

Changing the Oil

It is only necessary to drain the oil from the crankcase, other than for regular oil changes, if it has become contaminated with water or dirt. In this case, you can drain the oil from the generator according to the following steps:

1. Place an approved oil disposal container underneath the generator to catch the oil as it drains.
2. Using a 10 mm hex wrench, unscrew the oil drain plug, which is located on the crankcase underneath the oil filler/dipstick cap (see figure 16). Allow all the oil to drain from the generator.
3. Replace the oil drain plug and tighten with a 10 mm hex wrench.

To add oil to the crankcase, follow these steps:

1. Make sure the generator is on a level surface.
2. Unscrew the oil filler/dipstick cap from the engine as shown in figure 14 above.
3. Using a funnel, add motor oil to the crankcase. We recommend SAE 10W30 motor oil for general use. When full, the oil level should come close in the top of the oil fill opening. (See figure 17).

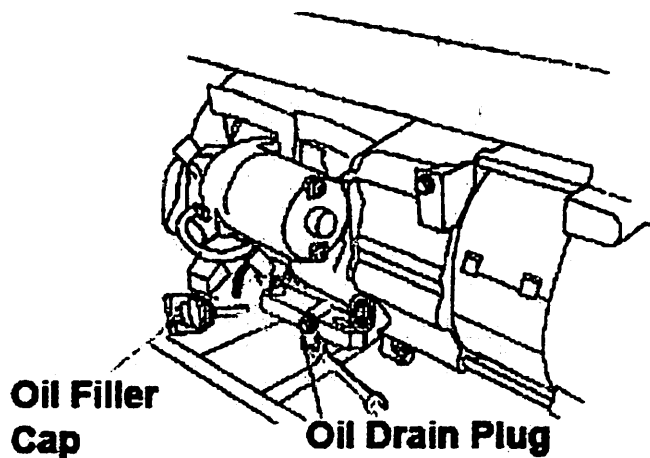


Figure 16: Draining the Oil



Figure 17: Adding Oil

Air Cleaner Maintenance

Routine maintenance of the air cleaner helps maintain proper airflow to the carburetor. Occasionally check that the air cleaner is free of excessive dirt.

1. Unhinge the clasps at the top and bottom of the air cleaner cover (see figure 18).
2. Remove the sponge-like elements from the casing.
3. Wipe the dirt from inside the empty air cleaner casing.
4. Wash the sponge-like elements in household dish detergent and warm water.
5. Allow the elements to dry completely.
6. Soak the dry elements in a small amount of engine oil. Ring out any excess oil.
7. Replace the sponge-like elements in the air cleaner casing and replace the cover.

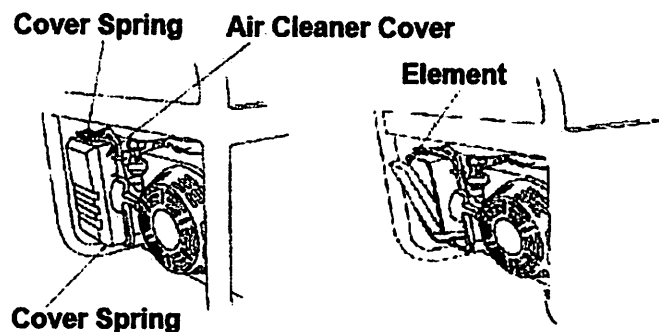


Figure 18 - Removing the Air Cleaner Casing.

Fuel Filter Cup Cleaning

The fuel filter cup is a small well underneath the fuel valve. It helps to trap dirt and water that may be in your fuel tank before it can enter the engine.

To clean the fuel filter cup:

1. Turn the fuel valve to the "OFF" position.
2. Unscrew the fuel filter cup from the fuel valve using a wrench. Turn the valve towards you to unscrew (see figure 19).
3. Clean the cup of all sediment using a rag or brush.
4. Reinstall the fuel filter cup.

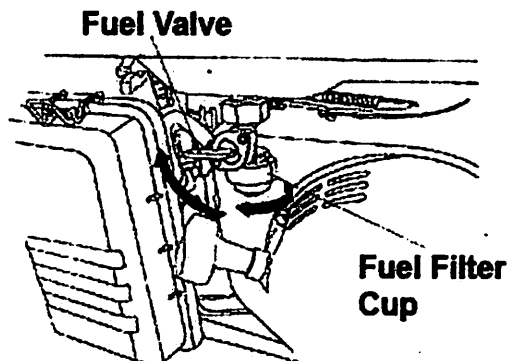


Figure 19 – Removing the Fuel Filter Cup

Spark Plug Maintenance

The spark plug is important for proper engine operation. A good spark plug should be intact, free of deposits, and properly gapped.

To inspect your spark plug:

1. Pull on the spark plug cap to remove it.
2. Unscrew the spark plug from the generator using the spark plug wrench included with this product (see figure 20).
3. Visually inspect the spark plug. If it is cracked or chipped, discard and replace with a new spark plug. We recommend using a F6RTC spark plug such as NGK BPR5ES.
4. Measure the plug gap with a gauge (see figure 21). The gap should be 0.7-0.8 mm (0.028-0.031 in).
5. If you are re-using the spark plug, use a wire brush to clean any dirt from around the spark plug base and then re-gap the spark plug.
6. Screw the spark plug back into its place on the generator using the spark plug wrench.
7. Replace the spark plug cap.

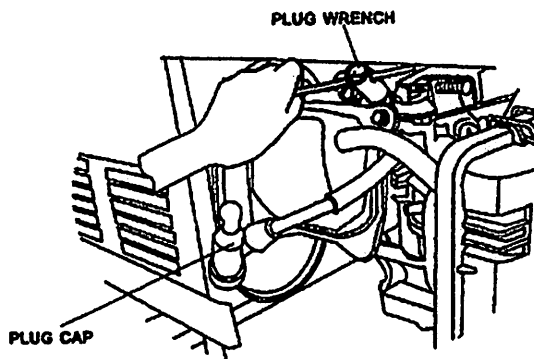


Figure 20- Removing the spark plug

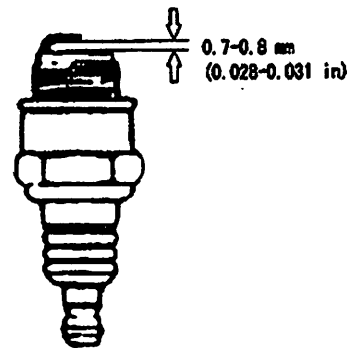


Figure 21- Measuring the spark plug gap

Emptying the Gas Tank

Before storing your generator for extended periods of time, you should drain your generator of gasoline.

To drain the generator of gas:

1. Turn the fuel valve to the "OFF" position.
2. Remove the fuel filter cup (see "Removing the Fuel Filter Cup" earlier in this section).
3. Empty the fuel filter cup of any fuel.
4. With a receptacle underneath the generator to catch the gas, turn the fuel valve to the "ON" position. Drain all the gas from the generator.
5. Turn the fuel valve to the "OFF" position.
6. Replace the fuel filter cup.
7. Store the emptied gasoline in a suitable place.

⚠ CAUTION: Do not store fuel from one season to another.

Storage and Transportation

⚠ CAUTION: Never place any type of storage cover on the generator while it is still hot.

When transporting your generator:

- Empty the gas tank (see "Emptying the Gas Tank" in the "Maintenance" section).
- Disconnect the spark plug.
- Do not obstruct any ventilation openings.
- Keep the generator in a cool dry area.

When storing your generator:

If you plan on starting the unit again the same day:

1. Turn off the main breaker.
2. Allow the unit to run 3 - 5 minutes.
3. Turn off the key.
4. Store.

If you plan on starting the unit again within 30 days:

1. Turn off the main breaker.
2. Allow the unit to run 3 - 5 minutes.
3. Turn off the fuel valve.
4. Allow the unit to stall out.
5. Turn off the key.
6. Add fuel stabilizer to the gas remaining in the tank.
7. Store.

If you do not plan to start the unit for longer than 30 days:

1. Turn off the main breaker.
2. Allow the unit to run 3 - 5 minutes.
3. Turn off the fuel valve.
4. Allow the unit to stall out.
5. Turn off the key.
6. Drain the fuel tank (See "Emptying the Gas Tank" in the "Maintenance" section)
7. Drain the carburetor
 - a. Remove the drain bolt from the carburetor.
 - b. Drain the small amount of remaining fuel from the carburetor bowl.
8. Oil the cylinder
 - a. Remove the spark plug.
 - b. Put 2 tbsp. of 10w30 motor oil directly into the spark plug hole
 - c. Pull the recoil start one time.
 - d. Replace the plug.
9. Remove the battery and place on tender indoors.

