLC, et al.

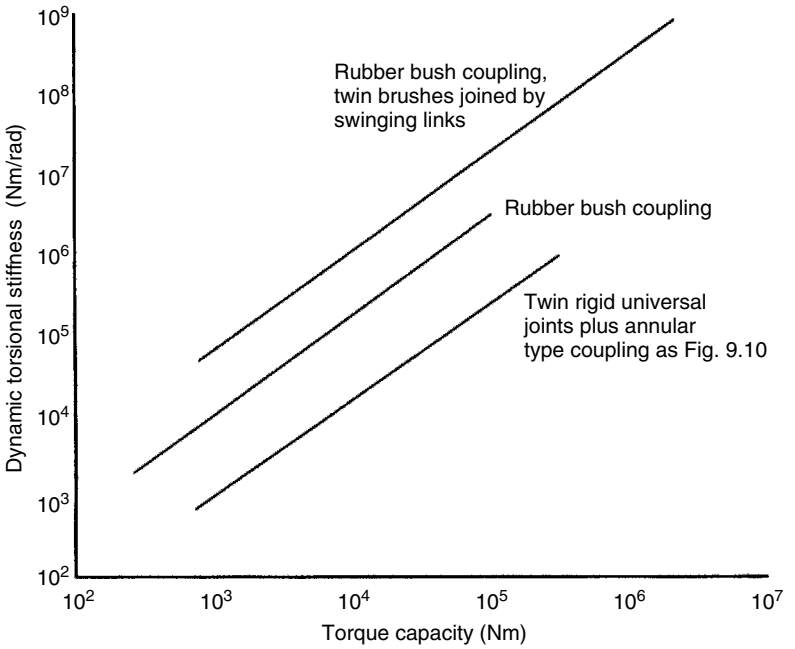


Figure 9.11 *Ranges of torsional stiffness for different types of rubber coupling*

The role of the engine clutch

Vehicle engines are invariably fitted with a clutch and this may or may not be retained on the test bed. The advantage of retaining the clutch is that it acts as a torque limiter under shock or torsional vibration conditions. The disadvantages are that it may creep, particularly when torsional vibration is present, leading to ambiguities in power measurement, while it is usually necessary, when the clutch is retained, to provide an outboard bearing. Clutch disc springs may have limited life under test bed conditions.

Balancing of drive line components

This is a matter which is often not taken sufficiently seriously and can lead to a range of troubles, including damage (which can be very puzzling) to bearings, unsatisfactory performance of such items as torque transducers, transmitted vibration to unexpected locations and serious drive line failures. Particular care should be taken in the choice of couplings for torque shaft dynamometers: couplings such as the multiple disc type, Fig. 9.7, cannot be relied upon to centre these devices sufficiently accurately.

It has sometimes been found necessary to carry out in situ balancing of a composite engine drive line where the sum of the out of balance forces in a particular radial relationship causes unacceptable vibration; specialist companies with mobile plant exist to provide this service.

Conventional universal joint type cardan shafts are often required to run at higher speeds in test bed applications than is usual in vehicles; when ordering, the maximum speed should be specified and, possibly, a more precise level of balancing than the standard specified.

BS 5265, Mechanical balancing of rotating bodies,[9] gives a valuable discussion of the subject and specifies 11 Balance Quality Grades. Drive line components should generally be balanced to Grade G 6.3, or, for high speeds, to grade G 2.5. The standard gives a logarithmic plot of the permissible displacement of the centre of gravity of the rotating assembly from the geometrical axis against speed. To give an idea of the magnitudes involved, G 6.3 corresponds to a displacement of 0.06 mm at 1000 rev/min, falling to 0.01 mm at 5000 rev/min.

Alignment of engine and dynamometer

This is a fairly complex and quite important matter. For a full treatment and description of alignment techniques, see Ref. 4. Differential thermal growth and the movement of the engine on its flexible mountings when on load should be taken into account and if possible the mean position should be determined. The laser-based alignment systems now available greatly reduce the effort and skill required to achieve satisfactory levels of accuracy. In particular, they are able to bridge large gaps between flanges without any compensation for droop and deflection of arms carrying dial indicators, a considerable problem with conventional alignment methods.

There are essentially three types of shaft having different alignment requirements to be considered:

1. Rubber bush and flexible disc couplings should be aligned as accurately as possible as any misalignment encourages heating of the elements and fatigue damage.
2. Gear type couplings require a minimum misalignment of about 0.04° to encourage the maintenance of an adequate lubricant film between the teeth.
3. Most manufacturers of universal joint propeller shafts recommend a small degree of misalignment to prevent brinelling of the universal joint needle rollers. Note that it is essential, in order to avoid induced torsional oscillations, that the two yokes of the intermediate shaft joints should lie in the same plane.

Distance between end flanges can be critical, as incorrect positioning can lead to the imposition of axial loads on bearings of engine or dynamometer.

Guarding of coupling shafts

Not only is the failure of a high speed coupling shaft potentially dangerous, as very large amounts of energy can be released, it is quite common. The ideal shaft-guard will contain the debris resulting from a failure of any part of the drive line and prevent accidental or casual contact with rotating parts, while being sufficiently light and adaptable not to interfere with engine rigging and alignment. A guard system that is very inconvenient to use will eventually be misused or neglected.

A really substantial guard, preferably a steel tube not less than 6 mm thick, split and hinged in the horizontal plane for shaft access, is an essential precaution. The hinged 'lid' should be interlocked with the emergency stop circuit to prevent engine cranking or running while it is open. Many designs include shaft restraint devices loosely fitting around the tubular portion, made of wood or a non-metallic composite and intended to prevent a failing shaft from whirling; these should not be so close to the shaft as to be touched by it during normal starting or running otherwise they will be the cause of failure rather than a prevention of damage.

Engine to dynamometer coupling: summary of design procedure

1. Establish speed range and torque characteristic of engine to be tested. Is it proposed to run the engine on load throughout this range?
2. Make a preliminary selection of a suitable drive shaft. Check that maximum permitted speed is adequate. Check drive shaft stresses and specify material. Look into possible stress raisers.
3. Check manufacturer's recommendations regarding load factor and other limitations.
4. Establish rotational inertias of engine and dynamometer and stiffness of proposed shaft and coupling assembly. Make a preliminary calculation of torsional critical speed from eq. (1). (In the case of large multicylinder engines consider making a complete analysis of torsional behaviour.)
5. Modify specification of shaft components as necessary to position torsional critical speeds suitably. If necessary, consider use of viscous torsional dampers.
6. Calculate vibratory torques at critical speeds and check against capacity of shaft components. If necessary specify 'no go' areas for speed and load.
7. Check whirling speeds.
8. Specify alignment requirements.
9. Design shaft guards.

Flywheels

No treatment of the engine/dynamometer drive line would be complete without mention of flywheels which may form a discrete part of the shaft system. A flywheel

is a device that stores kinetic energy. The energy stored may be precisely calculated and is instantly available. The storage capacity is directly related to the mass moment of inertia which is calculated by:

$$I = k \times M \times R^2$$

where:

I = moment of inertia (kgm^2)
k = inertia constant (dependent upon shape)
m = mass (kg)
R = radius of flywheel mass

In the case of a flywheel taking the form of a uniform disk, which is the common form found within dynamometer cells and chassis dynamometer designs:

$$I = \frac{1}{2}MR^2$$

The engine or vehicle test engineer would normally expect to deal with flywheels in two roles:

1. As part of the test object, as in the common case of an engine flywheel where it forms part of the engine/dynamometer shaft system and contributes to the system's inertial masses taken into account during a torsional analysis.
2. As part of the test equipment where one or more flywheels may be used to provide actual inertia that would, in 'real life', be that of the vehicle or some part of the engine driven system.

No mention of flywheels should be made without consideration of the safety of the application. The uncontrolled discharge of energy from any storage device is hazardous. The classic case of a flywheel failing by bursting is now exceptionally rare and invariably due to incompetent modification rather than the nineteenth century problems of poor materials, poor design or overspeeding.

In the case of engine flywheels, the potential danger in the test cell is the shaft system attached to it. This may be quite different in mass and fixing detail from its final application connection, and can cause overload leading to failure. Cases are on record where shock loading caused by connecting shafts touching the guard system due to excessive engine movement has created shock loads that have led to the cast engine flywheel fracturing, with severe consequential damage.

The most common hazard of test rig mounted flywheels is caused by bearing or clutch failure where consequential damage is exacerbated by the considerable energy available to fracture connected devices or because of the time that the flywheel and connected devices will rotate before the stored energy is dissipated and movement is stopped.

It is vital that flywheels are guarded in such a manner as to prevent absolutely accidental entrainment of clothing or cables, etc.

A common and easy to comprehend use of flywheels is as part of a vehicle brake testing rig. In these devices, flywheels supply the energy that has to be absorbed and dissipated by the brake system under test. The rig motor is only used to accelerate the chosen flywheel combinations up to the rotational speed required to simulate the vehicle axle speed at the chosen vehicle speed. Flywheel brake rigs have been made up to the size that can provide the same kinetic energy as fully loaded high speed trains. Flywheels are also used on rigs used to test automatic automotive gearboxes.

Test rig flywheel sets need to be rigidly and securely mounted and balanced to the highest practical standard. Multiples of flywheels forming a common system that can be engaged in different combinations and in any radial relationship require particular care in the design of both their base frame and individual bearing supports. Such systems can produce virtually infinite combinations of shaft balance and require each individual mass to be as well balanced and aligned on as rigid a base as possible.

Simulation of inertia* versus iron inertia

Modern a.c. dynamometer systems and control software have significantly replaced the use of flywheels in chassis and engine dynamometer systems in the automotive industry. Any perceived shortcoming in the speed of response or accuracy of the simulation is usually considered to be of less concern than the mechanical simplicity of the electric dynamometer system and the reduction in required cell space.

Finally, it should be remembered that, unless engine rig flywheels are able to be engaged through a clutch then the engine starting/cranking system will have to be capable of accelerating engine, dynamometer and flywheel mass up to engine start speed.

Notation

Frequency of torsional oscillation	n cycles/min
Critical frequency of torsional oscillation	n_c cycles/min
Stiffness of coupling shaft	C_s Nm/rad
Rotational inertia of engine	I_e kg m^2
Rotational inertia of dynamometer	I_b kg m^2
Amplitude of exciting torque	T_x Nm
Amplitude of torsional oscillation	θ rad
Static deflection of shaft	θ_0 rad
Dynamic magnifier	M
Dynamic magnifier at critical frequency	M_c
Order of harmonic component	N_o

* Some readers may object to the phrase ‘simulation of inertia’ since one is simulating the effects rather than the attribute, but the concept has wide industrial acceptance.

Number of cylinders	N_{cyl}
Mean turning moment	M_{mean} Nm
Indicated mean effective pressure	p_i bar
Cylinder bore	B mm
Stroke	S mm
Component of tangential effort	T_m Nm
Amplitude of vibratory torque	T_v Nm
Engine speed corresponding to n_c	N_c rev/min
Maximum shear stress in shaft	τ N/m^2
Whirling speed of shaft	N_W rev/min
Transverse critical frequency	N_t cycles/min
Dynamic torsional stiffness of coupling	C_c Nm/rad
Damping energy ratio	ψ
Modulus of elasticity of shaft material	E Pa
Modulus of rigidity of shaft material (for steel, $E = 200 \times 10^9$ Pa, $G = 80 \times 10^9$ Pa)	G Pa

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8. BS 6613 *Methods for Specifying Characteristics of Resilient Shaft Couplings*.
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Further reading

BS 4675 Parts 1 and 2 *Mechanical Vibration in Rotating Machinery*.
BS 6861 Part 1 *Method for Determination of Permissible Residual Unbalance*.
BS 6716 *Guide to Properties and Types of Rubber*.
Nestorides, E.J. (1958) *A Handbook of Torsional Vibration*, Cambridge University Press, Cambridge.

The layout of power cables of both multicore and single core is shown in Fig. 10.7 following recommended practice.

Conductive coupling interference

Conductive coupling interference may occur when there is a supply voltage difference between a number of control or measuring devices. It is usually caused by long supply line length or inappropriate distribution layout.

Countermeasures include:

- Keep supply cables short and of sufficient conductor size to minimize voltage drop.
- Avoid common return lines for different control or measurement devices by running separate supply lines for each device.

The same problem may occur when two devices are fed from different power supplies.

To minimize interference in analogue signal cables, an equipotential bonding cable or strap should connect the two devices running as close as possible to the signal cable. It is recommended that the bonding cable resistance be $<1/10$ (one tenth) of the cable screen resistance.

Integration of a.c. dynamometer systems

Careful consideration should be made when integrating an a.c drive system into an engine test facility. Where possible it is most advisable to provide a dedicated electrical supply to an a.c drive or number of a.c. drives.

There needs to be a clear understanding of the status of the existing electrical supply network, and the work involved in providing a new supply for an a.c dynamometer system. It is important to calculate the correct rating for a new supply transformer. This is a highly specialized subject and the details may change depending on the design of the supply system and a.c. devices, but the general rule is that the mains short circuit apparent power (S_{SC}) needs to be at least 20 times greater than the nominal apparent power of one dynamometer (S_N). Hence,

$$S_{SC}/S_N > 20.$$

In a sample calculation based on a single 220 kW dynamometer, the following sizing of the supply transformer may be found:

$P_N = 220\,\text{kW}$ nominal power, $\cos\varphi = 1$, overload 25%
$S_N = P_N \times \cos\varphi = 220\,\text{kVA}$
$S_{SC} = S_N \times 20 = 4400\ \text{kVA}$
Standard transformer (ST) $\sim U_{sc} = 6\%$

Integration of exhaust emission instrumentation

The great majority of engine test work is concerned with the taking of measurements that are in principle quite simple, even though great skill may be needed to interpret them: forces, pressures, masses, flow rates, speeds, displacements, temperatures, oscillations. Where emissions measurements are concerned, we are forced to move into a totally different and very sophisticated field of instrumentation engineering. The apparatus makes use of subtle and difficult techniques borrowed from the field of physics.

It is very desirable, for all but the simplest garage-type emissions apparatus, that a technician should be specially trained to take responsibility for the maintenance and calibration of the instruments involved.

There are significant subsystems that require to be housed in engine or vehicle test facilities, each of which require some special health and safety considerations due to the handling of hot and toxic gases within a building space. These will be considered in the following text.

The full flow of exhaust gases has to be safely ducted and samples for analysis have to be removed in the correct amounts at critical points in any vehicular after-treatment system.

It is important that water, which is a product of the combustion process, particularly that of the gasoline and some biofuels, is not allowed to condense within the sampling system. Some of the compounds produced by the partial combustion of fuels are soluble in water, so it is possible to lose a significant amount of these compounds in the condensation on the equipment surfaces.

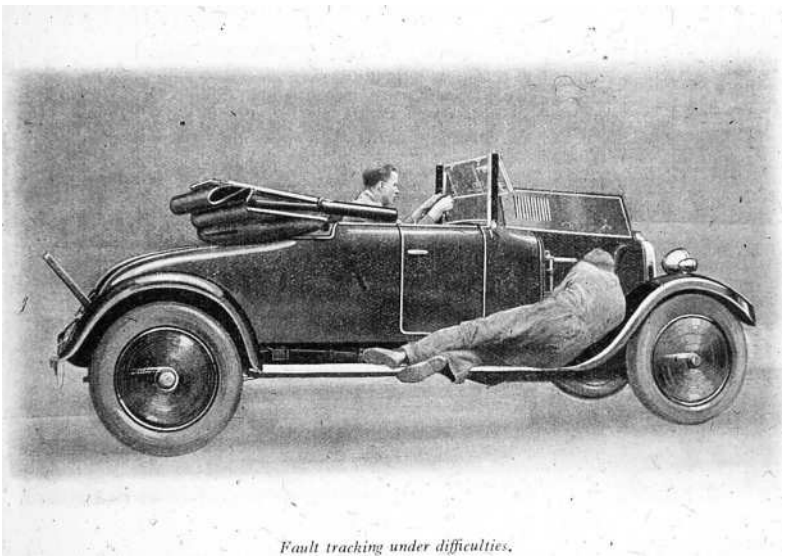
To prevent condensation in critical parts of the exhaust, the sampling system will be heated. For example, the sampling system for THC may consist of heated probe, sampling line filter and sampling pump, all of which are required to prevent the temperature of the equipment being lower than the dew point of the exhaust gas.

Calibration, span gases, storing gas distribution system

In order to function correctly exhaust gas analysers have to be calibrated regularly, using gases of known composition. The principle of calibration is similar to that of any transducer in that the zero point is set, in this case by purging with pure nitrogen (sometimes referred to as ‘zero-air’), then the 100 per cent value is set with a gas of that composition, known as a ‘span gas’. Good practice then dictates that a gas of an intermediate composition is then used as a check of system linearity. The whole routine may be highly automated by a calibration routine built into the analyser control unit. After calibration, the gas supply system is purged of at least NO_x to prevent degrading in the lines.

Many R&D analysers are capable of part-spanning to change the measuring sensitivity and can be calibrated by using either known span gas precisely diluted with a zero-air using a device known as a ‘gas divider’ or, more commonly, with

18 Chassis or rolling road dynamometers



Fault tracking under difficulties.

Introduction

This chapter deals with an area of testing activity that is closely allied to the testing of automotive engines and which increased enormously in importance during the last quarter of the twentieth century. It is exclusively concerned with road vehicles and their power-train systems and its expansion has been driven by a number of developments:

- *Technical*: the need to perfect the complex engine, transmission and whole vehicle management systems that are demanded by the modern road user, the economical and repeatable performance of endurance testing, investigations of noise, vibration

and harshness (NVH), emissions testing, checking performance under extreme climatic conditions, etc.
- *Legislative*: the need to comply with ever more exacting statutory requirements governing both new vehicles and the after-market in such areas as vehicle safety, brake performance, NVH, electromagnetic compatibility (EMC), fuel consumption and dominating all other tasks, exhaust emissions.

These requirements have called into existence a hierarchy of dynamometers and test chambers of increasing complexity:

- brake testers;
- in-service tuning, ECU remapping;
- end-of-line production testing;
- emissions certification/testing installations;
- mileage accumulation facilities associated with emission testing laboratories;
- anechoic and NVH and EMC test chambers;
- chassis dynamometers within climatic chambers and wind tunnels;
- independent wheel dynamometers.

The major types and uses are discussed later in this chapter.