Generator Specifications		
AC Rated Wattage	3500W	
AC Surge Wattage	4400W	
AC Rated Voltage	120/240V	
AC Rated Frequency	60 Hz	
AC Phase	Single	
DC Voltage	12V	
DC Amperage	8.3A	
Dimensions (in.)	Length	23.2
	Width	17
	Height	17
Engine Type	4-Stroke OHV Forced-Air	
Ignition System	Non-Contact Transistor	
Displacement	210cc	
Starting Type	Electric	
Fuel Tank Capacity	3.96 US Gal. (15L)	
Oil Capacity	20 fl. oz. (0.6L)	
Run Time @ 50% (Gasoline)	12 hr.	
Run Time @ 50% (Propane)	16 hr. (5 Gallon)	
Noise Level	<69db	

Troubleshooting		
Problem	Cause	Solution
Engine will not start	Engine Switch is "Off"	Set Engine Switch to "Run"
	Fuel Valve is "Closed"	Turn Fuel Valve to "Open"
	Choke is open.	Close the Choke
	Engine is out of fuel.	Add Fuel
	Fuel is old or contaminated.	Change Fuel
	Spark Plug is dirty.	Clean Spark Plug
	Spark Plug is broken.	Replace Spark Plug
	Generator is not level.	Move generator to a level surface
	Oil is low.	Add/Change oil
Engine runs, but there is no electrical output.	Circuit Breaker is "Off"	Turn "On" Circuit Breaker
	Wiring connection is bad.	Replace Extension Cord(s)
	Device connected to generator is Malfunctioning.	Disconnect Malfunctioning Device
Generator runs, but does not support all electrical devices connected.	Generator is overloaded.	Disconnect 1 or more items to reduce the load
	Device connected to generator is bad.	Disconnect Malfunctioning Device
	Air Cleaner is Dirty.	Clean/Replace the Air Filter

Wheel Kit Assembly

The generator includes a wheel kit for easy transportation.

To install the wheel kit:

1. Please the generator on a flat, even surface. Temporarily place blocks under the generator to easy assembly.
- NOTE: DO NOT flip the generator upside down to install the wheel kit.**
2. Secure the support legs to the frame with provided bolts and lock nuts.
(See figure 22)
 3. Secure the axle bracket welded to axle the frame with the provided bolts and lock nuts.
(See figure 23)
 4. Secure the loose axle bracket to the frame with the provided bolts and lock nuts.
(See figure 24)
 5. Slide one wheel over each axle end and secure with the provided retaining pins.
(See Figures 25)
 6. Attach the handles to the brackets on the frame using the provided bolts and nuts.
(See Figures 26 and 27)
 7. Check that all fasteners are tight.

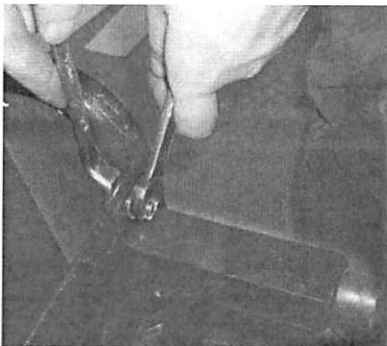


Figure 22



Figure 23



Figure 24



Figure 25



Figure 26

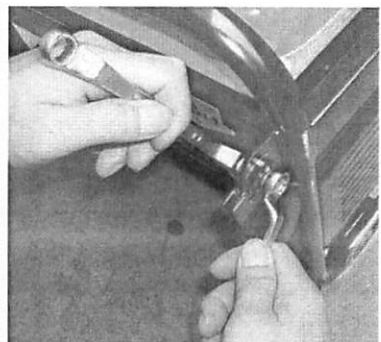
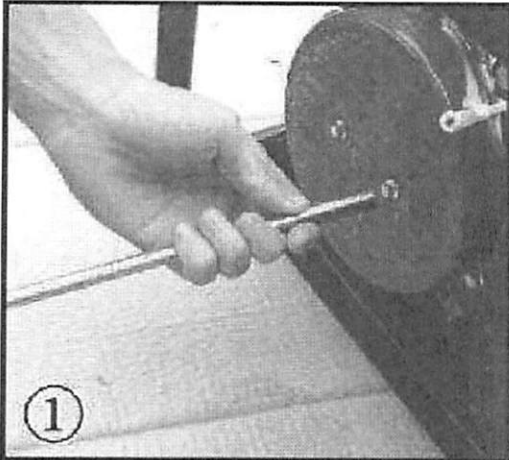
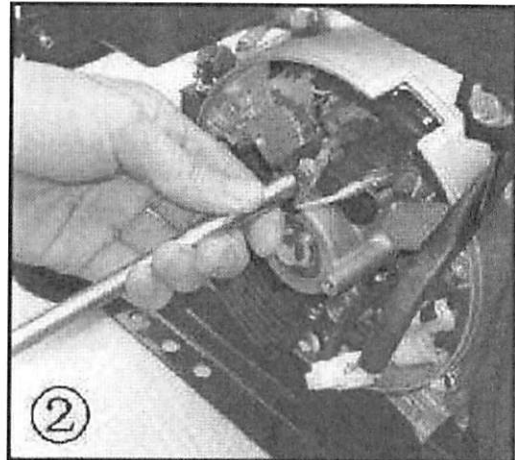


Figure 27

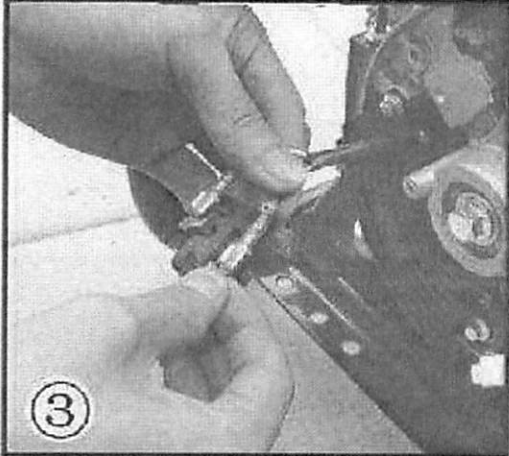
Changing/Inspecting the Carbon Brushes



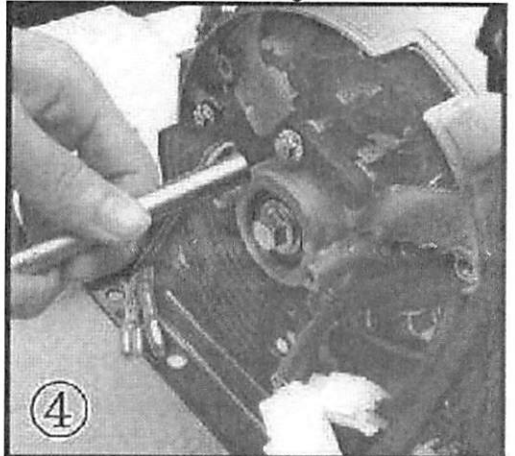
① Remove the 2 bolts on the generator cover.



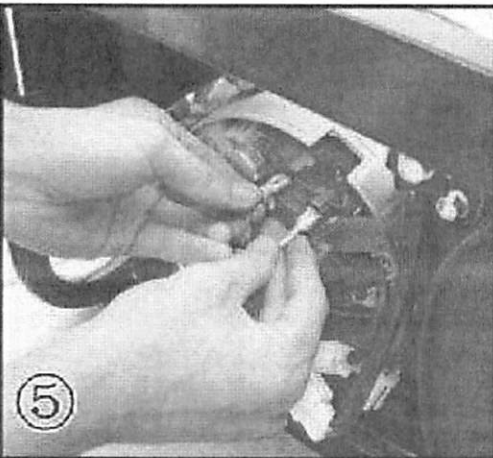
② Remove the bolt holding the carbon brush.



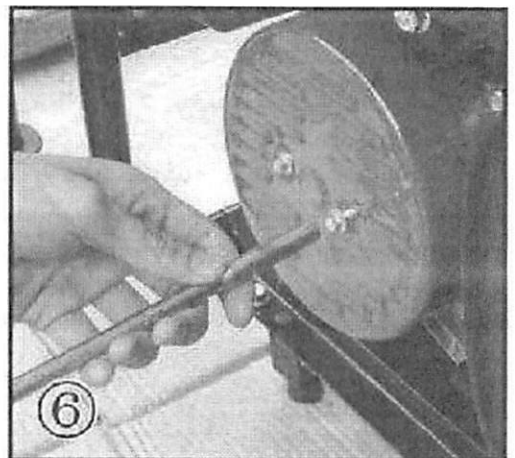
③ Remove the 2 wires from the AVR on the carbon brush.



④ Install a new carbon brush with bolt.

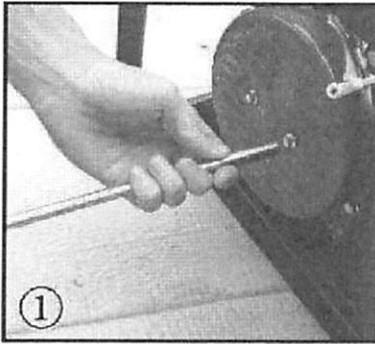


⑤ Insert and connect the 2 wires from the AVR, be sure to connect + and - correctly.

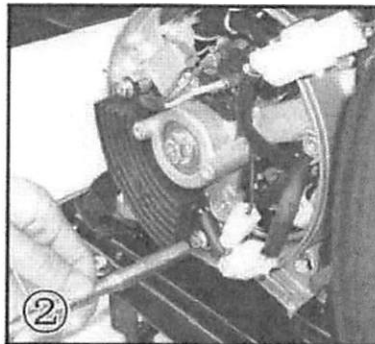


⑥ Replace the back cover of the generator and secure with the 2 bolts.

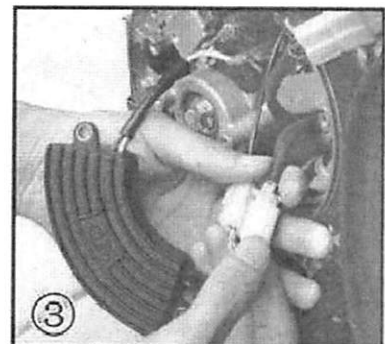
Changing/Inspecting the AVR



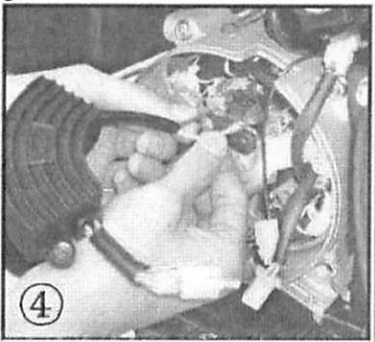
①
Remove the 2 bolts on the generator cover.



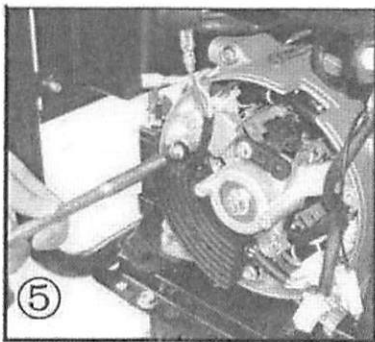
②
Remove the 2 bolts holding the AVR.



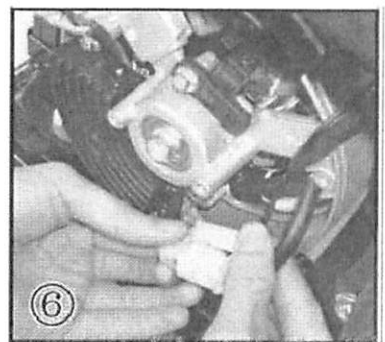
③
Disconnect the wire clip.



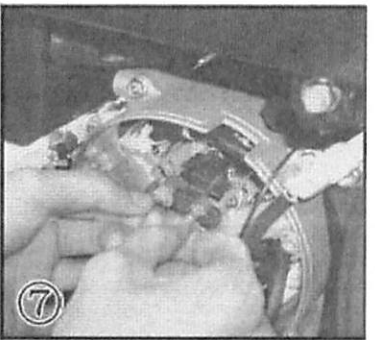
④
Remove the 2 wires from the AVR on the carbon brush.



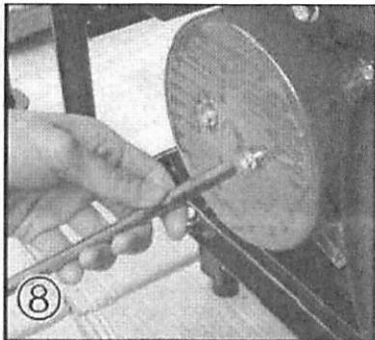
⑤
Install the new AVR with the 2 bolts.



⑥
Reconnect the wire clip.



⑦
Insert and connect the 2 wires from the AVR, be sure to connect + and - correctly

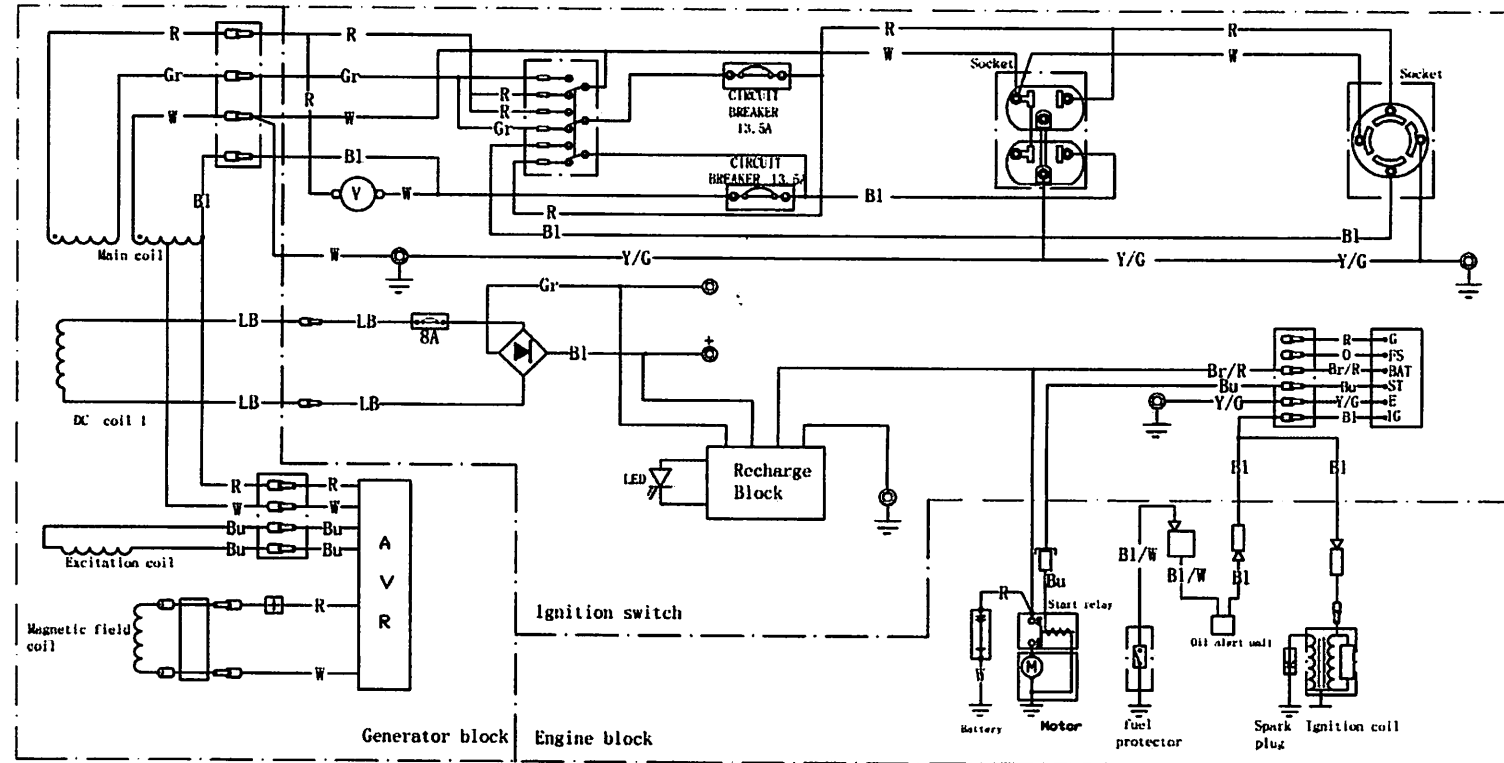


⑧
Replace the back cover of the generator and secure with the 2 bolts.

Maintenance Log

[illegible]