The road load equation

The behaviour of a vehicle under road conditions is described by road load equations (RLE). The RLE is a fundamental requirement of a true chassis dynamometer which has to resist the torque being produced at the vehicle drive wheels in such a way as to simulate 'real-life' resistance to vehicle motion. It is necessary to have a clear understanding of this control model before studying the various types of machines coming under the heading of chassis dynamometers or rolling roads.

Increasing use of transient testing and road simulation in modern engine test cells means that the following section is relevant to engine test students and engineers.

The RLE is the formula that calculates the change in torque required with change of vehicle speed in order to simulate the real-life performance of a given vehicle; it defines the traction or braking force that is called for under all possible conditions, which include

- steady travel at constant speed on a level road;
- hill climbing and descent;
- acceleration, over-run and braking;
- transitions between any of the above;
- effects of atmospheric resistance, load, towed load, tyre pressure, etc.

The road load equation for a given vehicle defines the *tractive force* or *retarding force, F* (Newton), that must be applied in order to achieve a specified response to these conditions. It is a function of the following parameters:

Vehicle specific:		
Mass of vehicle	M	kg
Components of rolling resistance	a_0	N
Speed dependent	a_1V	N
Aerodynamic	a_2V^2	N
External:		
Speed	V	m/s
Road slope	θ	rad

The usual form of the road load equation is:

$$F = a_0 + a_1V + a_2V^2 + MdV/dt + Mg\sin\theta \tag{1}$$

where $M\ dv/dt$ = force to accelerate/brake vehicle and $Mg\ \sin\ \theta$ = hill climbing force.

More elaborate versions of the equation may take into account such factors as wheel spin and cornering.

The practical importance of the road load equation lies in its application to the simulation of vehicle performance. It forms the link between performance on the road and performance in the test department.

To give a feel for the magnitudes involved, the following equation relates to a typical four-door saloon of moderate performance, laden weight 1600 kg:

$$F = 150 + 3V + 0.43V^2 + 1600dV/dt + 1600 \times 9.81\sin\theta$$

The various components that make up the vehicle drag are plotted in Fig. 18.1a which shows the level road performance and makes clear the preponderant influence of wind resistance.

The road load equation predicts a power demand at the road surface of 14.5 kW at 60 mph, rising to 68.8 kW at 50 m/s (112 mph) for a level road.

These demands are dwarfed by the demands made by hill climbing and acceleration. Thus, Fig. 18.1b, to climb a 15 per cent slope at 60 mph, calls for a total power input of $14.5 + 63.2 = 77.7$ kW while, Fig. 18.1c, to accelerate from 0 to 60 mph in 10 s at constant acceleration, calls for a maximum power input of $14.5 + 115 = 129.5$ kW.

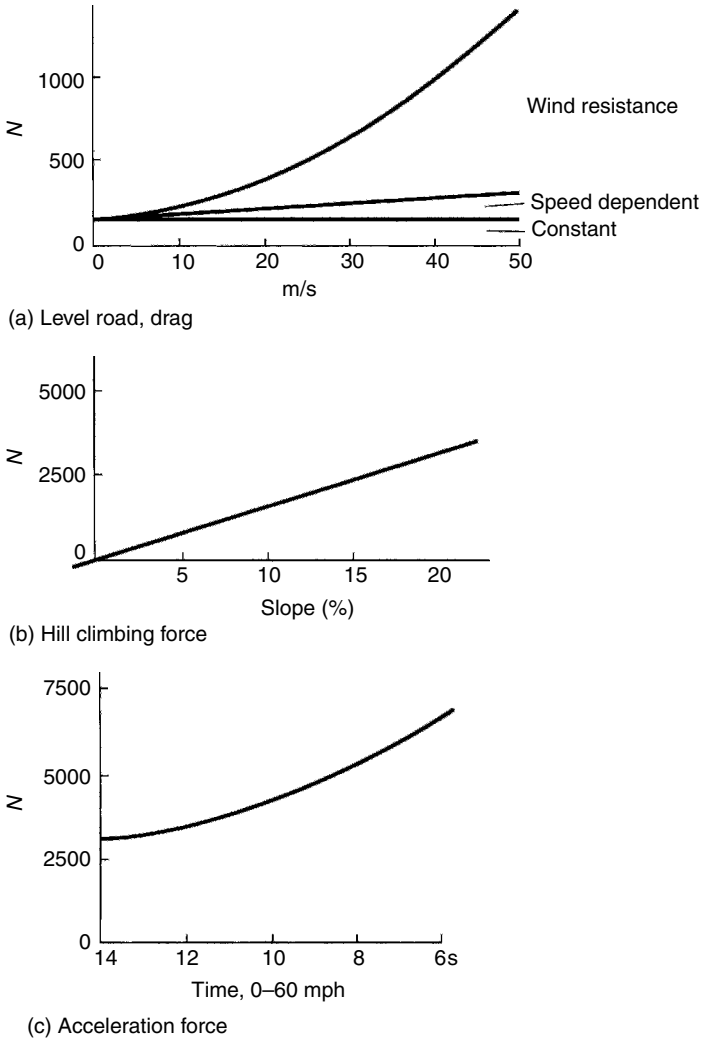


Figure 18.1 *Vehicle drag: saloon of laden weight 1600 kg (a) level road performance; (b) hill climbing; (c) acceleration force*

Genesis of the rolling road dynamometer

The idea of running a complete vehicle under power while it was at rest was first conceived by railway locomotive engineers before being adopted by the road vehicle industry. As a matter of historical record the last steam locomotives built in the UK

were tested on multiple roll units, with large eddy-current dynamometers* connected to each driven roll, the tractive force being measured by a mechanical linkage and spring balance. To an observer close to the locomotive on the test rig it must have presented an awe-inspiring sight, running at full power and a wheel speed of 90 mph.

Today the chassis dynamometer is used almost exclusively for road vehicles, although there are special machines designed for fork-lift and articulated off-road vehicles. The advantages to the designer and test engineer of having such facilities available are obvious; essentially, they allow the static observation and measurement of the performance of the complete vehicle while it is, in most respects, in motion.

Before the 1970s, most machines were comparatively primitive rolling roads, characterized by having rollers of rather small diameter, which inadequately simulated the tyre contact conditions and rolling resistance experienced by the vehicle on the road and fitted with various fairly crude arrangements for applying and measuring the torque resistance. A single fixed flywheel was commonly coupled to the roller to give a simulation of a vehicle inertia.

The main impetus for development came with the rapid evolution of emissions testing in the 1970s. The diameter of the rollers was increased, to give more realistic traction conditions, while trunnion mounted d.c. dynamometers with torque measurement by strain gauge load cells and more sophisticated control systems permitted more accurate simulation of road load. The machines were provided with a range of flywheels to give steps in the inertia; precise simulation of the vehicle mass was achieved by 'trimming' the effective inertia electrically.

In recent years the development in digital control techniques and electrical drives has meant that incremental flywheels have been dispensed with in most emission testing designs and been replaced by single (2×2) or double (4×4) roll machines reliant on electrical simulation of all aspects of 'road behaviour'.

Brake testers

These simple machines, commonly installed in service garages and government test stations, are used mainly for the statutory 'in-service' testing of cars and commercial vehicles. They are installed in a shallow pit, and consist essentially of two pairs of rollers, typically of 170-mm diameter and having a grit-coated surface to give a high coefficient of adhesion with the vehicle tyre. The rollers are driven by geared variable speed motors and two types of test are performed. Either the vehicle brakes are fully applied and torque increased until the wheels slip, or the rollers are driven at a low surface speed, typically 2 km/h, and the relation between brake pedal effort and braking force is measured.

The testing of a modern ABS brake system requires a separately controlled roller for each wheel and a much more complicated interaction with the vehicle braking

* The National Railway Museum in York holds records and displays the Heenan & Froude specification plate.

system. This calls for computerized control of the tester. In many vehicle management systems, a plug is provided for connection of the brake system to a suitable brake tester computer. Modern 'end of line' brake test machines are used by vehicle manufacturers in a series of final vehicle check stations and therefore have features, such as communication with the test vehicle control systems, that are model specific.

Rolling roads for in-service tuning and assessment

For the typical vehicle maintenance garage, the installation of a rolling road and its installation with a building represents a considerable investment, only justified if the garage has a good market for its expertise in vehicle tuning or other specialized skills.

To cater for this market a number of manufacturers produce complete installations based on small diameter rollers, thus requiring minimum subfloor excavation. This market has tended to be dominated by American suppliers and grew rapidly when the US Environmental Protection Agency (EPA) called for annual emissions testing of vehicles, based on a chassis dynamometer cycle (IM 240).

Road load simulation capability at this level is usually limited to that possible with a choice of one or two flywheels and a comparatively simple control system; most of the testing is concerned with optimizing (tuning) the engine management system and straightforward maximum power certification. The tests have to be of short duration to avoid overheating of engine and tyres.

In-service rolling roads for testing large trucks are confined almost exclusively to the USA. They are based on a single large roller capable of running both single- and double-axle tractor units. The power is usually absorbed by a portable water brake, such as that shown in Fig. 4.1, which may also be used for direct testing of engines, a useful facility in a large agency test facility.

Rolling roads for end-of-line production testing

Test rigs for this purpose range from simple roller sets to multi-axle units with interconnected roll sets and road load simulation. Certain special design features are required and any specification for such an installation should take them into account:

- The design and construction of the machine must minimize the possibility of damage from vehicle parts falling into the mechanism. It is quite common for fixings to shake down into the rolling road, where they can cause damage if there are narrow clearances or converging gaps between, for example, the rolls and lift-out beam, or in the wheel-base adjustment mechanism.
- The vehicle must be able to enter and leave the rig quickly yet be safely restrained during the test. The usual configuration is for all driven wheels to run between two rollers. Between each pair of rolls there is a lift-out beam that allows the vehicle to enter and leave the rig. When the beam descends it lowers the wheels between the

rolls and at the same time small 'anti-climb-out' rollers swing up fore and aft of each wheel. At the end of the test, with the wheels at rest, the beam rises, the restraining rollers swing down, and the vehicle may be driven from the rig.

- To restrain the vehicle from slewing from side to side to a dangerous extent, specially shaped side rollers are positioned between the rolls at the extreme width of the machine where they will make contact with the vehicle tyre to prevent further movement. These rollers can cause problems if a wide variety of tyre profiles are used on test vehicles particularly in the case of low profile tyres.
- Particular care should be taken with the operating procedures if 'rumble strips' are to be used. These are raised strips or keys sometimes fitted to the rolls in line with the tyre tracks, their purpose being to excite vibration to assist in the location of vehicle rattles. However, if the frequency of contact with the tyre resonates with the natural frequency of the vehicle suspension this can give rise to bouncing of the vehicle and even to its ejection from the rig. They can also give rise to severe shock loads in the roll drive mechanism.

Rigs designed to absorb power from more than one axle may have to be capable of adjustment of the centre distance between axles. The complexity of a shaft or belt connection between the front and rear roller sets has been obviated by modern electronic speed synchronization. Special purpose roller rigs for checking wheel alignment sometimes form part of the end-of-line test equipment.

Chassis dynamometers for emissions testing

The majority of chassis dynamometers manufactured worldwide in the last 25 years have been used for some form of emissions testing or homologation.

The standard emissions tests developed in the USA in the 1970s were based on a rolling road dynamometer developed by Clayton and having twin rolls of 8.625-inch (220 mm) diameter. This machine became a *de facto* standard despite its limitations, the most serious of which was the small roll diameter, which resulted in tyre contact conditions much different from those on the road.

Later models used rollers of 500 mm diameter coupled to a four-quadrant d.c. dynamometer with declutchable flywheels to simulate steps of vehicle inertia within which gaps the electrical power operated, Fig. 18.2.

The machine is designed for roller surface speeds of up to 160 km/h and the digital control system is capable of precise road load control, including gradient simulation. Such machines must be manufactured to a very high standard to keep vibration to an acceptable level. It is possible to abuse them, notably by violent brake application which can induce drive wheel bounce and thus very high stresses in the roller and flywheel drive mechanism.

The US Environmental Protection Agency (EPA), which remain the authority defining chassis dynamometers of this class worldwide, have approved machines having a single pair of rollers of 48-inch diameter and 100 per cent electronic inertia

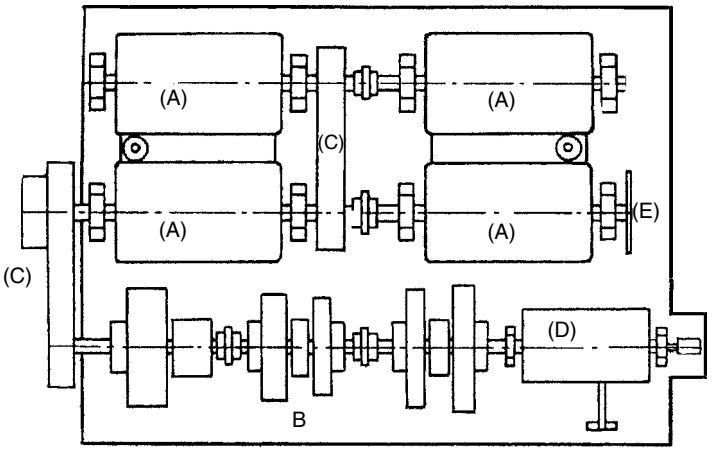


Figure 18.2 *Schematic plan view of four roller (2 × 2 = single driven axle) emission chassis dynamometer of late 1980s design. (A) Four 500-mm rollers; (B) selectable flywheel set; (C) belt drive; (D) d.c. motor fitted with shaft encoder; (E) disk brake for vehicle loading*

simulation. A number of standard designs are now available from the major suppliers and most are of a compact design having a double-ended a.c. drive motor with rollers mounted overhung on each stub shaft. This type of design can produce a standard, pretested package that enables chamber installation time to be minimized.

Mileage accumulation facilities

As a direct result of emissions regulations and the practice of homologation, it has become necessary for manufacturers to check on the change and probable deterioration in emissions performance after the vehicle has been driven for up to 80 000 km. One current legislative trend is to increase the length of this certified 'useful life' of the vehicles emission control system from 50 000 to 100 000 miles or greater.

The cost and the physical strain of using human drivers on test tracks or public roads for driving vehicles the prescribed distances in as little time as possible are high so special chassis dynamometer systems have been developed for running the specified sequences under automatic control, commonly for a period extending to 12 weeks.

Mileage accumulation dynamometers comprise a single large roll, of 48 inches or larger diameter, directly coupled to a d.c. or a.c. motor having sufficient power to run the required repetitive test sequence; for the average saloon car a motor capacity of 150 kW is sufficient.

In order to fully automate the process, the test vehicle is usually fitted with a robot driver. A bewildering range of these devices is available on the world market: for

this particular application reliability must be the prime consideration. Some devices can be mounted on the driver's seat and some are capable of also operating the footbrake and engaging reverse gear, but these refinements are not required by current legislative drive cycles. The set-up time of the robot may be appreciable but should be assessed against a test duration of 12 weeks or more. If wear or deterioration takes place in the robot mechanism or in the vehicle controls, for example by change in the clutch 'bite' position, the control system should be capable of automatic rectification, usually by means of periodic 'relearn' and adjustment cycles.

Due to the obvious problems of ventilating such a facility it is invariably located outside, under a simple roof. The control room housing the computerized control and safety systems can occupy a small building at one end of the facility, which in some cases may comprise up to 10 chassis dynamometers.

As the vehicle's own cooling system is stationary, a motorized cooling fan, facing the front grill, is essential. The fan will be fitted with a duct to give a reasonable simulation of air flow under road conditions, and must be firmly anchored. The fan speed is usually controlled to match apparent vehicle speed up to about 130 km/h; above that speed noise and fan power requirements become an increasing problem. Since mileage accumulation facilities usually run 24 h a day, they must be suitably shielded to prevent noise nuisance.

Automatic refuelling systems may be fitted in the facility for each chassis dynamometer unit.

Special purpose chassis dynamometers

Noise, vibration and harshness (NVH), electromagnetic compatibility (EMC) rigs and anechoic chamber dynamometers

Some features of anechoic cell design are discussed in Chapter 3. A critical requirement is that the chassis dynamometer should itself create the minimum possible noise. The usual specification calls for the noise level to be measured by a microphone located 1 m above and 1 m from the centreline of the rolls. Typically, the required sound level when the rolls are rotating at a surface speed of 100 km/h will be in the region of 50 dBA.

To reduce the contribution from the dynamometer motor and its drive system, it is usually located outside the main chamber in its own sound-proofed compartment and connected to the rolls by way of a long shaft (the design of these shafts can present problems; lightweight tubular carbon fibre shafts are sometimes used). The dynamometer motor will inevitably require forced ventilation and the ducting will require suitable location and treatment to avoid any contribution of noise.

Hydrodynamic bearings on quiet roll systems are common, but not without their own problems as their theoretical advantages in terms of reduced shaft noise may not be realized in practice, unless noise from the pressurized supply oil system is sufficiently attenuated.

For a well-designed chassis dynamometer, the major source of noise will be the windage generated by the moving roll surface and this is not easily dealt with. Smooth surfaces and careful shrouding can reduce the noise generated by the roller end faces, but there is inevitably an inherently noisy jet of air generated where the roller surface emerges into the test chamber. If the roller has a roughened surface to simulate road conditions the problem is exacerbated. The noise spectrum generated by the emerging roller is influenced by the width of the gap and in some cases adjustment is provided at this point.

The flooring over the dynamometer pit, usually of steel plate, must be carefully designed and appropriately damped to avoid resonant vibrations. A particular feature of NVH test cells is that the operators may require access to the underside of the vehicle during operation. This is usually by way of a trench at least 1.8 m deep, lying between the vehicle wheels.

Testing for electromagnetic compatibility (EMC)

Worldwide safety standards include regulations governing the acceptable levels of electromagnetic interference produced by road vehicles and the vulnerability of vehicle electronic systems to powerful directional beams of electromagnetic energy, such as may be produced by civil and military aviation systems.

To deal with this problem, a few units have been designed with the chassis dynamometer mounted on a large turntable capable of being rotated through 360°.

Climatic test cells for vehicles

There is a requirement for this kind of facility for development work associated with various aspects of performance under extreme climatic conditions. Subjects include driveability, cold starting, fuel waxing and solar loading, vapour lock, air conditioning and the ubiquitous topic of emissions.

The design of air conditioning plants for combustion air has been discussed in Chapter 5. The design of an air conditioned chamber for the testing of complete vehicles is a very much larger and more specialized problem. Figure 18.3 is a schematic drawing of such a chamber, built in the form of a recirculating wind tunnel so that the vehicle on test, mounted on a chassis dynamometer, can be subjected to oncoming air flows at realistic velocities, and in this particular case over a temperature range from +40 to −30°C.