Wiring Diagram



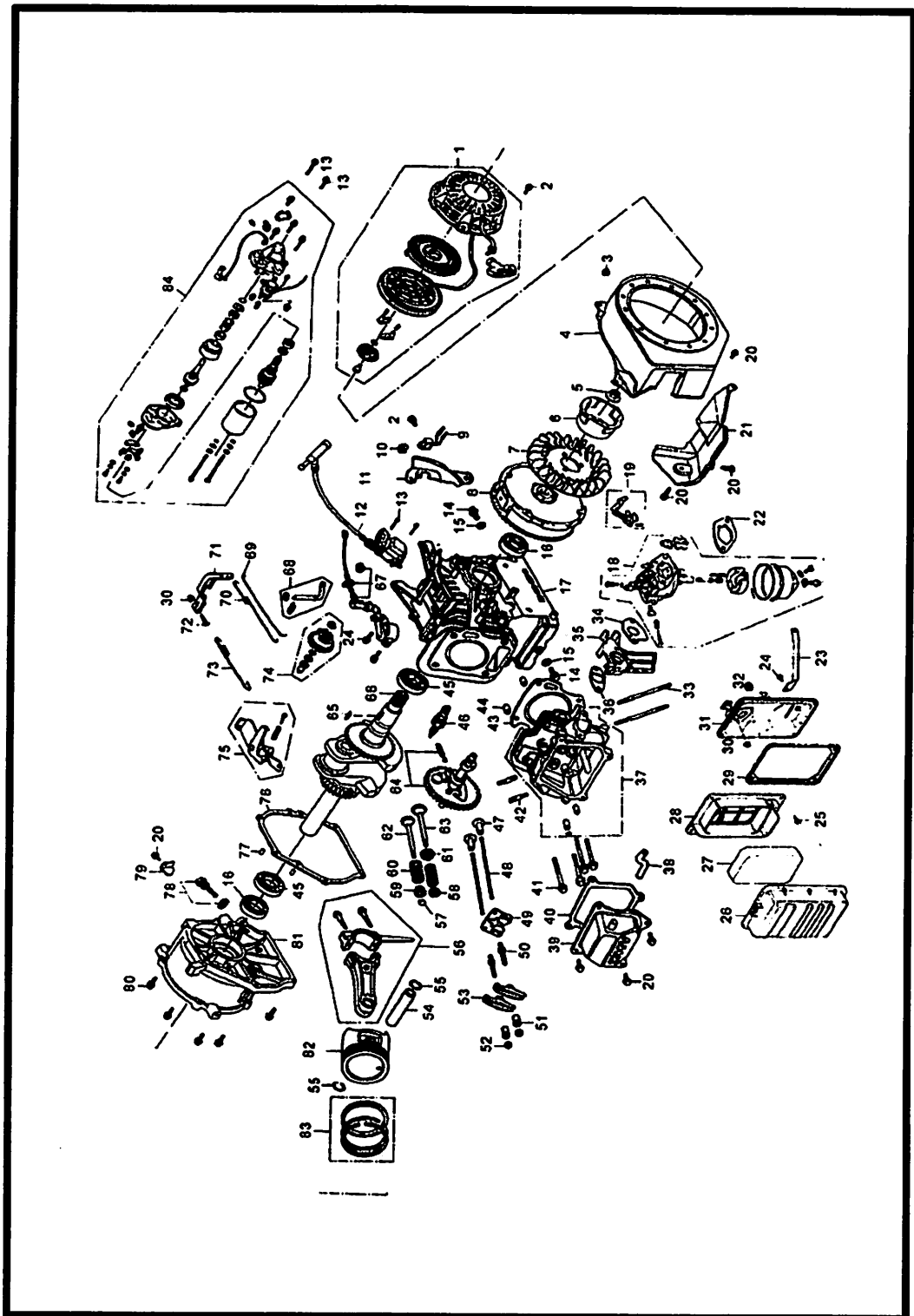
B1	Black	Br	Brown
O	Orange	Br/R	Brown/Red
Bu	Blue	LB	Light blue
Gr	Gray	Y/G	Yellow/Green
R	Red	W	White

Ignition switch connecting

	IG	E	BAT	ST	FS	G
OFF	○				○	
ON						
Start			○			

WIRING DIAGRAM OF 120V/240V GENERATOR SET

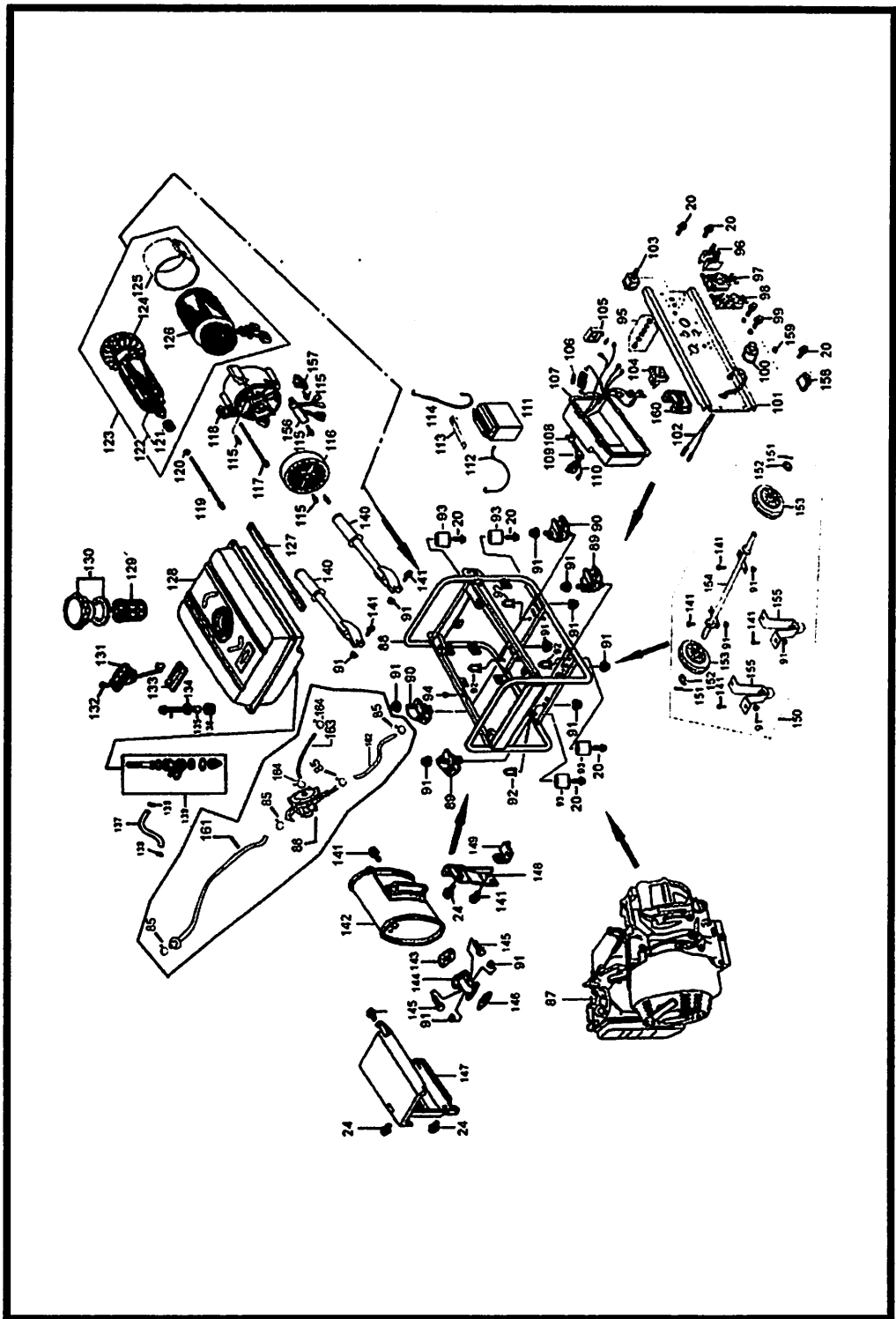
Engine Parts



Part #	Description	Qty	Item #	Cross-Reference
1	Recoil Starter (Complete)	1	XP4400EH-1	DJ168F-16121-K
2	Flange Bolt (M6x8)	4	XP4400EH-2	GBT5787-B6-8
3	Grommet, Drain Hole	1	XP4400EH-3	DJ168F-16118
4	Flywheel Fan Cover (Comp)	1	XP4400EH-4	DJ170N-16121-A
5	Flange Nut (M14)	1	XP4400EH-5	GBT8177/10-N-14
6	Starter Pulley	1	XP4400EH-6	DJ168F-16000-A
7	Flywheel Fan Cover (Comp)	1	XP4400EH-7	DJ168F-16002-A
8	Flywheel	1	XP4400EH-8	DJ168F-16200-I
9	Low Oil Switch	1	XP4400EH-9	DJ168F-18300-A
10	Low Oil Switch Clip	1	XP4400EH-10	XP4400EH-10
11	Side Plate	1	XP4400EH-11	DJ170S-11400-A
12	Ignition Coil Assy	1	XP4400EH-12	DJ168F-18100-C
13	Flange Bolt (M6x25)	2	XP4400EH-13	GBT5787-B6-25
14	Drain Lug Bolt	2	XP4400EH-14	DJ168F-11004-A
15	Drain Lug Washer	2	XP4400EH-15	DJ168F-11005-A
16	Oil Seal (25x41x6)	2	XP4400EH-16	DJ168F-11014-A
17	Crankcase Assy	1	XP4400EH-17	DJ170F-11100-C
18	LPG/Gasoline Carburetor Assy	1	XP4400EH-18	DJ170FD-14100-C
19	Carburetor Choke Lever	1	XP4400EH-19	DJ168F-14119
20	Flange Bolt (M6x12)	11	XP4400EH-20	GBT5787-B6-16
21	Shroud Complete	1	XP4400EH-21	DJ168F-11012-A
22	Gasket, Carb to Air	1	XP4400EH-22	DJ168F-14027-A
23	Air Cleaner Bracket	1	XP4400EH-23	DF2500H-14205-A
24	Flange Bolt (M6x12)	3	XP4400EH-24	GBT5787-B6-12
25	Bolt (M5x12)	4	XP4400EH-25	GBT5787-B5-14
26	Air Cleaner Cover	1	XP4400EH-26	DJ168F-14204
27	Air Cleaner Element	1	XP4400EH-27	DJ170F-14201
28	Air Cleaner Element Holder	1	XP4400EH-28	DJ168F-14202
29	Air Cleaner Seal	1	XP4400EH-29	DJ168F-14206
30	Flange Nut (M6)	2	XP4400EH-30	GBT6177-N-6
31	Air Cleaner Case (Comp)	1	XP4400EH-31	DJ168F-14205-A
32	Grommet, Fender	1	XP4400EH-32	DJ168F-14014-A
33	Guide Bolt (M6x94)	2	XP4400EH-33	DJ168FD-14001-A
34	Carburetor Gasket	1	XP4400EH-34	DJ168F-14005-A
35	Carburetor Heat Isolator	1	XP4400EH-35	DJ168F-14004-A
36	Carburetor Heat Isolator Gasket	1	XP4400EH-36	DJ168F-14003-B
37	Cylinder Head (Comp)	1	XP4400EH-37	DJ170F-11200-B
38	Cylinder Head Breather	1	XP4400EH-38	DJ168FD-11013-A
39	Cylinder Head Cover (Comp)	1	XP4400EH-39	DJ168F-11300-C
40	Cylinder Head Cover Gasket	1	XP4400EH-40	DJ168F-11011-A
41	Flange Bolt (M8x60)	4	XP4400EH-41	GBT5787-B8-60
42	Guide Bolt (M8x32)	2	XP4400EH-42	DJ168F-14002-A