The outstanding feature of such an installation is its very large thermal mass and, since it is usually necessary, from the nature of the tests to be performed, to vary the temperature of the vehicle and the air circulating in the tunnel over a wide range

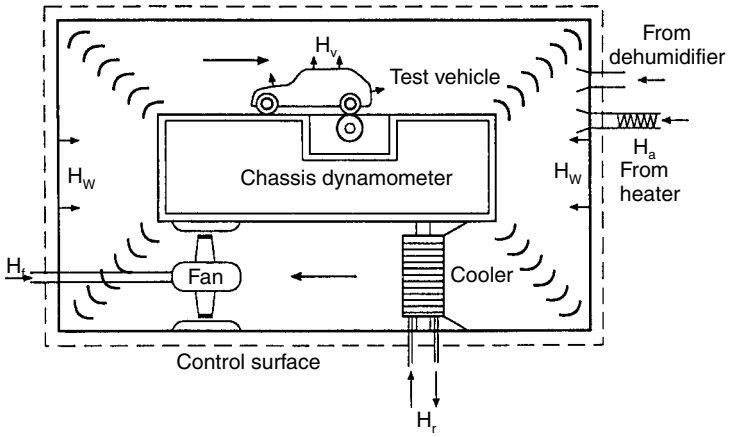


Figure 18.3 *Schematic of climatic wind tunnel facility fitted with a chassis dynamometer*

during a prolonged test period. This thermal mass is of prime significance in sizing the associated heating and cooling systems.

The thermodynamic system of the chamber is indicated in Fig. 18.3 and the energy inflows and outflows during a particular cold test, in accordance with the methods described in Chapter 2, were as follows:

The thermal capacity (thermal mass) of the various elements of the system is defined in terms of the energy input required to raise the temperature of the element by 1°C.

In	.	*Out*	.
H_a make-up air	20 kW	H_r refrigerator circuit	495 kW
H_v vehicle convection/radiation	100 kW	.	
H_r tunnel fan power	154 kW	.	
H_w walls, etc. by difference	221 kW	.	
Totals	495 kW	.	495 kW

C_a air content of chamber, return duct, etc. volume approximately 550 m^3, approximate density 1.2 kg/m^3, $C_p = 1.01$ kJ/kg.K, thermal mass = $550 \times 1.2 \times 1.01 = 670$ kJ/deg C

C_v vehicle, assume a commercial vehicle, weight 3 tonnes, specific heat of steel approximately 0.45 kJ/kg.K, thermal mass, say $3000 \times 0.45 = 1350$ kJ/deg C

C_e fan, cooler matrix, internal framing etc. estimated at 2480 kJ/deg C

C_s structure of chamber. This was determined by running a test with no vehicle in the chamber and a low fan speed. The rate of heat extraction by the refrigerant circuit was measured and a cooling curve plotted. Concrete is a poor conductor of heat and the coefficient of heat transfer from surfaces to air is low. Hence during the test a temperature difference between surfaces and wall built up but eventually stabilized; at this point the rate of cooling gave a true indication of effective wall heat capacity. The test showed that the equivalent thermal mass of the chamber was about 26 000 kJ/deg C, much larger than the other elements. Concrete has a specific heat of about 8000 kJ/m^3 deg C, indicating that the 'effective' volume involved was about $26\,000/8000 = 3.2\,m^3$.

The total surface area of the chamber is about 300 m^2, suggesting that a surface layer of concrete about 1 cm thick effectively followed the air temperature.

Adding these elements together we arrive at a total thermal mass of 30 500 kJ/deg C. Then since 1 kJ = 1 kW-sec, we can derive the rate at which the temperature of the air in the chamber may be expected to fall for the present case.

This shows that the 'surplus' cooling capacity available for cooling the chamber and its contents, H_w, amounts to 221 kW. This could be expected to achieve a rate of cooling of $221/30\,500 = 0.0072$ deg C/s, or 1°C in 2.3 min. However, this is the final rate when the temperature difference between walls and air had built up to the steady state value of about 20°C. The observed initial cooling rate is much faster, about 1.5°C/min. Heating presents less of a problem, since the heat released by the test vehicle and the fan assist the process rather than opposing it.

A further effect is associated with the moisture content of the tunnel air. On a warm summer's day this could amount to about 10 kg of water and during the cooling process this moisture would be deposited, mostly on the cooler fins, where it would eventually freeze, blocking the passages and reducing heat transfer. To deal with this problem, it is necessary to include a dehumidifier to supply dried air to the tunnel circuit.

Special features of chassis dynamometers in climatic cells

Chassis dynamometers intended to operate over the temperature range +40 to −10°C call for no special features, apart from sensible precautions to deal with condensation; for temperatures below this range and particularly those working below −25°C certain special features will be required.

Until the compact 'motor between rollers' ('motor-in-middle' in USA) design of dynamometer became common, the dynamometer motor in climatic cells was often isolated from the cold chamber and operated at normal temperatures. With the new designs two quite different strategies can be adopted to prevent low temperatures causing temperature related variability in the dynamometer system.

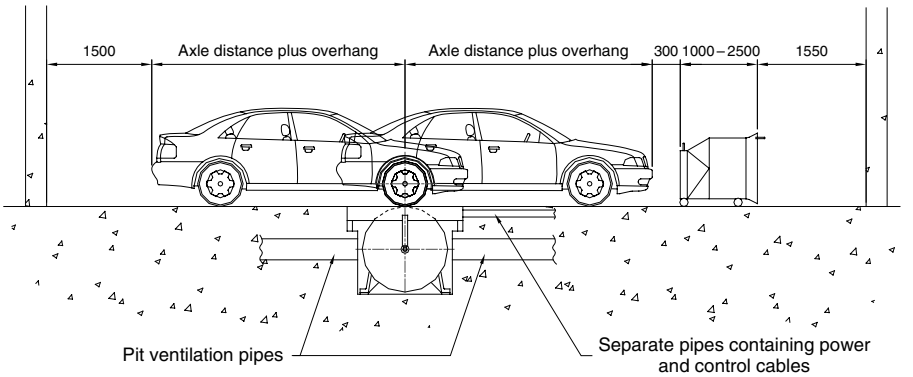


Figure 18.4 *A single roll set (2 × 2) chassis dynamometer plus road speed following cooling fan, within a cell sized to test rear and front wheel drive cars (AVL Zöllner)*

- The dynamometer pit can be kept at a constant temperature above or below that of the chamber by passing treated (factory) air through the pit via ducts cast in the floor (see Fig. 18.4). This strategy submits portions of the roll's surface to differential temperatures during the static cooldown phase but without reported problems.
- The pit can be allowed to chill down to the cell temperature but the parts of the chassis dynamometer that are crucial to accurate and consistent performance, such as bearings and load cell, are trace heated.

Whatever the layout components exposed to temperatures below 25°C such as rolls and shaft should be constructed of steel having adequate low-temperature strength to avoid the risk of brittle fracture.

Independent wheel dynamometers

A limitation of the conventional rolling road dynamometer is that it is unable to simulate cornering, during which the wheels rotate at differing speeds, wheel spin and skidding. With the increased adoption of electronically controlled traction and braking (ABS), there is a growing requirement for test beds that can simulate these conditions. There are two types of solution:

- *Four roll-set rolling roads*. These machines range from end-of-line rigs consisting of four sets of independent double-roller units, to complex development test rigs, some having steered articulation of each roll unit. The production rigs permit the checking of onboard vehicle control systems, transducers and system wiring by simulation of differential resistance and speeds of rotation. The development of

hybrid vehicles may give impetus to the development of independent four-wheel chassis dynamometers since some designs are based on individual wheel drive motors. Electronically controlled transmission systems on 4×4 vehicles also require this type of dynamometer to simulate the road conditions for which they are designed to compensate.

- *Wheel substitution dynamometers*. A major problem faced by engineers carrying out NVH testing on a chassis dynamometer is that tyre noise can dominate sound measurements. One answer is to absorb the power of each drive wheel with an individual dynamometer. If the wheel hub is modified the vehicle can still sit on its tyres, giving approximately the correct damping effect, while the tyre contact and windage noise is eliminated. Alternatively, the wheels may be removed and the individual wheel dynamometers used to support the vehicle. The dynamometers should be four-quadrant machines, to simulate both driving and coasting/braking conditions. Hydrostatic, a.c. and d.c. machines have been used for this application.

The installation of chassis dynamometers

Whatever the type of unit the reader has to specify and install, there are some common planning and logistical problems that need to be taken into account. For single roll machines built within a custom-built chamber, the longitudinal space requirement may have to take account of front- and rear-wheel drive vehicles plus room for tie-down mechanisms and the vehicle cooling fan. A typical layout is shown in Fig. 18.4.

As with engine test cells dealing with the testing of exhaust emissions, chassis dynamometers associated with ultralow emissions work (SULEV) should be built on sites that are out of the drift zone of automotive and industrial pollutants.

Chassis dynamometer pit design and construction details

The minimum requirements, if not the exact specification, of the pit into which a chassis dynamometer is to be installed will invariably be provided by the dynamometer's manufacturer which will include the loading on the pit floor to aid the structural designer. The pit must be built to a standard of accuracy rather higher than is usual in some civil engineering practice since all pit installed dynamometers have a close dimensional relationship with the building in which they are installed; this is particularly true for the pit construction which has to meet three critical dimensional standards:

1. The depth of the pit in relation to the finished floor level of the test cell needs to be held to tight dimensional and level limits. Too deep is recoverable, too shallow can be disastrous.

2. The lip of the pit has to be finished with edging steel that interfaces accurately with the flooring plates that span the gap between the exposed rolls' surface and the building floor.
3. The centre line of the dynamometer has to be positioned and aligned with the building datum and the vehicle hold-down structure that may also be cast into the cell floor. Fine alignment can be achieved by the dynamometer installers, but movement at that stage will be very restricted.

Pit depth

To achieve point 1 above, it is normal practice to make the pit floor lower than the datum dimension and then cast into the floor levelled steel 'sole plates' at the required height minus 5–10 mm to allow some upward adjustment of the machine to the exact floor level. The final levelling can be done by some form of millwright's levelling pads, or more usually levelling screws and shim plates, as shown in Fig. 18.5.

The system described above allows the pit floor to have some slight 'fall' towards a small drainage sump fitted with a level alarm or float operated pump. If the site has a high water table, the pit will have been suitably 'tanked' before final concrete casting to prevent ground water seepage; however, spillage of vehicle liquids and cell washing will drain into the pit so a means of taking these liquids into a foul liquid intercept drain is advisable.

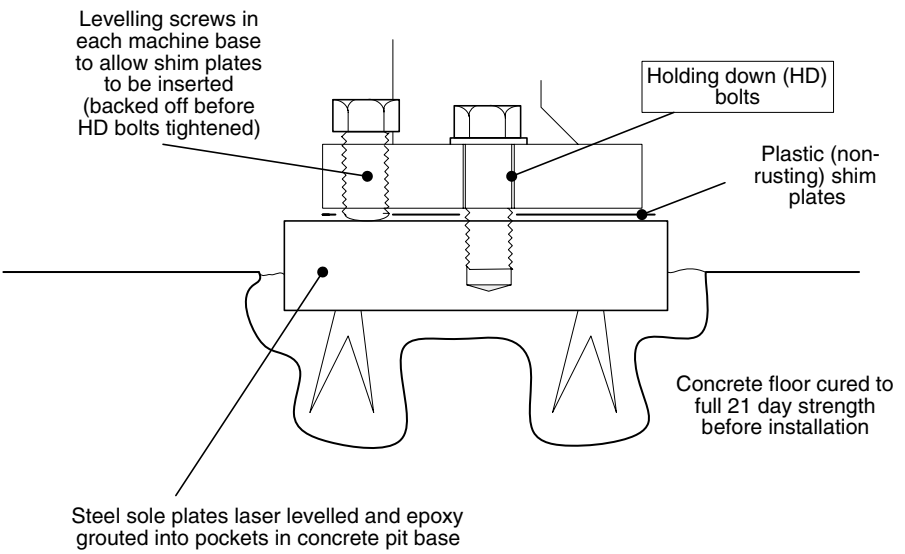


Figure 18.5 *One of the levelled steel plates exactly levelled and cast in the pit floor to support the chassis dynamometer (one per dynamometer HD bolt location)*

The need to run subfloor cable and ventilation ducts, between the control room and the pit and the drive cabinets and the pit requires preplanning and collaboration between the project engineer and builder, it can also complicate the task of creating a watertight seal against ground water. Normally there will be the following subfloor ducts:

- signal and separate small power cable ducts between control room and dynamometer pit;
- high power drive cables and separate signal cable ducts between the drive cabinet room/space and the dynamometer pit;
- pit ventilation ducts which at a minimum will have to include a purge duct for removal of hydrocarbon vapours (see ATEX requirements, Chapter 4).

On the wall of the test cell it is usual to fit a 'break-out' box that allows transducers, microphones or CAN bus plugs to be used for special vehicle-related communications between control room and test unit.

Project managers of any major test facility building are advised to check the layout and take photographic records of all substructure service pipes and steel work before concrete is poured.**

When, due to diminishing access, the dynamometer has had to be located in its pit before the building has been completed and before permanent electrical services are available, it may be vulnerable until building completion to the pit flooding and roll surface damage; provision must be made to guard against both eventualities.

Pit flooring

In most large chassis dynamometer projects, the false floor and its support structure are provided by the dynamometer manufacturer. It may be centrally supported by the dynamometer structure and will have access hatches for maintenance and calibration equipment. The steel or aluminium floor plates will be lifted from time to time and any hatches will need to be interlocked with the control system. It will be clear to readers that unless the pit shape is built to a quite precise shape and size, there will be considerable difficulty in cutting floor plates to suit the interface and that errors in this alignment will be highly visible. It is strongly recommended that the civil contractor should be supplied with the pit edging material by the chassis dynamometer manufacturer and that it is cast into the pit walls under his supervision before final edge grouting. In some cases the pit edging assembly can be fixed with jig beams in temporary place to hold the rectangular shape true while fixing (Fig. 18.6).

** AJM was once told by a steel fixer that his trade was comparable with that of a surgeon in that they both bury their mistakes; henceforth detailed records have been made of prepour structures.

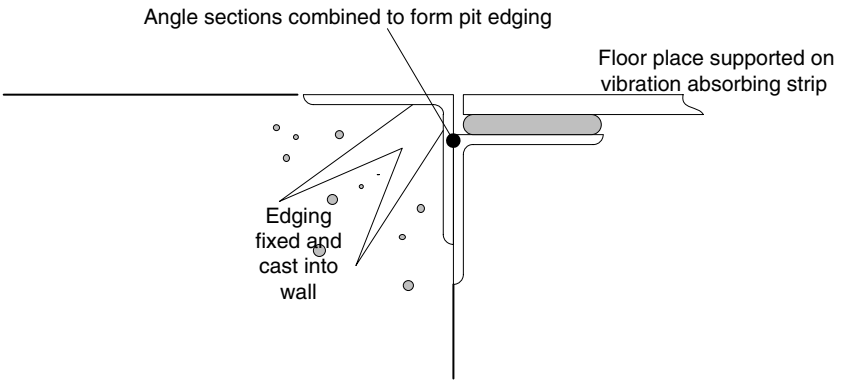


Figure 18.6 *Section through one form of pit edging required to support pit flooring*

Design and installation of variable geometry (4 × 4) dynamometers

Many chassis dynamometers intended for testing four, or more, wheel drive vehicles are designed so that the interaxle distance can be varied so as to accommodate a range of vehicles in a product family. The standard method adopted in 4 × 4 designs is for one, single-axle dynamometer module to be fixed within the building pit and the identical second module to be moved towards or away from the datum module on rails inside the pit. The range of interaxle distance for light duty vehicles (LDV) will be in the range of 2 to 3.5 m, but for commercial vehicle designs the range of movement will be far greater. This range of movement is important in that it tends to determine the design needed to accommodate that section of cell flooring that is required to move with the traversing set of rolls. In the LDV designs where the total module movement is around 1.5 m the moving plates can be based on a telescopic design with moving plates running over or under those fixed. Where the movement considerably exceeds 1.5 m, then the moving floor designs may have to be based on a slatted sectional floor that runs down into the pit at either end, while being tensioned by counterweights or as part of a tensioned cable loop. All these designs can be problematic in operation and require good housekeeping and maintenance standards.

The traversing module is normally moved by two or more electrically powered lead screw mechanisms coupled to a linear position transducer so that a set position can be selected from the control room. For operational reasons, it is recommended that some positional indication, vehicle-specific or actual interaxle distance, is marked on the operating floor.

LDV units are usually traversed on flat machined surface rails having lubricated foot pads fitted with sweeper strips to prevent dirt entrainment; this is very similar to long established machine tool practice and has to be installed with a similar degree of accuracy. Large commercial vehicle rigs often use crane traversing technology rather than that of machine tools with crane rails installed on cast ledges within the pit wall design and the axle modules running within 'crane beams' running on wheels some of which are electrically powered.

Drive cabinet housing

The same precautions relating to the integration of high powered IBGT or thyristor controlled d.c. drives covered in Chapter 10 relate to those fitted to chassis dynamometers.

The drive and control cabinet for high power chassis dynamometers may be over 6 m long, 2.2 m high and, although only 600–800 mm deep, will require space for front and sometimes rear access. These units are heavy with high centres of gravity and vulnerable to damage through damp or dirty atmosphere; if positioned before the building is clean and heated then adequate provision for protection should be made.

Offloading and positioning

Even small machines that are delivered in one unit require lifting equipment to offload, transport and position them in the pit. Forty-eight-inch roll machines or larger units may have to be positioned in already built or partly completed chambers; therefore, the installation will often require specialist equipment and contractors to lift and manoeuvre the machine in a building area of limited headroom and limited floor access as in the case of lined climatic chambers. It is a project phase that is used as an example in the 'Project timing chart' section of Chapter 1, which is recommended reading.

Tyre burst detectors

On all mileage accumulation rigs and others undertaking prolonged automatic test sequences, there should be some form of detector that can safely shut down the whole system in the event of a tyre deflating. The most common form is a limit switch mounted on a floor stand with a long probe running under the car at its midpoint with limit switches fitted one on each side. Any deflation will cause the vehicle body to drop, shutting down the test in a safe manner.

Cable layout

The general treatment of this subject in Chapter 10 applies in the present case, but special attention should be given to the avoidance of electrical interference from power cables with instrumentation lines. In many cases, all the cables running to the chassis dynamometer will at some point run through plastic tubes set within the concrete of the pit walls and plant room floors. These cable tubes should be allocated to either power or signal and should have a drainage slight fall to the pit. After installation is complete, the tubes can be 'stopped' at both ends with acoustic and fire attenuating material.

Loading and emergency brakes

It must be possible to lock the rolls to permit loading and unloading of the vehicle. These consist of either disc brakes fitted to the roll shafts or brake pads applied to the inside surface of one or both rollers. In normal operation these brakes have to be of sufficient power to resist the torques associated with driving the vehicle on and off the rig. Pneumatic or hydraulically powered, they are controlled either by the operational controls (vehicle loading) or by the safety instrumentation (EM stop) and are designed to be normally 'on' and require active switching to be off (machine operational).

It is not considered good practice to rely solely on these brakes to bring the rig to rest in the case of an emergency. They should only be used to supplement the braking effort of the main drive system. In addition there must be a well-thought out 'lock-out' system to ensure that all personnel are outside restricted areas before start-up, particularly important in large facilities where the operator may not be able to see all areas.

Vehicle restraints

The test vehicle must be adequately restrained against fore and aft motion under the tractive forces generated, and against slewing or sideways movements caused by the wheels being set at an angle to the rig axis. It should be pointed out to inexperienced readers that while the vehicle has no momentum driving on a rolling road, the chassis dynamometer does have momentum and it operates in the opposite direction of travel to that which the vehicle would have at equivalent velocity. Sudden braking of the vehicle therefore will cause it to try to jump off the rolls backwards rather than forwards. Vehicle restraint is less of a problem when the powered wheels are

located between a pair of rolls rather than resting on a single roller; however, paired rolls are only used for short duration test work because of their limited diameter and hence poor contact pattern. They are generally used for low power end-of-line testing, where they are usually relied on as the only means of restraint so as to allow for rapid loading and unloading of the test vehicles. Where full restraint systems are used, the cell floor must be provided with strong anchorage points. There are many different designs of vehicle restraint equipment, ranging from the use of large pneumatic bags to simple chain or loading strap attachments. Three different types of restraint system may be distinguished:

- A vehicle with rear wheel drive is the easiest type to restrain. The front wheels can be prevented from moving by fore and aft chocks linked by a tie bar. The rear end may be prevented from slewing by two high strength luggage straps with integral tensioning devices; these should be arranged in a cross-over configuration with the floor fixing points outboard and to the rear of the vehicle.
- Front wheel drive vehicles need careful restraint, since any movement of the steering mechanism with the rolls running at speed can lead to violent yawing. Restraints may be similar to those described for rear wheel drives, but with the straps at the front and chocks at the rear. The driver in manned runs should be familiar with the handling characteristics of the vehicle in normal operation, while for unmanned operation the steering wheel should be locked; otherwise, disturbances such as a burst tyre could have serious results. Tyre pressures, usually set above the normal level, should be carefully equalized.
- Four-wheel drive vehicles usual rely on cross-over strapping at both ends, but details depend on fixing points built into the vehicle. Where strong tow points are conveniently supplied they can be used to connect via a short articulated connector to a floor fixed restraint pillar. When using straps, it is good practice to tie the rear end first and then to drive the vehicle slowly ahead with the steering wheel loosely held. In this way, the vehicle should find its natural position on the rolls and can be restrained in this position, giving the minimum of tyre scrubbing and heating.

Guarding and safety

Primary operational safety of chassis dynamometer operators is achieved by restricting human access to machinery while it is rotating. Access to the pit should be hardwire interlocked with the EM stop system and latched EM stop buttons positioned in plant areas remote from the control room, such as the drive room and cell extremities.

The small exposed segment of dynamometer rolls on which the tyres rotate is the most obvious hazard to operators and drivers when the rig is in motion and personnel guards must always be fitted. Such guards are a standard part of the dynamometer system for good safety reasons, but perhaps also because it is the only operational

piece of the manufacture's equipment that is visible to the visitor after installation and usually will bear the company logo. Cover plates for the rolls for use when the vehicle is present are also advisable, particularly to prevent surface damage during maintenance periods.

Roll surface treatment

In the past, most twin roll and many single roll machines had no special treatment applied to the roll surfaces, which had a normal finish machined surface. Brake testers and some production rigs have a high-friction surface, which can give rise to severe tyre damage if the machine is used incorrectly. The roll surface of modern single roll machines may be sprayed with a fine-grained tungsten carbide coating, but other common surfaces include emery grit or plain steel.

'Road shells'

For development work, and particularly for NVH development, it is sometimes desirable to have a simulated road surface attached to the rolls. Usually the required surface is a simulation of quite coarse stoned asphalt; but a roughened roll surface greatly increases the windage noise of the rig itself.

Road shell design and manufacture is difficult. The shells should ideally be made up of four or more segments of differing lengths and with junction gaps that are as small as possible and cut helically. It is rarely required to induce low frequency vibrations in the test vehicle; if it is required, it should be generated by randomly distributed variations in roll surface height to avoid possible resonances in the vehicle suspension.

The most usual techniques for producing road surfaces for attachment to chassis dynamometer rolls are as follows:

- Detachable fibreglass road shells having an accurate moulding of a true road surface. These shells are usually thicker than the aluminium type, below, and in both cases the floor plates must be adjustable in order to accommodate them.
- Detachable cast aluminium alloy road shells, made in segments that may be bolted to the outside diameter of the rolls. The surface usually consists of parallel sided pits that give an approximation to a road surface.
- A permanently fixed road surface made up of actual stones bonded into a rubber belt which is bonded to the roll surface.

Most road shells are not capable of running at maximum rig speed because they are difficult to fix and to balance, also they may not be capable of transmitting full acceleration torque. It is therefore necessary to provide safety interlocks so that the

computerized speed and torque alarms are set to appropriate lower levels when shells are in use. Road shells also change the effective rolling radius and the base inertia of the dynamometer, requiring appropriate changes in the software constants.

Driver aids

The control room needs to be in touch with the driver in a test vehicle on a chassis dynamometer, either by voice transmission or visually by way of an aid screen which must be easily visible to him.

A major function of such a screen will be to give the driver instructions for carrying out standardized test sequences, for example production test programmes or emissions test sequences. This kind of VDU display is often graphical in form, showing for example the speed demanded and the actual speed achieved. Since in the case of emissions tests, the test profile must follow that prescribed within defined limits, it is usual to include an error checking routine in the software to avoid wasted test time. The screen may also be used to transmit instructions to supplement information conveyed by the two-way voice link.

Fire suppression

For a more general treatment of the subject of fire suppression, see Chapter 4.

The risk of a vehicle fire during chassis dynamometer running is quite high, since the air cooling, even with supplementary fans, is likely to be less than that experienced on the road. Underfloor exhaust systems, in particular, can become very hot and could ignite fuel vapour, whatever its source. A fixed fire suppression system is more difficult to design than in an engine test cell because of both the large size of rolling road cells and the more difficult access to the seat of the fire, which may be within the vehicle body.

All vehicle test facilities should be equipped with substantial handheld or hand-operated fire extinguishers and staff should be trained in their use. One method of fire protection is to fit the test vehicle with a system of the type designed for rally cars which enables the driver or control room to flood the engine compartment with foam extinguishant. Automatic gas-based systems of the type used in some engine test cells are less effective in vehicle cells in view of the greater difficulty in ensuring that all personnel, including the driver, have been evacuated before they are activated. The modern trend is towards water fog suppression systems, which may include discharge nozzles mounted beneath the vehicle and thus near the seat of most potential fires.

There must be a clear and unimpeded escape route for any test driver and the impairment in vision from steam or smoke must be taken into account, particularly in the case of anechoic cells, where the escape door positions should not be camouflaged within the coned surface.

The effect of roll diameter on tyre contact conditions

The subject of tyre rolling resistance has given rise to an extensive literature, which is well summarized in Ref. 1. The bulk of the rolling resistance is the consequence of hysteresis losses in the material of the tyre and this gives rise directly to heating of the tyre. A widely accepted formulation describing the effect of the relative radii of tyre and roll is:

$$F_{xr} = F_x \left(1 + \frac{r}{R}\right)^{\frac{1}{2}} \tag{2}$$

where:

F_{xr} = rolling resistance against drum
F_x = rolling resistance on flat road
r = radius of tyre
R = radius of drum

Figure 18.7a shows the situation diagrammatically and Fig. 18.7b the corresponding relation between the rolling resistances. This shows that rolling resistance (and hence hysteresis loss) increases linearly with the ratio r/R. For tyre and roller of equal

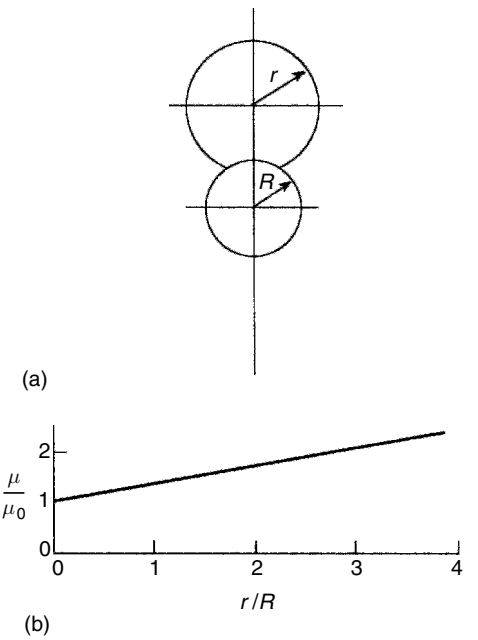


Figure 18.7 *(a) Relationship between tyre and chassis dynamometer roller diameters and (b) linear relationship between rolling resistance and the tyre/roll ratio*

diameter the rolling resistance is 1.414 × resistance on a flat road, while for a tyre three times the roller diameter, easily possible on a brake tester, the resistance is doubled. A typical value for the 'coefficient of friction' (rolling resistance/load) would be 1 per cent for a flat road.

To indicate magnitudes involved, a tyre bearing a load of 300 kg running at 17 m/s (40 mph) could be expected to experience a heating load of about 500 W on a flat road, increased to 1 kW when running on a roller of one third its diameter. Clearly the heating effects associated with small diameter rolls are by no means negligible.

Tyres used even for a short time on rolling roads may be damaged by heating and distortion effects, which are aggravated by the fact that the roll will also be heated in the course of the run. Some dynamometer systems are fitted with tyre cooling systems to reduce tyre damage but all tyres used for any but short duration tests should be specially marked and changed before the vehicle is allowed on public roads.

Specification, accuracy and calibration

A typical specification for a chassis dynamometer will include the following information as a minimum:

- base inertia;[†]
- minimum simulated inertia;
- maximum simulated inertia;
- continuous rated power, absorbing mode;
- continuous rated power, motoring mode;
- torque range, absorbing and motoring, sometimes in the form of tractive effort at base speed;
- number of rolls;
- roll diameter with tolerance (typically $< \pm 0.005$ inch on a 48-inch roll);
- roll concentricity (typically $< \pm 0.1$ mm (0.004 inch) on a 48 inch roll);
- maximum surface speed (km/hour);
- maximum permitted acceleration;
- maximum time to stop from maximum rated speed under regenerative braking;
- type of drive protection system (supply failure during full regenerative load, etc.).

These parameters determine the basic geometry of the rig and the size of motor and its associated variable speed drive system.

[†] It is common practice to use the term 'inertia' to mean the simulated or equivalent vehicle inertia, rather than the moment of inertia of the rotating components. The inertia is thus quoted in kg rather than kgm^2.

Where a purpose of the dynamometer is to carry out homologation tests, these should be clearly specified, together with the range of vehicles to be covered; the acceptance procedure should include performance of these tests.

The essential measurements performed by a chassis dynamometer, equivalent to the measurement of torque and speed in a conventional machine, are roller surface speed and the force (the tractive effort at the roll periphery) transmitted to or from the rolls.

This force is usually not measured directly, but derived from a torque measurement at the motor, performed either in the conventional manner by utilizing a trunnion-mounted motor with torque arm and load cell, or by some form of torque shaft dynamometer. Whatever method is used, there will be an in-built error arising from bearing losses and roll windage: these losses can be measured by running the machine through the speed range with no vehicle present and noting the necessary driving torque, which should be added to the observed torque when absorbing power and subtracted when braking, to give the true tractive force at the roller rim. (This is not the complete answer as there will be a further, unquantified, loss due to roll bearing friction when the rolls are under load.)

Some dynamometers include these corrections in the stated dynamometer constants (see Chapter 8 for a detailed discussion of torque calibration procedures). This is often not a simple matter in the case of chassis dynamometers as the torques involved can be very substantial and space very restricted: the preferred method using graduated weights to apply torque in 10 steps may not be practicable and dynamometers are often supplied with a single mass corresponding to full torque plus a few small weights adding up to about 10 per cent of full scale torque.

The accuracy of calibration demanded and the calibration methods to be used may be included in emissions legislation relevant to chassis dynamometer testing as part of the general tightening of standards.

Equivalent road speed measurement will usually be by means of a high resolution optical encoder on the roll shaft, thus introducing a possible source of error arising from uncertainty regarding true roll diameter, particularly the case with coated rolls. It is common practice to use a so-called ‘pi-tape’ to obtain the effective diameter by measuring the circumference.

A check on the actual base inertia of the dynamometer is made either by means of a number of ‘coast-down’ tests or by both accelerating and decelerating the rolls at an approximately constant rate, plotting motor torque and speed in both cases. The latter method is the more accurate since it effectively eliminates the internal drag of the dynamometer motor.

It will be apparent that really accurate measurements of traction/braking force are by no means easy to achieve. Much depends on the quality of manufacture and consistency of performance of the machine, the accuracy of the basic measurements of torque, roller diameter and speed and the capability of the associated computer.

A common operational problem experienced in test laboratories is that after rigging a vehicle on the chassis dynamometer and taking considerable time and effort to connect all of the required emission equipment and transducers, it is then required

to perform warm-up and coast-down checks on the dynamometer, during which the vehicle should be removed. The solution is provided on most high end designs by the incorporation of a mechanism that arcs around the drum and lifts the vehicle tyre on two small rollers just clear of the drum surface. On completion of the warm-up routine, the mechanism arcs down, out of contact, and the tyre is returned to its original position.

Limitations of the chassis dynamometer

Like all devices intended to simulate a phenomenon, the chassis dynamometer has its limitations, which are easily overlooked. It cannot be too strongly emphasized that a vehicle, restrained by elastic ties and delivering or receiving power through contact with a rotating drum, is not dynamically identical with the same vehicle in its normal state as a free body traversing a fixed surface. During motion at constant speed, the differences are minimized, and largely arise from the absence of air flow and the limitation of simulated motion to a straight line.

Once acceleration and braking are involved, however, the vehicle motions in the two states are quite different. To give a simple example, a vehicle on the road is subjected to braking forces on all wheels, whether driven or not, and these give rise to a couple about the centre of gravity, which causes a transfer of load from the rear wheels to the front with consequent pitching of the body. On a chassis dynamometer, however, the braking force is applied only to the driven wheels, while the forward deceleration force acting through the centre of gravity is absent, and replaced by tension forces in the front-end vehicle restraints (remember: braking on a rolling road tends to throw the vehicle backwards).

It will be clear that the pattern of forces acting on the vehicle, and its consequent motions, are quite different in the two cases. These differences make it very difficult to investigate vehicle ride and some aspects of NVH testing, on the chassis dynamometer. A particular area in which the simulation differs most fundamentally from the 'real-life' situation concerns all aspects of driveline oscillation, with its associated judder or 'shuffle'.

The authors have had occasion to study this problem in some detail and, while their analysis is too extensive to be repeated here, one or two of their conclusions may be of interest:

- The commonly adopted arrangement, whereby the roll inertia is made equal to that of the vehicle, gives a response to such disturbances as are induced by driveline oscillations, i.e. judder, that are significantly removed from on-road behaviour.
- For reasonably accurate simulation of phenomena such as judder, roll inertia should be at least five times vehicle inertia.
- Electronically simulated inertia is not effective in this instance: actual mass is necessary.

- The test vehicle should be anchored as lightly and flexibly as possible: this is not an insignificant requirement, since it is desirable that the natural frequency of the vehicle on the restraint should be at the lower end of the range of frequencies, typically 5–10 Hz, that are of interest.

It is strongly recommended that investigations of vehicle behaviour under conditions of driveline oscillation should proceed with caution if it is intended to involve running on a chassis dynamometer.

Summary

The subject of rolling roads and chassis dynamometers is a very extensive one, worthy of a textbook in its own right. The application of these machines, at all but an elementary level, is very largely in the field of transient testing and the road load equation. However, the testing techniques employed have a great deal in common with those applicable to engine testing dynamometers, and almost the whole of the present book is relevant.

In this chapter, an attempt has been made to survey the whole range of apparatus that may be described by the general heading of rolling roads/chassis dynamometers, from the simple garage brake tester to complex and very costly climatic chambers and anechoic cells. The design of the associated test cells, vehicle restraint systems, driver communications and safety precautions have been discussed and problems of accuracy and calibration considered.

The subject of the building/dynamometer interface has been covered in some detail because errors discovered at installation are expensive to rectify.

Reference

1. *Tire Rolling Resistance*, Symposium, 122nd Meeting, Rubber Division, American Chemical Society, 1982.

EXHIBIT 1014

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THE ENGLISH LANGUAGE



THIRD EDITION

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THE AMERICAN HERITAGE® DICTIONARY OF THE ENGLISH LANGUAGE

THIRD EDITION



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...tersunk

...sunk (koun′tər-sŭngk′) *v.* Past tense and past par... countersink.

...ten·or (koun′tər-tĕn′ər) *n.* *Music.* **1.** An adult ... with a range above that of tenor. **2.** A singer having ... within this range. —*attributive.* Often used to ... another noun: *a countertenor part; a countertenor ar...*

...ter·ror (koun′tər-tĕr′ər) *adj.* Intended to prevent ... terrorism: *counterterror measures; counterterror* ... *n.* Action or strategy intended to suppress terrorism. **—coun′ter·ter′ror·ism** *n.* ...ter′ror·ist *adj.* & *n.*

...top (koun′tər-tŏp′) *n.* A level surface on a cabinet, as in a kitchen or department store. —*attributive.* ...modify another noun: *a countertop appliance; coun...*

...trade (koun′tər-trād′) *n.* International barter in ... country agrees to import commodities from another ... which it exports commodities. **—coun′ter·trad′er** *n.*

...trans·fer·ence (koun′tər-trăns-fûr′əns, ...) *n.* The surfacing of a psychotherapist's own re... through identification with the emotions, exper... problems of a person undergoing treatment.