Part #	Description	Qty	Item #	Cross-Reference
43	Cylinder Head Gasket	1	XP4400EH-43	DJ170F-11010-C
44	Cylinder Head Bolt Guide	2	XP4400EH-44	DJ168F-11009-A
45	Radial Ball Bearing (6025)	2	XP4400EH-45	GB276-89-6205
46	Spark Plug	1	XP4400EH-46	DJ168F-18500-A
47	Valve Tappet	2	XP4400EH-47	DJ170F-13009-A
48	Push Rod	2	XP4400EH-48	DJ170F-13008-A
49	Push Rod Guide Plate	1	XP4400EH-49	DJ168F-13300-A
50	Rocker Arm Pivot Bolt	2	XP4400EH-50	DJ168F-13202-A
51	Rocker Arm Nut	2	XP4400EH-51	DJ168F-13203-A
52	Rocker Arm Pivot Bolt Nut	2	XP4400EH-52	DJ168F-13204-A
53	Rocker Arm	2	XP4400EH-53	DJ168F-13201-A
54	Piston Pin	1	XP4400EH-54	DJ170F-12002-A
55	Piston Pin Clip	2	XP4400EH-55	DJ168F-12003-A
56	Connecting Rod	1	XP4400EH-56	DJ170F-12200-B
57	Valve Cap	1	XP4400EH-57	DJ168F-13006-A
58	Intake Valve Spring Retainer	1	XP4400EH-58	DJ168F-13004-A
59	Exhaust Valve Spring Retainer	1	XP4400EH-59	DJ168F-13005-A
60	Valve Spring	2	XP4400EH-60	DJ168F-13003-A
61	Valve Spring Seat	1	XP4400EH-61	DJ168F-13010-A
62	Exhaust Valve	1	XP4400EH-62	DJ168F-13001-A
63	Intake Valve	1	XP4400EH-63	DJ168F-13002-A
64	Camshaft	1	XP4400EH-64	DJ168F-13100-B
65	Crankshaft Key (25x18)	1	XP4400EH-65	DJ168F-12104
66	Crankshaft	1	XP4400EH-66	DJ168F-12100-BB
67	Low Oil Sensor	1	XP4400EH-67	DJ168F-18200-A
68	Governor Shaft	1	XP4400EH-68	DJ168F-15001-A
69	Governor Arm Rod	1	XP4400EH-69	DJ168F-15006-A
70	Throttle Return Spring	1	XP4400EH-70	DJ170F-15008-A
71	Governor Arm	1	XP4400EH-71	DJ168F-15004-A
72	Governor Arm Bolt	1	XP4400EH-72	DJ168F-15005-A
73	Governor Spring	1	XP4400EH-73	DJ170F-15007-A
74	Governor Assembly (Comp)	1	XP4400EH-74	DJ168F-15100-A
75	Throttle Stop Assy	1	XP4400EH-75	DJ168FD-15200-A
76	Crankcase Cover Gasket	1	XP4400EH-76	DJ170F-11003-B
77	Crankcase Bolt Guide (8x14)	2	XP4400EH-77	DJ168F-11002-F
78	Oil Fill Cap	1	XP4400EH-78	DJ168F-11007-A
79	Plate	1	XP4400EH-79	DF1800H-33016-A
80	Flange Bolt (M8x32)	6	XP4400EH-80	GBT5787-B8-35
81	Crankcase Cover	1	XP4400EH-81	DJ168FD-11001-C
82	Piston	1	XP4400EH-82	DJ170F-12001-B
83	Piston Ring Set	1	XP4400EH-83	DJ170F-12300-A
84	Starter Motor	1	XP4400EH-84	DJ168F-18600-A

Other Generator Parts



Other Generator Parts

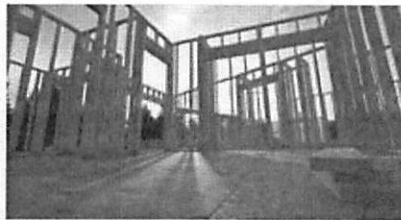
Part #	Description	Qty	Item #	Cross-Reference
85	Clip, Tube	4	XP4400EH-85	XP4400EH-85
86	Propane Regulator / Release Valve	1	XP4400EH-86	XP4400EH-86
87	Gasoline Engine Complete	1	XP4400EH-87	DJ170FD-8D
88	Generator Frame Complete	1	XP4400EH-88	XP4400E-31100-BK
89	Vibration Mount A	2	XP4400EH-89	DF2500H-31201-A
90	Vibration Mount B	2	XP4400EH-90	DF2500H-31202-A
91	Flange Nut (M8)	18	XP4400EH-91	GB16177-N-8
92	Flange Nut (M8)	4	XP4400EH-92	GB1889-N-8
93	Rubber Foot, Frame	4	XP4400EH-93	XP4400EH-93
94	Fuel Tank Rubber Pad	2	XP4400EH-94	DF2500H-14304-A
95	Earth Terminal (Complete)	1	XP4400EH-95	34107
96	Circuit Breaker (15A 240v)	1	XP4400EH-96	34206-100
97	Receptacle (30A)	1	XP4400EH-97	34204-003
98	Receptacle (20A, Duplex)	2	XP4400EH-98	34204-005
99	DC Output Post (Set)	2	XP4400EH-99	34204-038
100	Engine Switch (Key)	1	XP4400EH-100	34203-002
101	Control Panel Face, Metal Only	1	XP4400EH-101	**SEE REF #
102	Sub-Wiring Harness	1	XP4400EH-102	634213-002
103	Voltage Selection Switch	1	XP4400EH-103	34202-006
104	Voltmeter	1	XP4400EH-104	34205-001
105	Diode Assy	1	XP4400EH-105	34210-001
106	Fuse	1	XP4400EH-106	XP4400EH-106
107	Control Panel Rear Cover	1	XP4400EH-107	34102-A
108	Rubber Boot, Main Wiring Harness	1	XP4400EH-108	34109
109	Rubber Boot, Generator AC Out	1	XP4400EH-109	34117
110	Main Wiring Harness	1	XP4400EH-110	34212-001
111	Battery (7Ah 12V DC)	1	XP4400EH-111	31300-001
112	Positive Battery Cable	1	XP4400EH-112	31038-001
113	Battery Support Bar	1	XP4400EH-113	DF1800H-31033-A
114	Negative Battery Cable	1	XP4400EH-114	31040-001
115	Flange Bolt (M5x12)	2	XP4400EH-115	GB15789-B5-16
116	Generator End Cover	1	XP4400EH-116	DF1800H-33006-B
117	Flange Bolt (M6x125)	4	XP4400EH-117	DF3500H-33015-A
118	Generator End Support	1	XP4400EH-118	DF1800H-33005-A
119	Bolt (M8x180)	1	XP4400EH-119	DF1800H-33003-A
120	Plain Washer (10)	1	XP4400EH-120	GB193-LW-8
121	Bearing 6204-2RS	1	XP4400EH-121	GB276-89-6204
122	Rotor (Comp)	1	XP4400EH-122	DF3500H-33110-B
123	Stator and Rotor (Comp)	1	XP4400EH-123	DF3000H3103-B-23602
124	Generator Fan	1	XP4400EH-124	DF2500H-33023
125	Stator Cover	1	XP4400EH-125	DF3500H-33129
126	Stator (Comp)	1	XP4400EH-126	DF3500H-33120
127	Fuel Tank Rubber Pad	1	XP4400EH-127	DF2500H-14322-D
128	Fuel Tank	1	XP4400EH-128	DF2500H-14300-B
129	Fuel Tank Filler Cup	1	XP4400EH-129	14307
130	Fuel Tank Cap	1	XP4400EH-130	14306
131	Fuel Tank Sensor (Complete)	1	XP4400EH-131	14303
132	Screw (M5x10)	2	XP4400EH-132	GB1819-B-5
133	Fuel Tank Sensor Gasket	1	XP4400EH-133	14313
134	Washer	4	XP4400EH-134	14311
135	Collar	4	XP4400EH-135	14305
136	Cushion	4	XP4400EH-136	14304
137	Gas Tube (4.5x165)	1	XP4400EH-137	DF188FD-14007-C
138	Tube Clip	2	XP4400EH-138	DJ168F-14008-A
139	Fuel Valve	1	XP4400EH-139	DF2500H-14302-A
140	Handles	2	XP4400EH-140	DF3800H-31058-C
141	Flange Bolt (M8x16)	17	XP4400EH-141	GB1889-N-8X20
142	Muffler	1	XP4400EH-142	DH2500H-14400-E
143	Muffler Gasket	1	XP4400EH-143	DJ168F-14008-A
144	Exhaust Pipe	1	XP4400EH-144	DF2500H-14009
145	Flange Bolt (M8x25)	2	XP4400EH-145	GB5787-B-6
146	Exhaust Pipe Gasket	1	XP4400EH-146	DJ168F-14010-A
147	Muffler Heat Shield	1	XP4400EH-147	DF2500H-14018-B
148	Muffler Bracket	1	XP4400EH-148	XP4400E-149
149	Muffler Bracket Plate	1	XP4400EH-149	XP4400E-149
150	Wheel Kit (Assy)	1	XP4400EH-150	XP4400E-WK
151	Wheel Retaining Pin	2	XP4400EH-151	DF2500H-31019-B
153	Wheel, Semi-Solid 8"	2	XP4400EH-153	31017-F
154	Wheel Shaft	1	XP4400EH-154	DF3000H-31018-F
155	Wheel Shaft Bracket	2	XP4400EH-155	DF3000H-31070-E
156	Voltage Regulator	1	XP4400EH-156	DF2500H-33011-A
157	Brush (Assy)	1	XP4400EH-157	DF3500H-33001
158	Circuit Protector (10A, DC)	1	XP4400EH-158	34208-004
159	Recharge Indicator LED	1	XP4400EH-159	XP4400E-159
160	12V DC Regulator	1	XP4400EH-160	34224-001
161	Propane Hose w/ Collar	1	XP4400EH-161	XP4400E-161
162	Propane Hose to Carburetor A	1	XP4400EH-162	XP4400E-162
163	Propane Hose to Carburetor B	1	XP4400EH-163	XP4400E-163
164	Propane Hose Clip (8)	2	XP4400EH-164	XP4400E-164