...vail (koun′tər-vāl′, koun′tər-vāl′) *v.* **-vailed,** ...vails. —*tr.* **1.** To act against with equal force; coun... **2.** To compensate for; offset. —*intr.* To act against an ...mental influence or power. [Middle English *coun...*, from Old French *contrevaloir*, *contrevail-* : *contre-*, ... to be worth (from Latin *valēre*, to be strong; see ...)]

...weigh (koun′tər-wā′) *intr.* & *tr.v.* **-weighed,** ...ing, **-weighs.** To counterbalance or cause to counter-...

...weight (koun′tər-wāt′) *n.* **1.** A weight used as a ... **2.** A force or influence equally counteracting. **—coun′ter·weight′ed** (-wā′tĭd) *adj.*

...wom·an (koun′tər-wo͝om′ən) *n.* A woman who ... as in a diner.

...ess (koun′tĭs) *n.* **1.** A woman holding the title of count ... **1.a.** The wife or widow of a count in various European ... **b.** The wife or widow of an earl in Great Britain. **3.** ... for such a noblewoman. [Middle English *countes*, ... *contesse*, feminine of *conte*, count. See COUNT[2].]

...ing·house also **count·ing house** (koun′tĭng-hous′) ... room or office in which a business firm carries on ... such as accounting and correspondence.

...less (kount′lĭs) *adj.* Incapable of being counted; innu... See Synonyms at **incalculable.** **—count′less·ly** *adv.*

...noun *n.* A noun for an object, such as *chair*, or for an ... experience, that speakers of a language identify as ... a single entity and that can form a plural or occur in ... construction with an indefinite article, with nu... with such terms as *many.*

...palatine *n., pl.* **counts palatine.** **1.a.** Any of various ... originally exercising certain royal powers within their ... especially a count of the Holy Roman Empire hav... powers in his own territories. **b.** A feudal lord hav... powers over his lands. **2.** The titled proprietor of a ... palatine in England or Ireland.

...fied also **coun·try·fied** (kŭn′trĭ-fīd′) *adj.* **1.** Re... having the characteristics of country life; rural. **2.** ... sophistication.

...try (kŭn′trē) *n., pl.* **-tries.** **1.a.** A nation or state. **b.** ... of a nation or state; land. **c.** The people of a nation ... populace: *The whole country will profit from the new* ... **2.** The land of a person's birth or citizenship: ... *is restricted in his country.* **3.** A region, territory, ... tract of land distinguishable by features of topography, ... culture: *hill country; Bible country.* **4.** A district out... cities and towns; a rural area: *a vacation in the country.* **5.** ... **6.** *Informal.* Country music. **—country** *adj.* **1.** Of, ... or typical of the country: *a country road; country* ... **2.** Of or relating to country music. **—idiom. in coun...** ... Vietnam during the period of U.S. military operations ... *been in country a month longer than the other four"* ... [Middle English *countre*, from Old French *con...*, from Vulgar Latin **(terra) contrāta*, (land) opposite, before, ... *contrā*, opposite. See **kom** in Appendix.]

...try and western *n.* *Abbr.* **C & W, C and W** See **coun...** ...music.

...try club *n.* A suburban club for social and sports activ... usually featuring a golf course.

...try cousin *n.* One whose ingenuousness or rustic ways ... amuse or entertain city dwellers.

...try-dance (kŭn′trē-dăns′) *n.* A folk dance of English ... in which two lines of dancers face each other.

...try·fied (kŭn′trĭ-fīd′) *adj.* Variant of **countrified.**

...try gentleman *n.* A man who owns a country estate.

...try·man (kŭn′trē-mən) *n.* **1.** A person from one's own ...; a compatriot. **2.** A native or inhabitant of a particular ... **3.** A man who lives in the country or has country ways.

...try mile *n.* *Informal.* A very great distance.

...try music *n.* *Music.* Popular music based on the folk style of the southern rural United States or on the music of cowboys in the American West. Also called *country and western*.

coun·try·seat (kŭn′trē-sēt′) *n.* An estate or mansion in the country.

coun·try·side (kŭn′trē-sīd′) *n.* **1.** A rural region. **2.** The inhabitants of a rural region.

country singer *n.* *Music.* One who sings country music.

coun·try·wide (kŭn′trē-wīd′) *adv.* & *adj.* Throughout a whole country; nationwide: *launched a fundraising campaign countrywide; a countrywide search.*

coun·try·wom·an (kŭn′trē-wo͝om′ən) *n.* **1.** A woman from one's own country; a compatriot. **2.** A woman from a particular country. **3.** A woman who lives in the country or has country ways.

coun·ty (koun′tē) *n., pl.* **-ties.** *Abbr* **co, co.** **1.** An administrative subdivision of a state in the United States. **2.a.** A territorial division exercising administrative, judicial, and political functions in Great Britain and Ireland. **b.** The territory under the jurisdiction of a count or earl. **3.** The people living in a county. [Middle English *counte*, territorial division, from Old French *conte*, the territory of a count, from Medieval Latin *comitatus*, from Late Latin, the office of count, from Latin, retinue, from *comes, comit-*, companion. See **ei-** in Appendix.] **—coun′ty** *adj.*

county agent *n.* A government employee who serves as a consultant and adviser in a chiefly rural county on such matters as agriculture, education, and home economics.

county fair *n.* A fair usually held every year at the same location in a county, especially for the competitive showing of livestock and farm products.

county palatine *n., pl.* **counties palatine.** The domain of a count palatine in England or Ireland.

county seat *n.* A town or city that is the administrative center of its county.

county town *n.* *Chiefly British.* A county seat. Also called *shire town.*

coun·ty·wide (koun′tē-wīd′) *adv.* & *adj.* Throughout a whole county: *found at locations countywide; a countywide war on drugs.*

coup (ko͞o) *n., pl.* **coups** (ko͞oz). **1.** A brilliantly executed stratagem; a masterstroke. **2.** A coup d'état. **3.** Among certain Native American peoples, a feat of bravery performed in battle, especially the touching of an enemy's body without causing injury. **—idiom. count coup.** Among certain Native American peoples, to ceremoniously recount one's exploits in battle. [French, stroke, from Old French *colp*, from Late Latin *colpus*. See COPE[1].]

coup de grâce (ko͞o′ də gräs′) *n., pl.* **coups de grâce.** **1.** A deathblow delivered to end the misery of a mortally wounded victim. **2.** A finishing stroke or decisive event. [French : *coup*, stroke + *de*, of + *grâce*, mercy.]

coup de main (də măɴ′) *n., pl.* **coups de main.** A sudden action undertaken to surprise an enemy. [French : *coup*, stroke, blow + *de*, of + *main*, hand.]

coup d'é·tat (dā-tä′) *n., pl.* **coups d'état** or **coup d'états** (-dā-tä′). The sudden overthrow of a government by a usually small group of persons in or previously in positions of authority. [French : *coup*, blow, stroke + *de*, of + *état*, state.]

coup de thé·â·tre (də tā-ä′trə) *n., pl.* **coups de théâtre.** **1.** A sudden, dramatic turn of events in a play. **2.** An unexpected and sensational event, especially one that reverses or negates a prevailing situation. [French : *coup*, stroke + *de*, of + *théâtre*, theater.]

coup d'oeil (dœ′yə) *n., pl.* **coups d'oeil.** A quick survey; a glance. [French : *coup*, stroke + *de*, of + *oeil*, eye.]

coupe[1] (ko͞op) *n.* **1.a.** A dessert of ice cream or fruit-flavored ice, garnished and served in a special dessert glass. **b.** The stemmed glass in which a coupe is served. **2.** A shallow, bowl-shaped dessert dish. [French, cup, from Late Latin *cuppa*.]

coupe[2] (ko͞op) *n.* Variant of **coupé** (sense 2).

cou·pé (ko͞o-pā′) *n.* **1.** A closed four-wheel carriage with two seats inside and one outside. **2.** Also **coupe** (ko͞op). A closed two-door automobile. [French, from past participle of *couper*, to cut, from *coup*, blow. See COUP.]

Cou·pe·rin (ko͞o-pə-răɴ′, ko͞op-răɴ′), **François.** 1668–1733. French composer who was court organist at Versailles during the reign of Louis XIV.

cou·ple (kŭp′əl) *n.* **1.** Two items of the same kind; a pair. **2.** Something that joins or connects two things together; a link. **3.** (*used with a sing. or pl. verb*). **a.** Two people united, as by betrothal or marriage. **b.** Two people together. **4.** *Informal.* A few; several: *a couple of days.* **5.** *Physics.* A pair of forces of equal magnitude acting in parallel but opposite directions, capable of causing rotation but not translation. **—couple** *v.* **-pled, -pling, -ples.** —*tr.* **1.** To link together; connect: *coupled her refusal with an explanation.* **2.a.** To join as spouses; marry. **b.** To join in sexual union. **3.** *Electricity.* To link (two circuits or currents) as by magnetic induction. —*intr.* **1.** To form pairs; join. **2.** To unite sexually; copulate. **3.** To join chemically. **—couple** *adj.* *Informal.* Two or few: *"Every couple years the urge strikes, to ... haul off to a new site"* (Garrison Keillor). [Middle English, from Old French, from Latin *copula*, bond, pair.]

SYNONYMS: *couple, pair, brace, yoke.* These nouns denote two individuals of the same kind together. *Couple* does not necessarily

ă pat	oi boy
ā pay	ou out
âr care	o͝o took
ä father	o͞o boot
ĕ pet	ŭ cut
ē be	ûr urge
ĭ pit	th thin
ī pie	*th* this
îr pier	hw which
ŏ pot	zh vision
ō toe	ə about, item
ô paw	♦ regionalism

Stress marks: ′ (primary) ′ (secondary), as in dictionary (dĭk′shə-nĕr′ē)

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EXHIBIT 1015

Internal Combustion Engines

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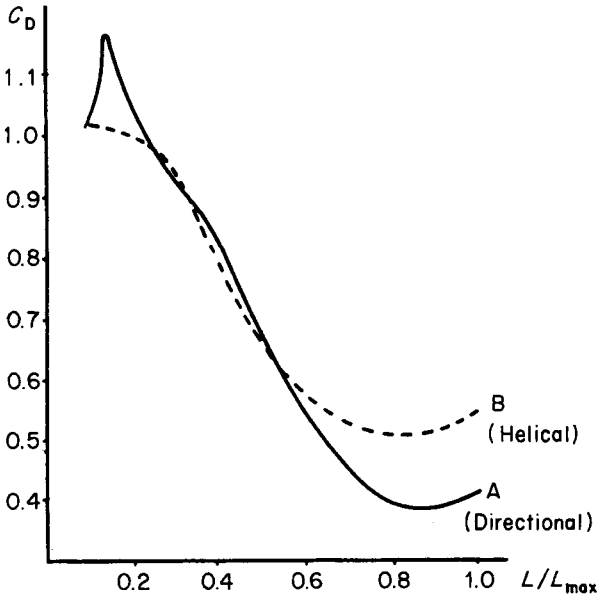


Figure 11 Variation of flow coefficient with valve lift for directed and helical inlet ports (Tindal and Williams, 1977). Reprinted with permission © 1977 Society of Automotive Engineers, Inc.

2. *Steady flow testing of inlet ports*

Before proceeding to discuss the development of the air motion produced by helical and directed ports during the induction and compression strokes, it will be useful to consider steady flow testing in more detail. This well-established technique is still widely used in the development of inlet ports. The cylinder head itself, or a prototype model which may be made of wood or other easily worked material, is mounted on a rig which supplies a steady flow of air through the port and valve. The cylinder is simulated by a long tube of equivalent diameter; this is usually made of transparent plastic in order to allow visualization of the flow. The rate of swirl of the air within the tube is measured over a range of fixed valve lifts, the pressure drop across valve and port being maintained at the same value (commonly 25 cm of water) for all lifts. Figure 12 (Watts and Scott, 1970) shows a typical rig. The swirl rate is measured at a fixed distance from the head (typically one cylinder diameter) by means of a vane anemometer or an impulse momentum meter (Figure 13). The vane anemometer employs a rotor supported on low friction bearings, mounted coaxial with the plastic tube; the speed of rotation may be determined by means of a stroboscope or by an electronic counter coupled to a sensor in the tube wall. It has the disadvantage that it will only respond

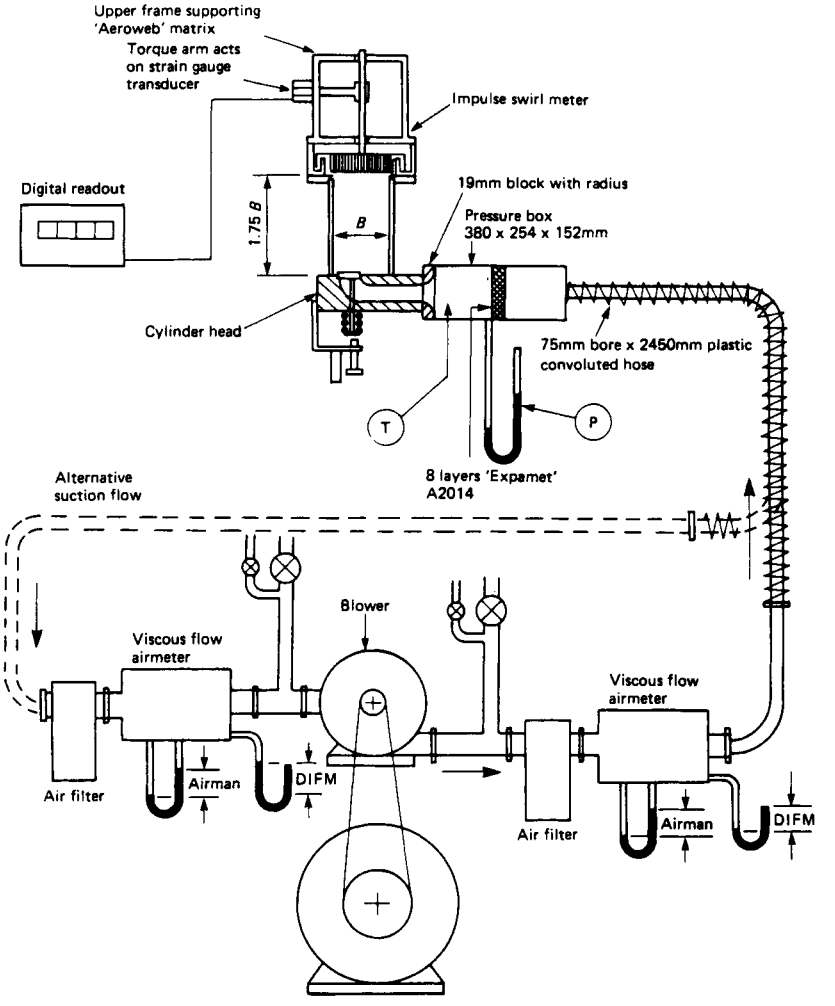


Figure 12 Steady flow rig for testing inlet ports (Watts and Scott, 1970). Reprinted by permission of the Council of the Institution of Mechanical Engineers from Symposium on Diesel Engine Combustion, 1970.

correctly to a swirl which is coaxial with the cylinder; it will give little or no reaction to a motion such as that illustrated in Figure 9(*a*), even though the air velocities may be quite high. The impulse momentum meter (Tippleman, 1977), on the other hand, responds directly to the angular momentum of the air in the tube. It embodies a circular element of honeycomb structure which is a clearance fit inside the tube and acts as a flow rectifier, turning the air into the axial direction. The torque acting on the flow rectifier can easily be measured and is equal to the angular momentum flux.

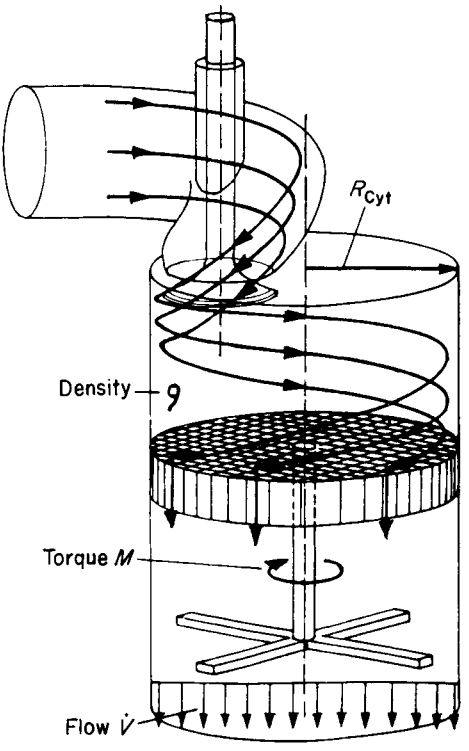


Figure 13 Impulse momentum meter (Tippleman, 1977). Reprinted with permission © 1977 Society of Automotive Engineers, Inc.

Thus the device provides an accurate measurement of angular momentum, irrespective of the velocity distribution. The steady flow test is performed under conditions which differ in a number of respects from those which exist in an engine. In particular the air flow into an engine cylinder is unsteady and must be influenced to some extent by the presence of the piston. Nevertheless the procedure is a valuable means of comparing the characteristics of different port shapes.

3. Cylinder air flow measurements in motored engines

To obtain accurate and detailed information about the air motion in the cylinder during the induction and compression strokes, it is necessary to make measurements inside a motored engine; the techniques most commonly employed are laser-Doppler anemometry and hot wire anemometry (Witze, 1980; Durst *et al.*, 1981; Monaghan *et al.*, 1981; Tindal *et al.*, 1982; Dyer, 1985). The former has a number of advantages — for example it causes no

fer. Consequently the rate of increase of temperature of the surface of the droplet slows down, the temperature gradient within the liquid becomes less steep, and a point is reached where all the heat received by the droplet goes to vaporize the fuel, the temperature becoming constant at a value which is often referred to as the "wet-bulb temperature" by analogy with psychometrics.

Godsave (1953) found that the burning rate of a fuel was less dependent on its volatility, as represented by the vapour pressure, than on the rate of heat transfer. He concluded that the critical property of the fuel was the enthalpy of evaporation. From the results of experiments, Godsave and others have found that the droplet size decreases in such a way that the square of the diameter is a linear function of time. More precisely,

$$D^2 = D_0^2 - \lambda t$$

where D_0 = diameter at $t = 0$ and λ is called the evaporative constant.

If the droplet is assumed to be spherical, its volume at time t is given by $\frac{\pi D^3}{6}$. It may easily be shown that

$$\frac{\mathrm{d}}{\mathrm{d}t}\left(\frac{\pi D^3}{6}\right) = -\frac{\pi\lambda}{4}D$$

and if a constant mean density, ρ_m, is assumed for the droplet liquid, the rate of change of mass is given by

$$-\frac{\rho_m \pi \lambda}{4}D.$$

In other words, the instantaneous rate of change of mass of the droplet is directly proportional to the diameter. Figure 26 shows the results of measurements by Nishiwaki (1955) of the way in which the size of water droplets of various initial diameters decreased after they were suddenly introduced into a furnace where the temperature was 620°C. For almost the whole of the evaporation period, D^2 is directly proportional to time, the temperature of the droplet being constant at the "wet-bulb" value. During the first half second or so, the diameter is decreasing at a much slower rate; this is because the droplet temperature is still rising to its final value.

In another experiment Nishiwaki built an apparatus in which a droplet of fuel, suspended on a thin filament of silica, could suddenly be plunged into an environment of high temperature air. The change in size of the droplet was monitored by cine-photography. Figure 27 shows D^2 plotted against time for a droplet of α-methyl napthalene; the air temperature was 713°C. The diagram may conveniently be divided into several phases. Initially the droplet temperature is rising and vaporization is negligible; D^2 increases slightly as

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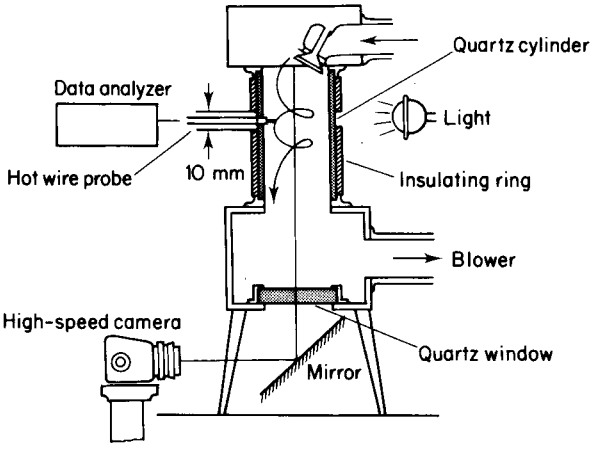


Figure 27 Experimental apparatus for gas flow study in cylinder by high speed photography and hot wire anemometry (Matsushita *et al.*, 1985).

hot wire anenometry was used in the engine development process for both diesel and spark-ignition engines. Although it is hard to find good examples of the actual application of LDA to engine design it is quite reasonable to expect that LDA could replace hot wire anemometry in most development tasks. For example, the development of the Toyota lean-burn system, which was previously discussed with respect to the use of flow visualization in engine design, also benefited from quantitative measures of the swirl velocity profiles using hot wire anemometry (see Figures 27 and 28). Today, the measurement of swirl could be performed both with more ease and accuracy through the use of LDA. In the previous section we discussed how the Toyota researchers used visualization techniques to develop their system. However, they supplemented the visualization results with quantitative measures of swirl in the cylinder to aid them in their decision process. In addition to the swirl profiles in the cylinder, the Toyota engine developers were also interested in more detailed measurements of velocity profiles around the intake valve to characterize the intake port flows. A similar measurement, coupled with three-dimensional modelling, is being performed using LDA to study the effects of port shape on in-cylinder flows by the AVL researchers Brandstatter *et al.* (1985). Since engine flows are particularly complicated, engine developers, such as Toyota and AVL, use all possible techniques to insure a successful design.

The effect of combustion chamber shape on the flow field is of particular interest to diesel engine developers and hot wire anemometry has been used extensively to characterize the flows produced by various geometries of

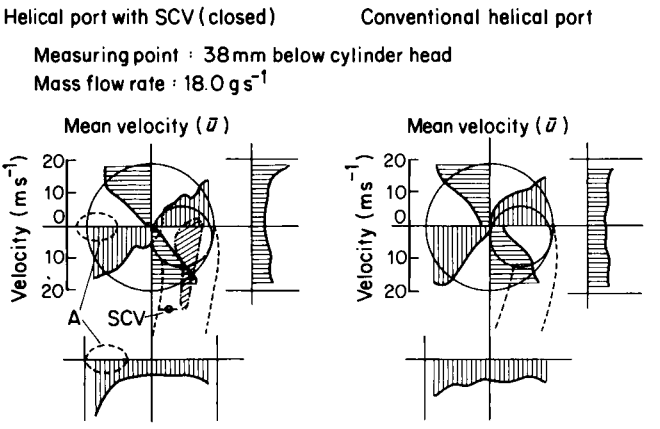


Figure 28 Flow velocity pattern for conventional helical port and SCV valve closed (Matsushita *et al.*, 1985).

the piston bowl. One study compares the flow fields produced by a toroidal design to those of a re-entrant design (Figure 29). By using shielded hot wires, it was determined that the re-entrant design held the swirl in the bowl longer during expansion process, thus explaining the better performance of this chamber at retarded injection timings. An LDA system would have yielded similar information with more confidence in the accuracy of the measured flow velocities and it is expected that LDA will replace the hot wire for most in-cylinder engine applications.

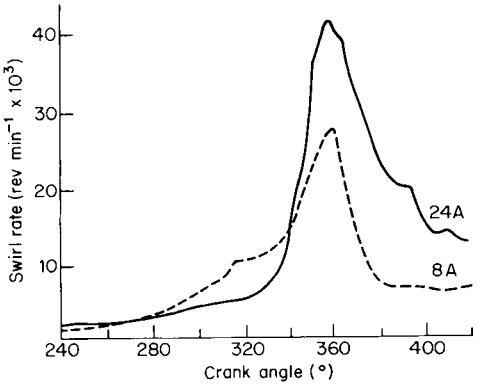


Figure 29 Comparison of swirl rates in toroidal and re-entrant chambers. Case 24a denotes the re-entrant chamber (Williams and Tindal, 1980).

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Utilisation in Combustion Engines



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Generally two types of viscometers are used to measure the viscosity of a fuel:

1. Redwood viscometer
2. Saybolt viscometer

6.3.2.1 Redwood Viscometer

A redwood viscometer consists of a cylindrical oil cup furnished with a gauge point, a metallic orifice jet at the bottom having a concave depression from inside to facilitate a ball with stiff wire to act as a valve to start or stop oil flow as shown in Figure 6.2. The outer side of the orifice jet is convex, so that the oil under test does not creep over the lower face of the oil cup. The oil cup is surrounded by a water bath with a circular electrical immersion heater and a stirring device. Two thermometers are provided to measure the water bath

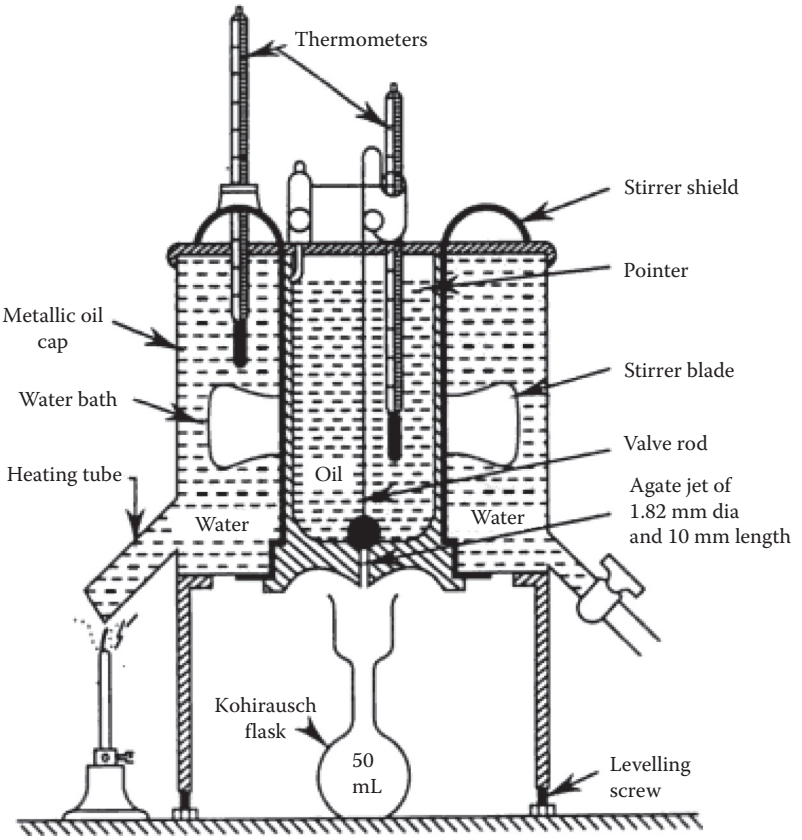


FIGURE 6.2
Redwood viscometer.

temperature and oil temperature under test. A round flat-bottomed flask of 50 mL marking is used to measure 50 mL of oil flow against time. The water bath with oil cup is supported on a tripod stand with levelling screws.

6.3.2.1.1 Procedure

The oil cup is cleaned with a solvent, preferably carbon tetrachloride (CTC), and it is wiped dry thoroughly with a paper napkin or a soft cloth (do not use cotton waste) and the orifice jet is wiped with a fine thread. The water bath is kept with the oil cup on the tripod stand and is levelled. Water is poured into the water bath up to 15–20 mm below the top portion. The ball (valve) is kept in position and clean filtered oil is poured as the sample (use a strainer not coarser than BS 100 mesh) to be tested into the oil cup up to the gauge point and it is covered with the lid. A 50-mL flask with clean dry is placed under the orifice jet of the oil cup. The ball (valve) is lifted and a stopwatch is simultaneously started and the oil is allowed into the receiving flask. The receiving flask (50 mL) is adjusted in such a way that the oil string coming out of the jet strikes the neck of the flask to avoid foaming (formation of air bubbles) on the oil surface. One has to wait till the oil level touches the 50-mL mark. The stopwatch is stopped and the time is recorded. The experiment is repeated at different temperatures above ambient.

6.3.2.2 Saybolt Viscometer

This apparatus mainly consists of a standard cylindrical oil cup surrounded with a water bath with an immersion heater and a stirring device (Figure 6.3). The apparatus is supplied with two stainless steel orifice jets, namely a universal jet and Furol jet, which can be fitted at the bottom of the oil cup as per our requirement. A rubber cork stopper arrangement is also provided at the bottom to facilitate the start and stop of the oil flow from the viscometer. Two thermometers are provided to measure the water bath temperature and oil temperature under test. A round flat-bottomed flask with a 60-mL marking on the neck is provided to measure 60 mL of oil flow against time. The oil cup with the water bath is supported on a stand with levelling screws.

6.3.2.2.1 Procedure

The oil cup is cleaned with a solvent, preferably CTC, and it is wiped dry thoroughly with a paper napkin or a soft cloth (cotton waste is not used) and the orifice jet is wiped with a fine thread. The water bath is kept with the oil cup on the tripod stand and is levelled. Water is poured into the water bath up to 15–20 mm below the top portion. The orifice opening is closed from the bottom with the rubber cork provided. The oil to be tested is poured into the strainer by keeping the strainer on the oil cup until the oil fills the cup as well as in the side well. The excess oil is withdrawn in the side well and the thermometers are positioned in the water bath and in the oil cup. A clean dry 60-mL flask is taken and placed under the orifice jet of the oil cup that centres

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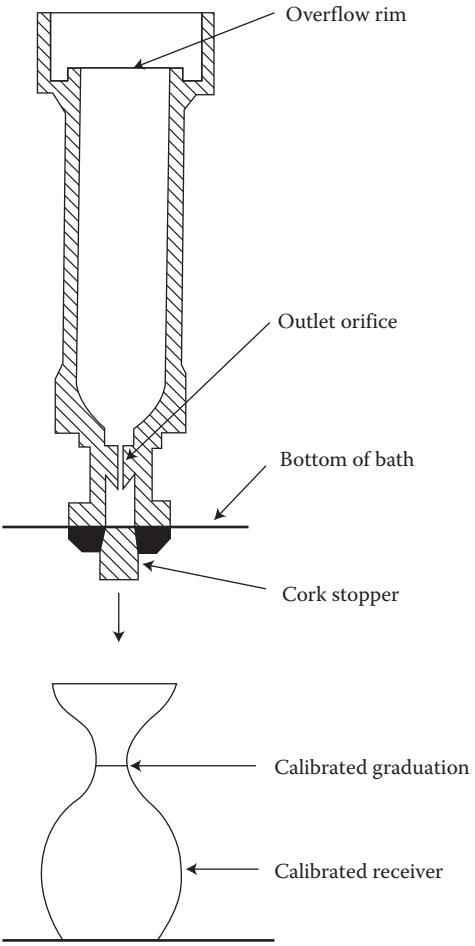


FIGURE 6.3
Saybolt viscometer.

it. The rubber cork is pulled open and a stop watch is simultaneously started and the oil is allowed into the receiving flask. The receiving flask (60 mL) is adjusted in such a way that the oil string coming out of the jet strikes the neck of the flask to avoid foaming (formation of air bubbles) on the oil surface. When the oil level reaches the 60-mL mark, the watch is stopped and the time is recorded in seconds. Repeat the experiment at different temperatures above ambient. Use a specific nozzle suitable for the lubricant or the oil.

6.3.2.2.2 Calculation

$$\textit{Kinematic viscosity } (\nu) = (A \times t - (B/t)) \times 10^{-6} \text{ in } \mathrm{m^2/s} \qquad (6.8)$$

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where
$A = 0.264$, $B = 190$, when $t = 40–85$ s
$A = 0.247$, $B = 65$, when $t = 85–2000$ s

The *density* of the given oil is

$$\rho = (w_2 - w_1) \times (10^3/50) \text{ in kg/m}^3 \tag{6.9}$$

where
w_1 = weight of the measuring jar in grams
w_2 = weight of 50 cc of oil in grams

$$\textit{Absolute viscosity } (\mu) = \rho \times \nu \text{ in Pa s or N s/m}^2 \tag{6.10}$$

6.3.3 Flash Point

The flash point is the lowest temperature corrected to a barometric pressure of 101.3 kPa (760 mm Hg), at which the application of an ignition source causes the vapours of a specimen of the sample to ignite under specified conditions of the test. The flash point temperature is one measure of the tendency of the test specimen to form a flammable mixture with air under controlled laboratory conditions. It is only one of a number of properties that must be considered when assessing the overall flammability hazard of a material. It is used in shipping and safety regulations to define *flammable* and *combustible* materials. An Ables apparatus is used to determine the flash point that is shown in Figure 6.4.

6.3.3.1 Description of Apparatus

In the Pensky–Martens closed cup flash point test, a brass test cup is filled with a test specimen and is fitted with a cover. The sample is heated and stirred at specified rates depending on what is being tested. An ignition source is directed into the cup at regular intervals with simultaneous interruption of stirring until a flash that spreads throughout the inside of the cup is seen. The corresponding temperature is its flash point. The Pensky–Martens closed cup is sealed with a lid through which the ignition source can be introduced periodically. The vapour above the liquid is assumed to be in reasonable equilibrium with the liquid. Closed cup testers give lower values for the flash point (typically 5–10 K) and are a better approximation of the temperature at which the vapour pressure reaches the lower flammable limit (LFL).

6.3.3.2 Procedure

The apparatus is installed on a table near a 230-V, 50-Hz, 5-A single-phase power source. The electrical heater is kept on the table. The oil cup holder is

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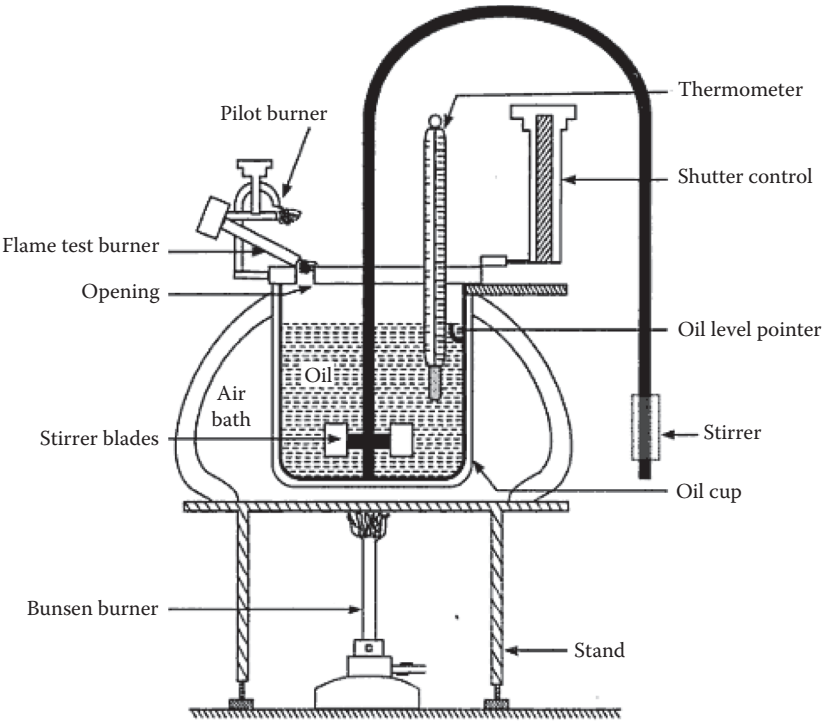


FIGURE 6.4
Ables apparatus.

positioned (air bath) on the heater. The oil cup is inserted into the bath and positioned in it. The oil to be tested is poured into the oil cup up to the mark. The lid is closed, the heater is connected to the electrical power source and the oil is heated at a slow and steady rate of 2°C/min with the help of the regulator. By stirring the oil with the stirring mechanism, a small flame is maintained on the wick. The flame is introduced to the oil surface by operating the circular handle, which makes the maintained flame that dips into the oil cup by opening the shutter. This is done at every 1/2 min, only after the sample oil reaches 15–17°C before the expected flash point. Record the temperature at which the first flash is recorded and that is reported as the flash point of the sample oil. To stop the experiment, switch off the heater and allow it to cool.

6.3.4 Cloud Point and Pour Point

The *cloud point* is the temperature of a liquid specimen when the smallest observable cluster of hydrocarbon (HC) crystals first occurs upon cooling under prescribed conditions. The purpose of the cloud point method is to detect the presence of wax crystals in the specimen; however, trace amounts

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of water and inorganic compounds may also be present. The intent of the cloud point method is to capture the temperature at which the liquids in the specimen begin to change from a single liquid phase to a two-phase system containing a solid and a liquid. The specimen is cooled at a specified rate and is examined periodically. The temperature at which a cloud is first observed at the bottom of the test jar is recorded as the cloud point. The cluster of wax crystals looks like a patch of whitish or milky cloud, hence the name of the test method. The cloud appears when the temperature of the specimen is low enough to cause wax crystals to form.