Rugged. Durable. Dependable. Power When You Need It!



Recreation



Construction



Portable Power



Ice Storms



Hurricanes



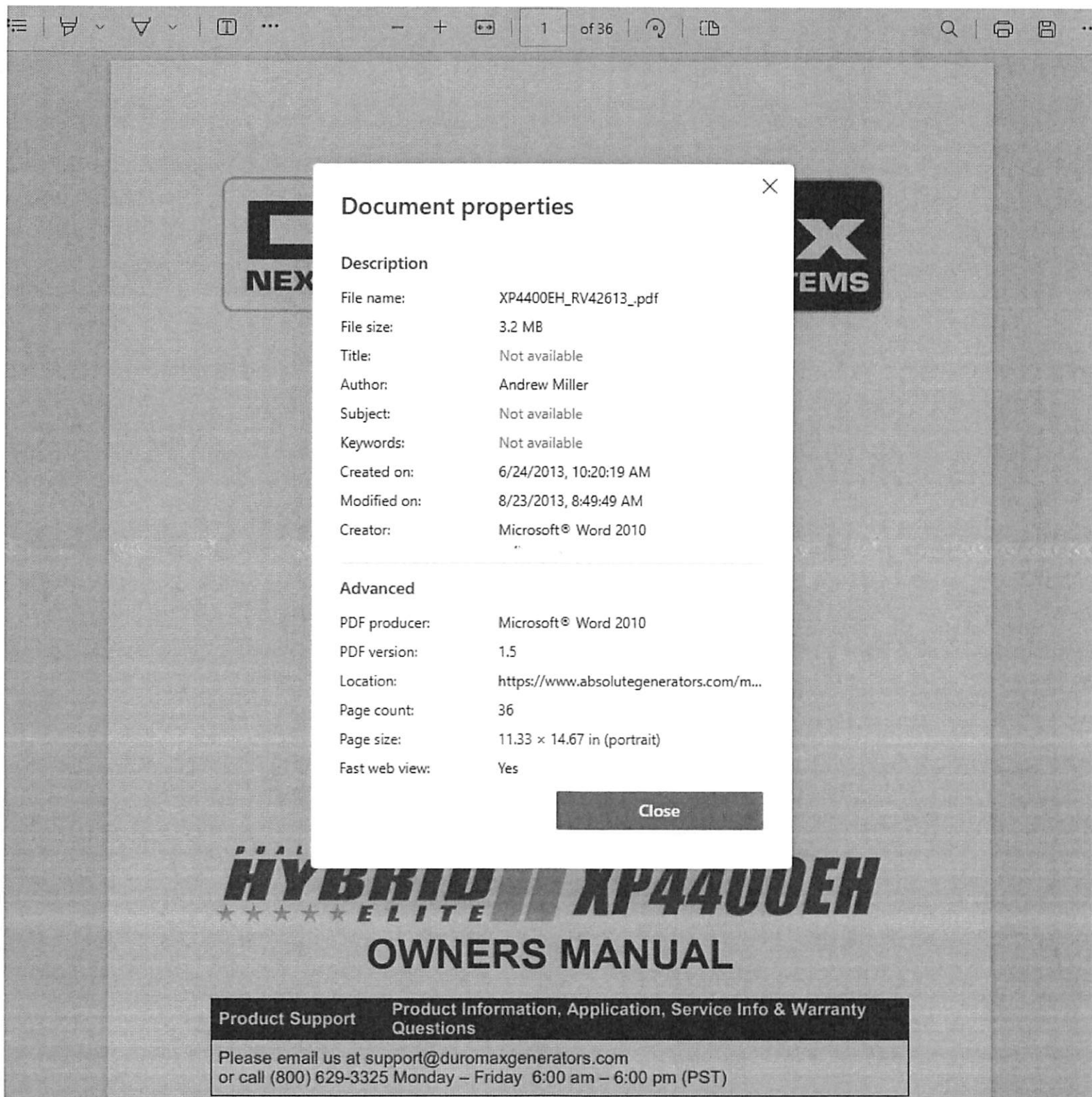
Electrical Storms

**Product
Support**

**Product Information, Application, Service Info & Warranty
Questions**

Please email us at support@duromaxgenerators.com
or call (800) 629-3325 Monday – Friday 6:00 am – 6:00 pm (PST)

EXHIBIT J



Document properties

Description

File name: XP4400EH_RV42613_.pdf
File size: 3.2 MB
Title: Not available
Author: Andrew Miller
Subject: Not available
Keywords: Not available
Created on: 6/24/2013, 10:20:19 AM
Modified on: 8/23/2013, 8:49:49 AM
Creator: Microsoft® Word 2010

Advanced

PDF producer: Microsoft® Word 2010
PDF version: 1.5
Location: <https://www.absoluteGenerators.com/m...>
Page count: 36
Page size: 11.33 × 14.67 in (portrait)
Fast web view: Yes

Close

Product Support

Product Information, Application, Service Info & Warranty Questions

Please email us at support@duromaxgenerators.com
or call (800) 629-3325 Monday – Friday 6:00 am – 6:00 pm (PST)

EXHIBIT 1019

Oct. 24, 1933.

G. L. HOLZAPFEL

1,931,698

FUEL SYSTEM FOR INTERNAL COMBUSTION ENGINES

Filed March 30, 1931

Fig. 1

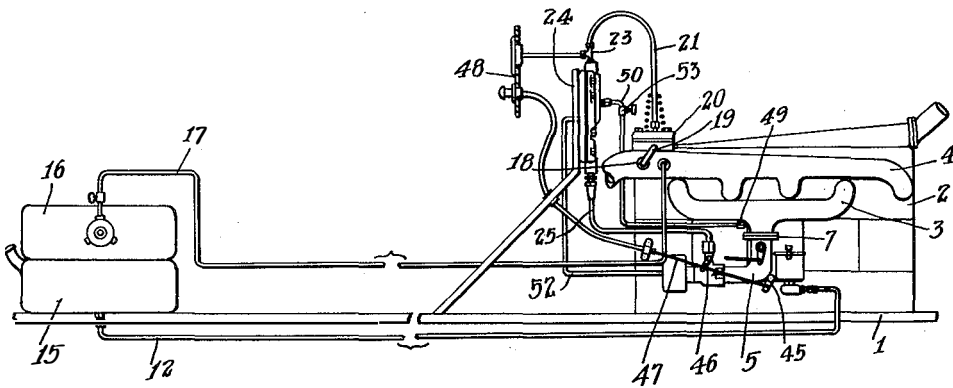
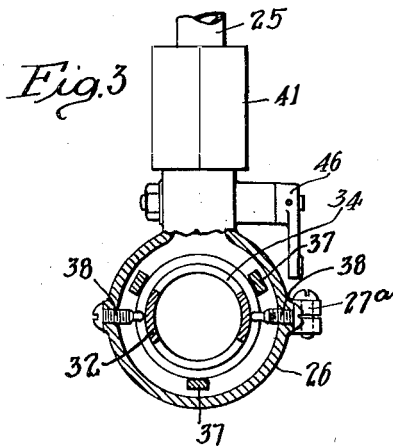
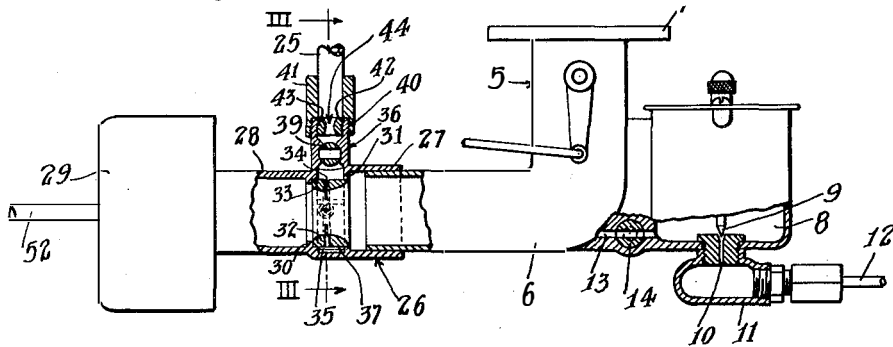


Fig. 2



Inventor.