The *pour point* is the lowest temperature at which movement of the test specimen is observed under the prescribed conditions of the test. The pour point of a petroleum specimen is an index of the lowest temperature of its utility for certain applications.

6.3.4.1 Description of Apparatus

The apparatus for measuring cloud and pour points is shown in Figure 6.5, which consists of a cylindrical glass test jar with a 33.2–34.8 mm outside diameter and 115 and 125 mm height. The inside diameter of the jar may range

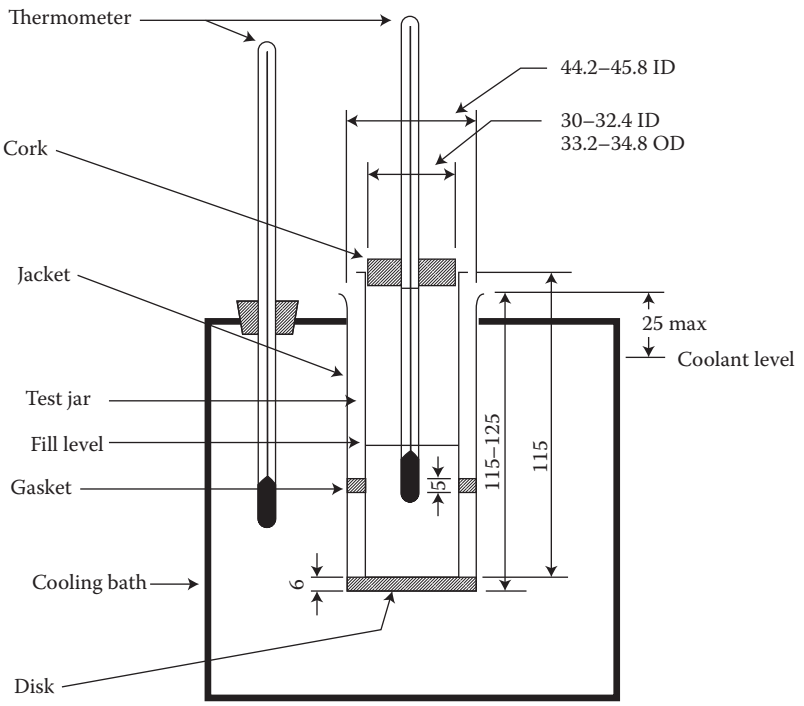


FIGURE 6.5
Apparatus for measuring cloud point and pour point.

from 30 to 32.4 mm within the constraint that the wall thickness must not be greater than 1.6 mm. The jar should be marked with a line to indicate the sample height 54 ± 3 mm above the inside bottom. This test jar is to be inserted in a metal or a glass jacket, watertight, cylindrical, flat bottom, about 115 mm in depth, with an inside diameter of 44.2–45.8 mm. It shall be supported free of excessive vibration and firmly in a vertical position in the cooling bath of 6.7 so that not more than 25 mm projects out of the cooling medium. Thermometers should be placed inside the apparatus as shown in Figure 6.5.

6.3.4.2 Procedure

6.3.4.2.1 Cloud Point

The sample to be tested is brought to a temperature at least 14°C above the expected cloud point. Any moisture that may be present is removed by a method such as filtration through dry lintels filter paper until the oil is perfectly clear, but makes such filtration at a temperature of at least 14°C above the approximate cloud point. The sample is poured into the test jar to the level mark. The test jar is tightly closed by the cork carrying the test thermometer. High cloud and pour thermometers are used if the expected cloud point is above –36°C and a low cloud and pour thermometers are used if the expected cloud point is below –36°C. The position of the cork and the thermometer is so adjusted that the cork fits tightly, the thermometers and the jar are coaxial and the thermometer bulb is resting on the bottom of the jar. See that the disk, gasket and the inside of the jacket are kept clean and dry. The disk is placed in the bottom of the jacket. The disk and jacket shall have been placed in the cooling medium for a minimum of 10 min before the test jar is inserted. A jacket cover may be used while the empty jacket is cooling. The gasket is placed around the test jar, 25 mm from the bottom. The test jar is inserted in the jacket. The jar should not be placed directly into the cooling medium. The temperature of the cooling bath is maintained at 0 ± 1.5°C.

At each test thermometer reading that is a multiple of 1°C, the test jar is removed from the jacket quickly but without disturbing the specimen; after inspecting the cloud, it is replaced in the jacket. This complete operation shall require not more than 3 s. If the oil does not show a cloud when it has been cooled to 9°C, the test jar is to be transferred to a jacket in a second bath maintained at a temperature of –186 ± 1.5°C (see Table 6.2). The jacket should not be transferred. If the specimen does not show a cloud when it has been cooled to –6°C, the test jar is transferred to a jacket in a third bath maintained at a temperature of –336 ± 1.5°C. For the determination of very low cloud points, additional baths are required; each bath needs to be maintained in accordance with Table 6.2. In each case, the jar is transferred to the next bath, if the specimen does not exhibit the cloud point and the temperature of the specimen reaches the lowest specimen temperature in the range identified for the current bath in use, based on the ranges stated in Table 6.2. The cloud

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EXHIBIT 1017

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PREFACE

The introduction of SI Metric Units has necessitat to British Standards and other National Stan technical literature should reflect or anticipat principal objective of the eleventh edition o Reference Book which has been completely rev given in SI units only, even though in some instar information are in the inch or MKSA systems.

It is hoped that this book will meet the nee management personnel in the Mechanical, Chem ing disciplines. It should also prove valuabl Engineers and College Lecturers and Students.

The contributors who are drawn from Ind Associations and Educational bodies have d sufficient depth to give a thorough coverage readily understood by the reader. Further infor numerous tables and illustrations. An extensive each section which gives reference to national an literature and standards covering the subject ma

Much of the technology is common to many work of the International Standards Bodies and for the removal of barriers to trade. Many of the referred to in the text may therefore be compare National bodies.

Absolute units at first present difficulty to thos inch and metric systems and it takes use to beco of a newton or a pascal as a force or pressure/ because the gravitational systems do not differ loads (mass) and forces. The expressions, however when the units are in the coherent (SI) system many special names used in the older systems. N convert empirical data to another system of units to do this unilaterally. A case in point is gearing, are not yet fully converted to SI. Some of the pr on the former unit systems as, for example, pipe the inch nominal sizes in the metric standards ar which are based on the material stresses in kgf un prevent the use of SI units for expressing the pro

This book describes a wide variety of enginee established practices and technology in metric International level so far as it is at present practi

Extracts from British Standard publications are the British Standards Institution, 2 Park Street, L copies of the complete publication may be obtain

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13 POWER TRANSMISSION

STANLEY W. JONES

A full consideration of power transmission elements involves reference to numerous British Standards which have not yet been issued in SI units. Many such standards contain empirical formulae which are not valid in a consistent system of units and in these special cases the original form has been quoted. It is undesirable to anticipate action by the British Standards Institution and where Imperial units are currently the major source of reference, a metric conversion has been added.

ENERGY TRANSFER

This section, which deals with methods of energy transfer, outlines the principles which define the range of operation of various shaft mounted elements concerned with the transmission of power. The basis of shaft design for various operating criteria will also be examined.

The diversity of requirements for power transmission connections between prime mover and driven machine has led to the development of a wide range of clutches and couplings.

Couplings

Some manufacturers market more than six different types of coupling, several in combination with clutch and braking capability. Although certain applications are not critical as to the type of connection used, it is wise to compare alternative designs on a feasibility basis.

In the simplest type of *rigid* coupling the shafts rotate as one and any initial errors are transmitted to the driven member. Conventionally one half has a concentric spigot on its flange which fits into a recess in the other half flange; the fit being arranged to prevent lateral loads being taken by the connecting bolts. End thrust cannot be accommodated, neither is rapid disconnection possible.

Oldham couplings are rigid types intended to connect parallel shafts. The flanged ends on each shaft, which have a central grooved face, are connected by a central member having a tenon on each face, offset by 90°.

Hooke's joint or coupling has a star-like configuration made up of four equidistant arms, the pairs of which are pin-jointed into forks carried by the two shafts. Relative angular displacement of the shafts about the perpendicular centres of the pins is thus provided. If there is an appreciable difference between centre lines of parallel shafts, a cardan shaft with a joint at each end can be provided and this reduces the variation in velocity ratio.

A *flexible* coupling specification should differentiate between the ability to tolerate misalignment and the inherent torsional flexibility. The specification should also clearly define the degree of flexibility required, whether this be radial (torsional) or in respect of an axial or end float motion. Data to be assembled should list the nature of torque

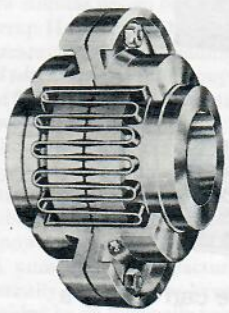
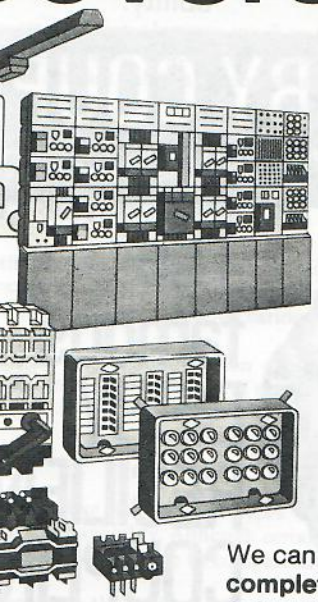
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fluctuation at particular speeds, the lack of concentricity between the shafts to be connected and any limitations in terms of space availability or environment. Manufacturer's recommendations should be sought in respect of peak loadings and the degree of misalignment that can be permitted.

Coupling characteristics exert considerable influence on the vibratory response of a shaft system and the type chosen should help in controlling response when the system is subjected to various stimuli. Lateral vibrations may be transmitted from adjacent machinery or arise within the system itself due to unbalanced masses. Torsional vibration will be minimised if operating ranges avoid critical speed areas. The degree of response in a 'soft' coupling will permit smooth running at speeds much above critical, alternatively a 'stiff' coupling will place certain orders of frequency outside the operating speed range.

The simplest *spider* type of flexible coupling takes its name from the rubber star or cruciform which engages with the metal jaws of each half coupling. Maximum capacity is 4 hp (3 kW) at 100 rev/min or its equivalent. In *Disc* couplings the half body flanges are fitted with steel drive pins which engage with a rubberised fabric centre disc. Torsional flexibility is small but compensated for by good capacity under impulsive loading and powers up to 700 hp (520 kW) at 1800 rev/min can be accommodated.

In *pin* type couplings, one half body has pins mounted on flexible buffers which engage in holes bored in the other half body. Some variations of this type provide a brake drum facility and incorporate shear pin overload protection. Powers up to 3500 hp (2610 kW) at 100 rev/min are typical.

Couplings which rely upon the power transmitting capability of rubber-like materials usually involve the element in shear or compression loading. Rubber in shear will permit a low stiffness design but is not capable of sustaining a large stress because of its tendency to creep. However, the high bulk modulus of rubber can be utilised by arranging mountings to create compression loading and a progressive increase in stiffness may thus be provided.

Hydrokinetic or fluid couplings transmit power due to the kinetic energy transfer in a fluid filling operation. Essentially, the torque developed depends upon the speed difference between the two halves and the parameter characteristics of the fluid. A variable speed facility is available if a convenient reservoir can be tapped.

Spring type couplings are grooved in the periphery of each half body to permit connection by flat springs which are intended to flex in the process of providing radial flexibility and absorbing shock loads. A totally enclosed lubricated container is typical for powers up to 650 hp (485 kW) at 100 rev/min.

A number of manufacturers have developed *toothed gear* couplings which utilise externally toothed hubs keyed to driving and driven shafts. Connection between them is provided by an internally toothed sleeve and the assembly is enclosed to ensure adequate lubrication between mating teeth. These types have a high power/size ratio and are readily available for capacities up to 500 hp (375 kW) at 100 rev/min.

A simple way of connecting two shafts is by the use of *chain* couplings. This involves mounting single row chain wheels on the end of each shaft and fitting a duplex chain for inter-connection. The assembly is enclosed and grease lubricated. Although misalignment flexibility is small, the simplicity of disconnection is attractive for many applications and powers up to 2500 hp (1865 kW) at 750 rev/min are typical. Effective coupling design depends on the type of fit contemplated for the bolts or other connecting media, since this determines how the load is shared. Methods which depend on an equation of the torsional capacity of shaft with bolt resistance are unlikely to yield accurate results for dynamic situations, although equality of shearing might be used for a preliminary assessment of non-critical dimensions.

Clutches

Clutches are used when a connection to an intermittently rotating member is required. A positive connection involves engagement between projecting jaws with provision

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EXHIBIT 1018

Carsten Baumgarten

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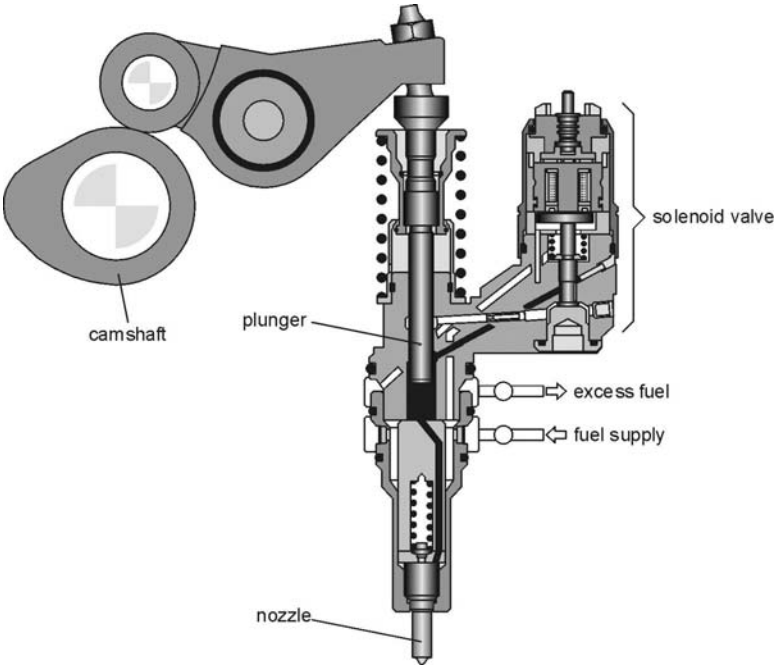


Fig. 2.27. Unit Injector [46]

The second main group of high-pressure injection systems are those in which the generation of injection pressure and the injection itself occur synchronously. These systems are driven by a camshaft, which is mechanically coupled with the engine. A basic characteristic of these systems is the intermittent pressure generation: high pressure is only available during a small crank angle interval.

In the case of the so-called unit injector system (UIS), the pump and the injection nozzle are combined into a single unit, see Fig. 2.27. Each cylinder of the engine is equipped with such a unit, which is driven by the cam via a rocker arm for example. The omission of the high-pressure pipes between the pump and the injector allows significantly higher peak injection pressures (about 200 MPa and more) than in the case of the unit pump system, see below. The shape of the cam determines the motion of the plunger and thus the generation of pressure as a function of crank angle. The spring at the upper part of the injector presses the plunger against the rocker arm and the rocker arm against the cam, and guaranteeing force closure during operation.

Fig. 2.28a shows a simplified picture of the unit injector with a closed needle. The solenoid valve is opened, and as soon as the plunger is pushed down, excess fuel is carried into the scavenge line. According to the desired start of injection, the control unit of the engine closes the solenoid valve, and the plunger compresses the fuel. As soon as the pressure is large enough to lift the spring-loaded needle, injection begins, Fig. 2.28b. Due to the large feed rate of the plunger, the

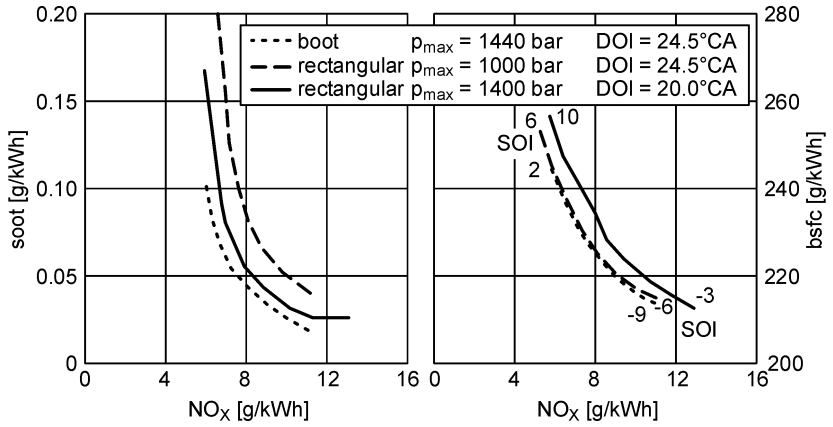


Fig. 6.7. Comparison of rectangular and boot-type injection, single cylinder engine, displacement: 1 liter, 1400 rpm, 100% load, no EGR [52]. SOI and DOI: start and duration of injection

specific power and decreasing the soot emissions, which are known to increase when applying EGR. In all cases, a very fast needle closing is advantageous in order to avoid the formation of soot due to a deteriorated mixture preparation.

Post-Injection

There are two kinds of post-injections, the late one and the early one. Late post-injection is applied in order to provide the exhaust gases with a well-defined quantity of evaporated but unburned fuel, which is then used in the catalytic reactions inside the exhaust gas after-treatment system. Injection pressure, amount of fuel, and injection timing must be carefully adjusted in order to prevent oil dilution. In the case of EGR, the late post-injection can alter the ignition delay, because some of the injected and unburned fuel is transported back into the cylinder.