George L. Holzappel

By Lyon & Lyon Attorneys

UNITED STATES PATENT OFFICE

1,931,698

FUEL SYSTEM FOR INTERNAL COMBUSTION
ENGINESGeorge L. Holzapfel, Palos Verdes Estates, Calif.,
assignor to Richfield Oil Company of California,
Los Angeles, Calif., a corporation of Delaware

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 60 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 70 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 75 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 80 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 85 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 90 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 95 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 100 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 110

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 carburetor 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 carburetor 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 carburetor tube 6 and may be clamped thereon as by means of clamping screw 27a. 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 carburetor 6. By reason of this reduced diameter the air cleaner 29, which normally is telescopically engaged upon the inlet end of the carburetor 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 thereto 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

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

5 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
10 provide a properly combustible mixture.

It will be observed that as the air is drawn through the venturi into the carburetor 5, the quantity of air so drawn will act on the venturi 32 to increase or decrease the quantity of gaseous
15 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
20 reducing valve 24. Thus at idling speeds a desirable quantity of fuel for idling may be furnished to the engine. The amount of fuel by-passed 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
25 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
30 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 carburetor comprising an air passage through which air may be passed into the intake
35

of the engine, a liquid fuel inlet communicating with the air passage of said carburetor for supplying liquid fuel to the air stream drawn through the carburetor, a cut off valve for cutting off the supply of liquid fuel to said carburetor, a gaseous
30 fuel inlet connected to and communicating with the air passage of said carburetor, 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
35 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
40 liquid or gaseous fuel to said carburetor.

2. In a fuel system for internal combustion engines having a carburetor, a source of gaseous fuel, means for introducing said gaseous fuel into said carburetor comprising a tube connectible to
45 the air inlet passage of said carburetor, 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
50 said gaseous fuel inlet for metering the quantity of fuel supplied therethrough.

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

4. An attachment for a liquid fuel carburetor having an air inlet nipple with an air cleaner slidably engageable therewith, which attachment
65 comprises a tube dimensioned at one end to engage with the air inlet of the carburetor normally engaging with the air cleaner, and dimensioned at the opposite end to engage with the end of the air cleaner normally engaging with
70 the carburetor inlet, whereby said tube may be inserted as a unit between the air inlet passage of the carburetor 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.

130

135

140

145

EXHIBIT 1020

The Wayback Machine - <https://web.archive.org/web/20130222055122/http://caloreequipment.com:80/products/cl/bbq-grill-low-pressure-regulators/two-stage-...>

Search entire store here...

[My Account](#) [My Wishlist](#) [My Cart](#) [Checkout](#) [Log In](#)
[Home](#) [About](#) [Products](#) [Media & Resources](#) [Customer Service](#) [Commercial Services](#) [Contact](#)
[Home](#) / [BBQ Grill Low Pressure Hose and Regulator](#) / **Two-Stage Low Pressure Hose & Regulator**

Two-Stage Low Pressure Hose & Regulator



2-Stage Low Pressure Regulator Assembly

\$70.00 SKU: 106238

Used for connecting commercial grade BBQ & Catering Grills requiring gas input in excess of 100,000 BTU's per hour to a propane tank via high volume green Type 1 tank connection. The green Type 1 allows a higher volume of gas to pass through the tank connection before going into bypass mode. *Standard Hose Length is 18" OAL [Learn More](#)

[Add to Wishlist](#) | [Add to Compare](#)


2-Stage Low Pressure Regulator Assembly with Stainless Steel Overbraid

\$90.00 SKU: 106221

Stainless Steel Overbraided Two-Stage Regulator Assembly used for connecting commercial grade BBQ & Catering Grills requiring gas input in excess of 100,000 BTU's per hour to a propane tank via high volume green Type 1 tank connection. The green Type 1 allows a higher volume of gas to pass through the tank connection before going into bypass mode. The overbraid keeps small animals from damaging the hose. *Standard Hose Length is 18" OAL [Learn More](#)

[Add to Wishlist](#) | [Add to Compare](#)


Type1 Cylinder Connection (Light Green)

\$12.00 SKU: 204039

The light green Type 1 is used to connect an outside threaded LP cylinder to high pressure appliances such as portable BBQ grills, camping stoves, and turkey fryers. Eliminates the need for wrenches and features a thermal shut-off and excess flow device. Max output is 200,000 BTU/hr. MB Sturgis has designed the green Type 1 to provide the highest flow rate possible before going into bypass mode. The light green Type 1 is also designed to allow the hose to depressurize itself when the hose is disconnected from the appliance allowing the pent up gas to escape rather than storing a pressurized LP gas hose. [Learn More](#)

[Add to Wishlist](#) | [Add to Compare](#)

COMPARE PRODUCTS

You have no items to compare.

MY CART

You have no items in your shopping cart.

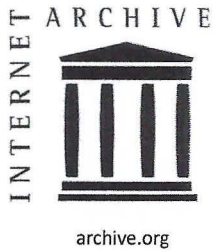
RECENTLY VIEWED PRODUCTS

[Sturqi-Stay Kit](#)[Sturqi-Stay Kit Deluxe](#)[Quick Disconnect Sturqi-Stay Deluxe Kit](#)[Quick Disconnect Sturqi-Stay Fitting](#)[Model 250 Male Plug X 1/8" MNPT](#)

AFFIDAVIT OF TANYA ZEIF

1. I am a Records Request Processor at the Internet Archive. I make this declaration of my own personal knowledge.
2. The Internet Archive is a website that provides access to a digital library of Internet sites and other cultural artifacts in digital form. Like a paper library, we provide free access to researchers, historians, scholars, and the general public. The Internet Archive has partnered with and receives support from various institutions, including the Library of Congress.
3. The Internet Archive has created a service known as the Wayback Machine. The Wayback Machine makes it possible to browse more than 450 billion pages stored in the Internet Archive's web archive. Visitors to the Wayback Machine can search archives by URL (i.e., a website address). If archived records for a URL are available, the visitor will be presented with a display of available dates. The visitor may select one of those dates, and begin browsing an archived version of the Web. Links on archived files in the Wayback Machine point to other archived files (whether HTML pages or other file types), if any are found for the URL indicated by a given link. For instance, the Wayback Machine is designed such that when a visitor clicks on a hyperlink on an archived page that points to another URL, the visitor will be served the archived file found for the hyperlink's URL with the closest available date to the initial file containing the hyperlink.
4. The archived data made viewable and browseable by the Wayback Machine is obtained by use of web archiving software that automatically stores copies of files available via the Internet, each file preserved as it existed at a particular point in time.
5. The Internet Archive assigns a URL on its site to the archived files in the format `http://web.archive.org/web/[Year in yyyy][Month in mm][Day in dd][Time code in hh:mm:ss]/[Archived URL]` aka an "extended URL". Thus, the extended URL `http://web.archive.org/web/19970126045828/http://www.archive.org/` would be the URL for the record of the Internet Archive home page HTML file (`http://www.archive.org/`) archived on January 26, 1997 at 4:58 a.m. and 28 seconds (1997/01/26 at 04:58:28). The date indicated by an extended URL applies to a preserved instance of a file for a given URL, but not necessarily to any other files linked therein. Thus, in the case of a page constituted by a primary HTML file and other separate files (e.g., files with images, audio, multimedia, design elements, or other embedded content) linked within that primary HTML file, the primary HTML file and the other files will each have their own respective extended URLs and may not have been archived on the same dates.





6. Attached hereto as Exhibit A are true and accurate copies of browser screenshots of the Internet Archive's records of the archived files for the URLs and the dates specified in the attached coversheet of each screenshot.
7. I declare under penalty of perjury under the laws of the United States of America that the foregoing is true and correct.

DATE: 04/30/2025

Tanya Zeif

Tanya Zeif

Please see attached
All Purpose
Jurat form
for additional
Notary Events



EXHIBIT A



The Wayback Machine - <https://web.archive.org/web/20130222055122/http://caloreequipment.com:80/products/cl/bbq-grill-low-pressure-regulators/two-stage-...>

Search entire store here...