The early post-injection directly follows the main injection and is a very effective measure to reduce soot emissions. Small amounts of fuel are injected with high pressure. Due to the small dwell time between main and post-injection, the injection is cut, and air is entrained in the near nozzle region and in the partial oxidation zone of the main combustion. It is very important to achieve a fast and complete closing of the injector needle in order to realize this additional entrainment of air and to avoid an unintended post-injection of low-speed large fuel drops which would significantly increase the emissions of soot and unburned hydrocarbons. The high-pressure post-injection contains enough kinetic energy to achieve an effective mixture formation and to entrain further oxygen into the partial oxidation zone. The combustion of this mixture increases the temperature during the late combustion phase and thus enhances the reduction of soot Fig. 6.8. The heat released by the post-injection is not sufficient to increase the NO_x emissions. Altogether, this kind of post-injection can result in a significant reduction of

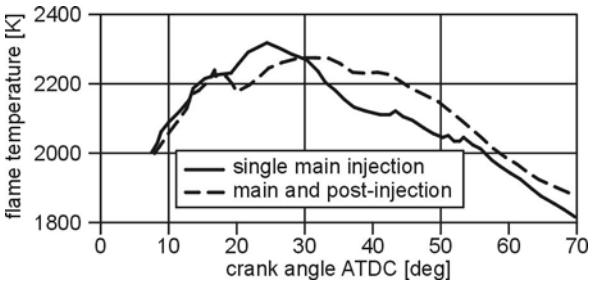


Fig. 6.8. Increase of temperature due to an early post-injection [12]

soot emissions (up to 70% reduction), while the fuel consumption is not affected (e.g. [21, 52]). The more soot produced by the main combustion, the more effective the post-injection.

6.2.3 Piezo Injectors

It has been shown in the previous sections that modern and future injection systems must be capable of performing multiple injections with variable injection timing and dwell times between two subsequent injections as well as variable injection pressures and extremely small pilot masses in order to reduce both, engine-out raw emissions and fuel consumption. The more flexible the injection system, the more accurate the optimization of the injection at each single operating point of the engine map can be. In addition to this, an extremely high reproducibility of injection timing and injected masses is necessary, especially in the case of short injection pulses. High needle speeds and a precise and complete needle closing are necessary in order to improve mixture formation and to avoid a deteriorated mixture formation and unintended post-injections at the end of each injection. Most of these demands can be fulfilled if piezo actuators instead of the slower and less precise solenoid valves are used in order to control the motion of the needle. The basic approach of the piezo technique is to use the elongation of a piezo quartz, which is proportional to the voltage impressed between two opposing surfaces, as the actuator. Piezo quartzes are extremely stiff, and the elongation is also extremely precise and fast. If the voltage is kept constant, a constant elongation can be realized. Because the elongation is proportional to the voltage, any elongation between zero and a maximum value can be precisely adjusted. Usually, a number of piezo quartzes are mechanically arranged in line (so-called piezo stack) in order to increase the resulting amplitude. In general, two main concepts are possible, the indirect and the direct needle control.

Fig. 6.9 shows the piezo-inline-injector from Bosch, where an indirect needle control is realized. The assembly consists of a piezo stack, a hydraulic coupling device, a control valve, and the nozzle module. The hydraulic coupling device amplifies the elongation of the piezo stack and transfers it to the control valve. In order to start the injection event, the needle of the control valve moves downwards and the pressure on top of the injection needle deceases because more fuel leaves

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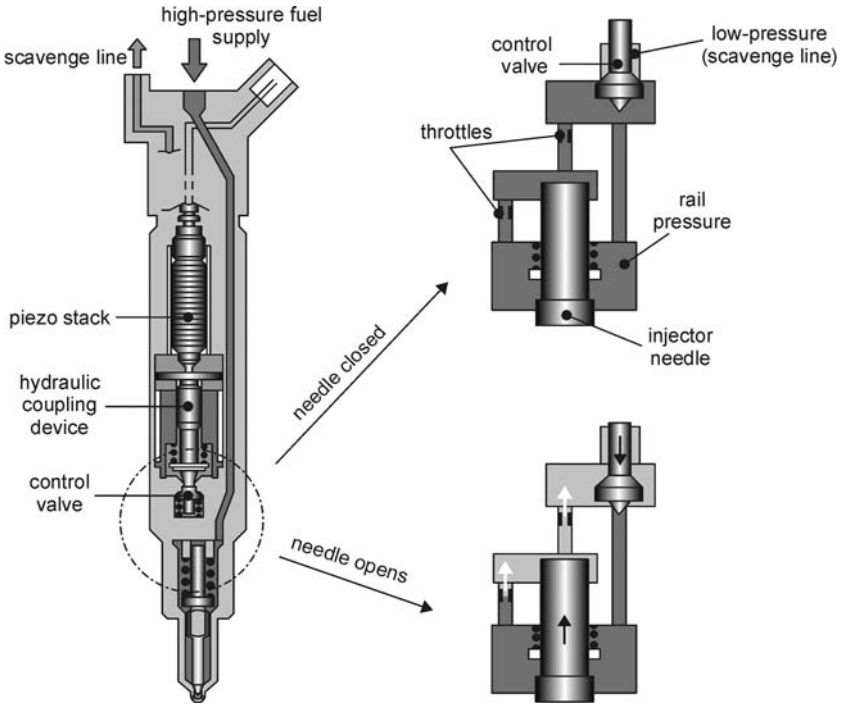


Fig. 6.9. Piezo in-line injector from Bosch [70]

the volume above this needle (larger cross-sectional area of the throttle) than enters from the high-pressure side (smaller throttle). At the end of injection the control valve closes and the increasing pressure above the injection needle pushes it downward again. Altogether, the stiff and close arrangement of the piezo actuator and needle, combined with small moving masses, allows the realization of an extremely reproducible, precise, and fast needle movement. Up to five injections per cycle with variable timings and extremely small injected fuel quantities can be performed [70].

The direct needle control offers even more flexibility, because well-defined partial needle lifts can be easily realized. Such a concept is shown in Fig. 6.10. The piezo stack is directly connected to the needle. This concept allows realizing any partial needle lift between zero and a maximum value, which is given by the maximum elongation of the stack. Because the needle lift is directly proportional to the voltage, it is possible to realize any kind of injection rate shape if the voltage can be controlled accordingly. This system has been designed as a research tool by the Institute of Technical Combustion at the University of Hanover in order to investigate the effect of different injection rate shapes on the combustion process of DI diesel engines (e.g. [54, 78, 75, 81]). The direct needle control has also turned out to be very effective in the case of pulsed injection in HCCI applications [66] (Sect. 6.4). In the case of ramp- or boot-type injections, the rate is controlled by needle seat throttling.

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EXHIBIT 1019

Günter P. Merker
Christian Schwarz
Rüdiger Teichmann
Editors

Combustion Engines Development

Mixture Formation, Combustion, Emissions and Simulation



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temperature changes at the pressure sensor and installation location alone. Thermal drifts represent the decisive measurement error in many measurements.

Besides the design of the pressure sensor, thermal drift is determined by the range and rate of the pressure sensor temperature change, which is highly contingent on the installation details of the pressure sensor mounting.

Two thermal drift phenomena are observable during pressure indications of internal combustion engines that must be clearly differentiated:

- Cyclical temperature drift
- Load change drift

Cyclical Temperature Drift

This concerns the faulty pressure indication within one cycle arising as a result of the cyclical heating of the pressure sensor. Cyclical temperature drift is also called short-term temperature drift or thermoshock.

Since cyclical temperature drift is active in many cases across a broad range of crank angles, its influence is considerable in the case of parameters that are integrated during one working cycle (e.g. the indicated mean effective pressure p_{mi}). Figure 3.13 shows the influence of a characteristic profile of the cyclical temperature drift on the determination of the indicated mean effective pressure and the energy balance (comparison of the fuel energy introduced per cylinder and working cycle with the energy converted in the combustion chamber calculated by the engine process computation under consideration of conversion losses). In comparison, a pressure-proportional error of the same scale – such as caused by a change in sensitivity for example – has a much smaller effect.

The characteristic values for describing cyclical temperature drift are determined on the one hand by means of a zero-pressure, cyclical heating of the sensor in a special test apparatus or on the other by means of comparative measurements in actual engine operation.

- Determination by zero-pressure, cyclical heating
  To this end, the test apparatus shown in Fig. 3.14 above (Glaser 1983) is employed, which makes it possible to subject the unstressed pressure sensor to a cyclical heating of similar size and frequency as arise in the engine. Opposite the sensor to be tested is a radiant heat surface which is covered and again uncovered by a rotating diaphragm wheel so that the heat flow profile also shown in Fig. 3.14 is generated. The radiant heat surface is heated electrically to temperatures of over 2,000°C, with which heat flows of more than 100 W/cm^2 can be produced. The pressure sensor is installed flushly in a solid cooled steel plate.
  The characteristic value used for the cyclical temperature drift is the maximum pressure deviation arising within one cycle at a certain frequency of the diaphragm wheel and at a certain radiant heating.

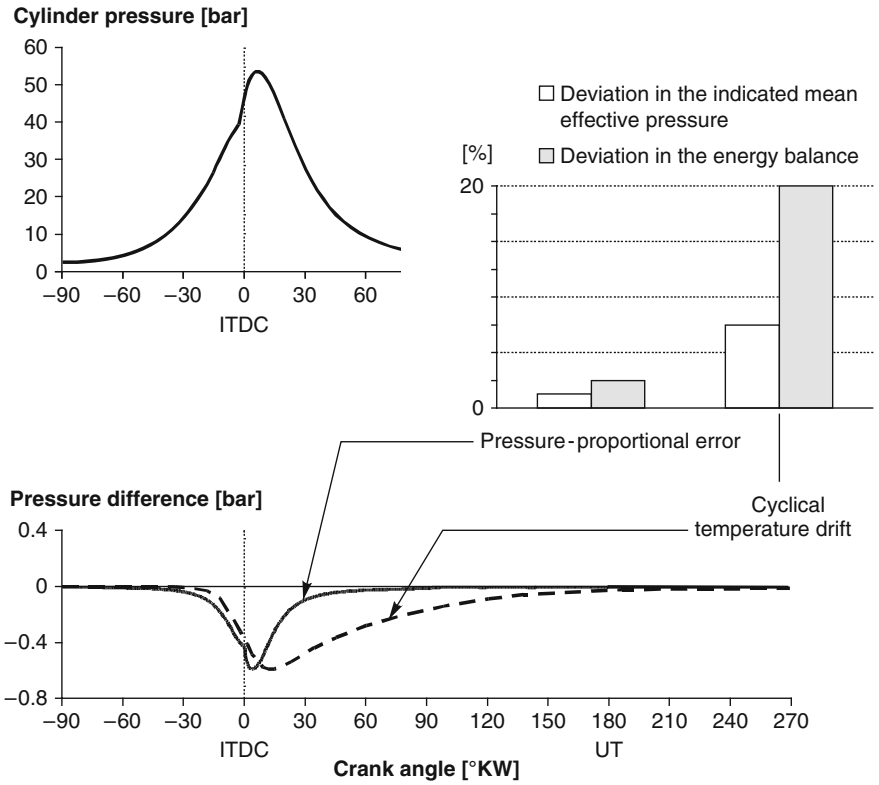


Fig. 3.13 Deviations in the indicated mean pressure and in the energy balance due to a pressure-proportionate error and cyclical temperature drift

This method is suitable for a basic evaluation of sensor attributes and for the preselection of sensors with especially low cyclical temperature drift. The results thereby obtained are not directly applicable to engine operation (i.e. radiant vs. convective heat flux), which is why the assessment of sensors should also be carried out during actual engine operation.

- Determination during engine operation with a reference pressure sensor
  In this case, a characteristic value for cyclical temperature drift is determined from the difference between the pressure profile of the sensor under investigation and that of a reference sensor. The reference sensor must satisfy very high demands for accuracy. As a rule, water-cooled sensors with special treatments to produce very low thermoshock-sensitivity are utilized. Also, the membranes of the reference sensor can be coated with silicon rubber, which significantly reduces the heat flow into the membranes caused by combustion. The maximum pressure deviation within a working cycle in a certain load state of the engine is given as the characteristic value for cyclical drift. Wimmer and Glaser (1996) provide an example of the use of this assessment method on uncooled miniature pressure sensors.

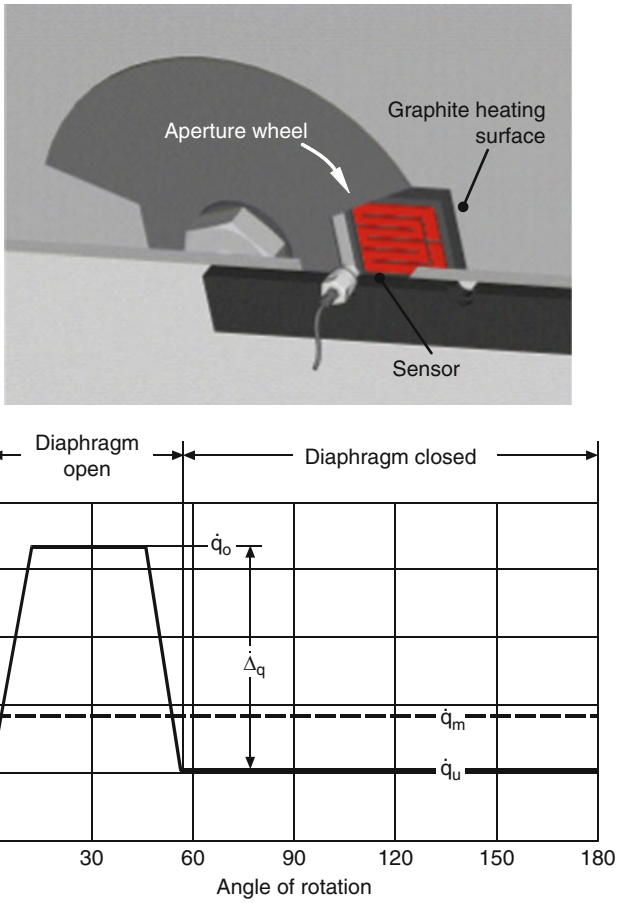


Fig. 3.14 Dynamic test apparatus to determine cyclical temperature drift

3.1.2.6 Measures to Reduce Cyclical Drift

Although the effect of cyclical temperature drift has been reduced considerably in the case of modern pressure sensors by means of special membrane and housing designs, it is still one of the most common sensor-related sources of error.

Manufacturers are aiming to optimize sensor design, in particular that of the membranes, such that the cyclical temperature drift is as low as possible. Presently, the use of the finite element method is particularly helpful in sensor design (Karst 2000). The prerequisite for this is, besides a realistic modeling, the exact knowledge of the marginal conditions – especially the knowledge of the heat flows arising on the side of the combustion chamber.

Besides selecting pressure sensors with as little cyclical temperature drift as possible, there are also some possibilities on the side of the operator. Since the

(surge limit in the case of the turbocharger), the pressure-wave charger is not subject to mass-production applications. The disadvantageous properties are as follows:

- Larger installation volume and weight than, for example, a turbocharger.
- Intake and exhaust pipes must be dimensioned larger because of their sensitivity to flow losses.
- When powered by a belt drive: arrangement in a direction parallel to the engine's longitudinal axis.
- Crankcase ventilation cannot be conducted before the pressure-wave charger (blockage of the cells by oil residues).
- Both on the intake and exhaust side, there is a very high-frequency noise (>500 Hz), which is difficult to muffle due to the low-pressure sensitivity.
- Compensation volume is necessary on the hot gas high-pressure side in order to reduce the vibrations caused by the exhaust impacts.
- Higher charge air temperatures due to heat exchange in the cells as the scavenge air sinks (calibration).
- High amount of scavenge air lowers the volumetric efficiency of the engine, especially at higher speeds.
- Exhibits weaknesses at start-up.
- Limitations of the power range for vehicle applications by the installation size.

Calculation of the pressure-wave charger in the context of overall process simulation can be achieved as a "pipe bundle" with variable reflection conditions (open and closed tube end) with the help of the one-dimensional gas dynamics described in Sect. 7.4. Since the calculation is very complex and because pressure-wave chargers are not mass-produced, we will dispense with a detailed description here.

8.1.2 Mechanical Supercharging

In the case of mechanical supercharging, a charging aggregate mounted on the engine is usually powered by a belt or gearwheel drive, which is why the engine and compressor speeds are usually rigidly coupled (Fig. 8.5). There is also electromechanical coupling, in which the charger is powered electrically. In the case of a rigid coupling between the internal combustion engine and the compressor, a compressor by pass is necessary to control the excess boost pressure.

In principle, charging aggregates can involve both turbo compressors (see Sect. 8.1.3) and positive displacement chargers.

8.1.2.1 Designs

The displacement chargers suitable for internal combustion engines can be subdivided into different designs that work either with or without internal compression. The speeds of positive displacement chargers are usually about double the engine

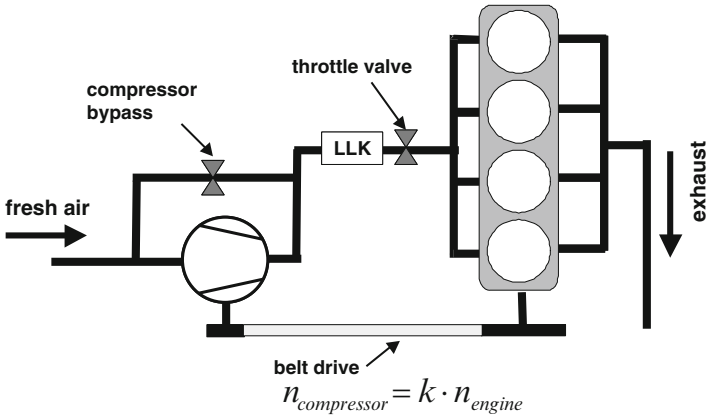


Fig. 8.5 Mechanical supercharging: block diagram

speed. Turbo compressors require a very high transmission ratio, since they are designed for much higher speeds (>100,000 rpm).

The following lists different designs of mechanical supercharging aggregates:

- Oscillating motion
  - Piston compressors (extra aggregate or crankcase for two-stroke engines, internal compression)
- Rotating motion
  - Hier fehlt Strömungsverdichter (siehe Kap. 8.1.3)
  - Twin-cam rotary piston compressors
    - Rotation piston compressors (e.g. roots compressor, no internal compression; internal-axis rotary piston charger, internal compression/expansion depending on the control edge)
    - Screw-type compressors (internal compression)
- Forced rotating compression
  - Spiral chargers (e.g. G-chargers, without internal compression)

Despite the variety of designs of positive displacement compressors, there are some basic common denominators:

- Positive displacement chargers always work between two fixed volume limits.
- Due to their discontinuous functioning, acoustical effects occur in case of an abrupt opening of the conveyor room against the volume lying behind the charging aggregate.
- The volume flow increases approximately in proportion with the speed.
- The performance map is stable in the lower speed and high pressure range despite somewhat increased leakage.
- All designs result in the same characteristic profile of lines of constant speed in the pressure ratio/volume flow map.