[My Account](#) [My Wishlist](#) [My Cart](#) [Checkout](#) [Log In](#)
[Home](#) [About](#) [Products](#) [Media & Resources](#) [Customer Service](#) [Commerical Services](#) [Contact](#)
[Home](#) / [BBQ Grill Low Pressure Hose and Regulator](#) / **Two-Stage Low Pressure Hose & Regulator**

Two-Stage Low Pressure Hose & Regulator



[2-Stage Low Pressure Regulator Assembly](#)

\$70.00 SKU: 106238

Used for connecting commercial grade BBQ & Catering Grills requiring gas input in excess of 100,000 BTU's per hour to a propane tank via high volume green Type 1 tank connection. The green Type 1 allows a higher volume of gas to pass through the tank connection before going into bypass mode. *Standard Hose Length is 18" OAL [Learn More](#)

[Add to Wishlist](#) | [Add to Compare](#)


[2-Stage Low Pressure Regulator Assembly with Stainless Steel Overbraid](#)

\$90.00 SKU: 106221

Stainless Steel Overbraided Two-Stage Regulator Assembly used for connecting commercial grade BBQ & Catering Grills requiring gas input in excess of 100,000 BTU's per hour to a propane tank via high volume green Type 1 tank connection. The green Type 1 allows a higher volume of gas to pass through the tank connection before going into bypass mode. The overbraid keeps small animals from damaging the hose. *Standard Hose Length is 18" OAL [Learn More](#)

[Add to Wishlist](#) | [Add to Compare](#)


[Type1 Cylinder Connection \(Light Green\)](#)

\$12.00 SKU: 204039

The light green Type 1 is used to connect an outside threaded LP cylinder to high pressure appliances such as portable BBQ grills, camping stoves, and turkey fryers. Eliminates the need for wrenches and features a thermal shut-off and excess flow device. Max output is 200,000 BTU/hr. MB Sturgis has designed the green Type 1 to provide the highest flow rate possible before going into bypass mode. The light green Type 1 is also designed to allow the hose to depressurize itself when the hose is disconnected from the appliance allowing the pent up gas to escape rather than storing a pressurized LP gas hose. [Learn More](#)

[Add to Wishlist](#) | [Add to Compare](#)

COMPARE PRODUCTS

You have no items to compare.

MY CART

You have no items in your shopping cart.

RECENTLY VIEWED PRODUCTS

[Sturgi-Stay Kit](#)
[Sturgi-Stay Kit Deluxe](#)
[Quick Disconnect Sturgi-Stay Deluxe Kit](#)
[Quick Disconnect Sturgi-Stay Fitting](#)
[Model 250 Male Plug X 1/8" MNPT](#)

<https://web.archive.org/web/20130222055122/http://caloreequipment.com:80/products/cl/bbq-grill-low-pressure-regulators/two-stage-low-pressure-regulator.html>



JURAT ATTACHMENT

A notary public or other officer completing this certificate verifies only the identity of the individual who signed the document to which this certificate is attached, and not the truthfulness, accuracy, or validity of that document.

STATE OF Texas }

COUNTY OF Harris }

The foregoing instrument was subscribed and sworn before me this date of
04/30/2025, by Tanya Zeif

This notarial act was an online notarization.



Document Notarized using a Live Audio-Video Connection

(Notary Seal)

Notary's Signature Ana Laura Salazar Uribe

Registration No.: 131757026

Commission Expiration Date: October 11, 2026



EXHIBIT 1021

Internal Combustion Engines

Edited by

CONSTANTINE ARCOUMANIS

**Department of Mechanical Engineering,
Imperial College of Science and Technology,
London, UK**

1988



ACADEMIC PRESS

Harcourt Brace Jovanovich, Publishers
London San Diego New York Berkeley
Boston Sydney Tokyo Toronto

ACADEMIC PRESS LIMITED
24/28 Oval Road,
London NW1

United States Edition published by
ACADEMIC PRESS INC.
San Diego, CA 92101

Copyright © 1988 by
ACADEMIC PRESS LIMITED
except Chapter 6,
Copyright © 1988
FORD MOTOR COMPANY, USA

All Rights Reserved

No part of this book may be reproduced in any form by photostat, microfilm, or
any other means without written permission from the publishers

British Library Cataloguing in Publication Data

Internal combustion engines.
1. Internal combustion engines
I. Arcoumanis, C.
621.43

ISBN 0-12-059790-X

Typeset by Colset Private Limited, Singapore
Printed in Great Britain by
St Edmundsbury Press Limited, Bury St Edmunds, Suffolk

Combustion Treatise

Series Editor

J. H. WHITELAW

*Department of Mechanical Engineering
Imperial College of Science and Technology,
London, UK*

Advanced Chemical Rocket Propulsion

Y. M. Timnat, 1987

Principles of Combustion Engineering for Boilers

C. J. Lawn (ed), 1987

Internal Combustion Engines

C. Arcoumanis (ed), 1988

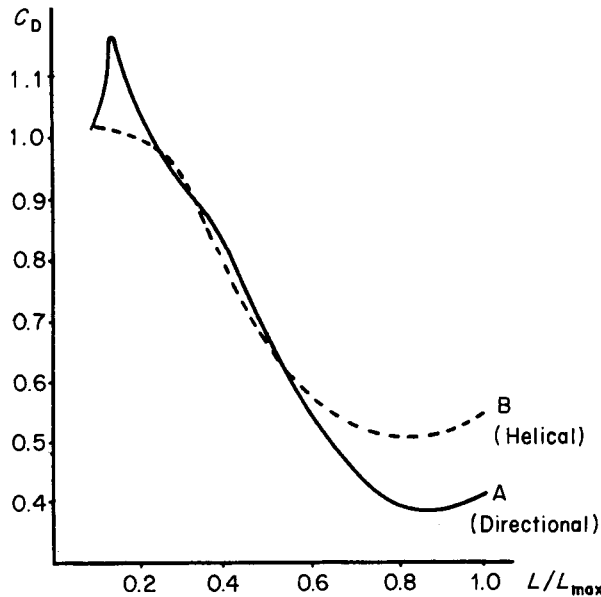


Figure 11 Variation of flow coefficient with valve lift for directed and helical inlet ports (Tindal and Williams, 1977). Reprinted with permission © 1977 Society of Automotive Engineers, Inc.

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

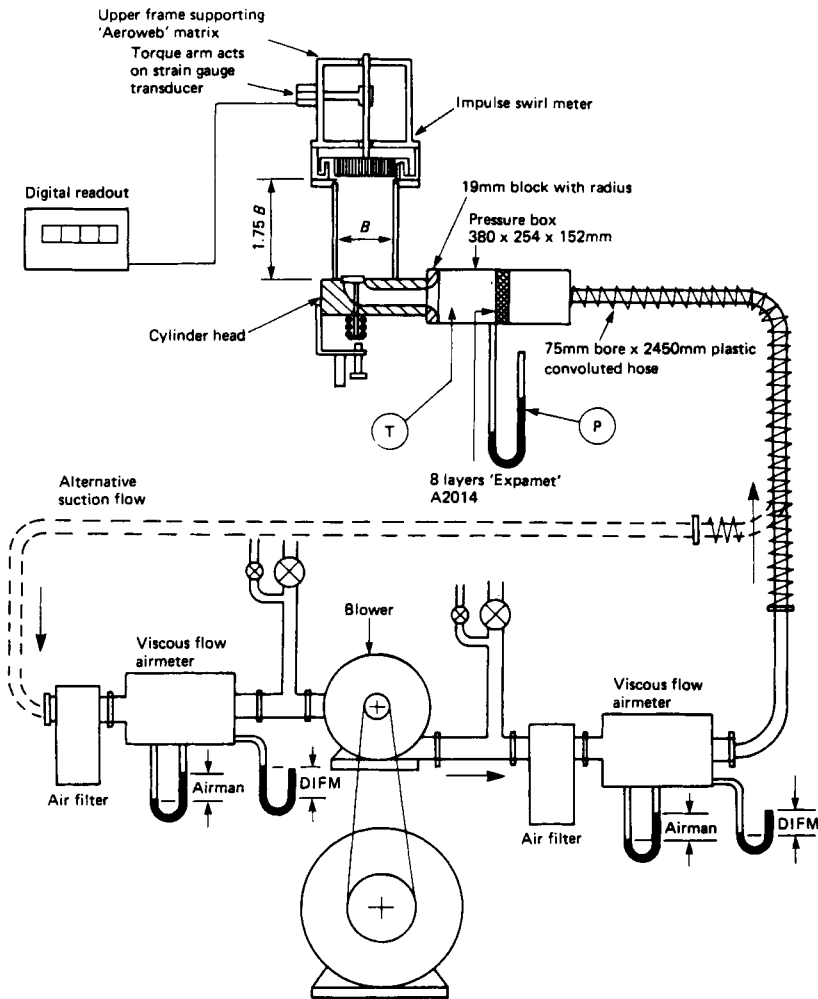


Figure 12 Steady flow rig for testing inlet ports (Watts and Scott, 1970). Reprinted by permission of the Council of the Institution of Mechanical Engineers from Symposium on Diesel Engine Combustion, 1970.

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.

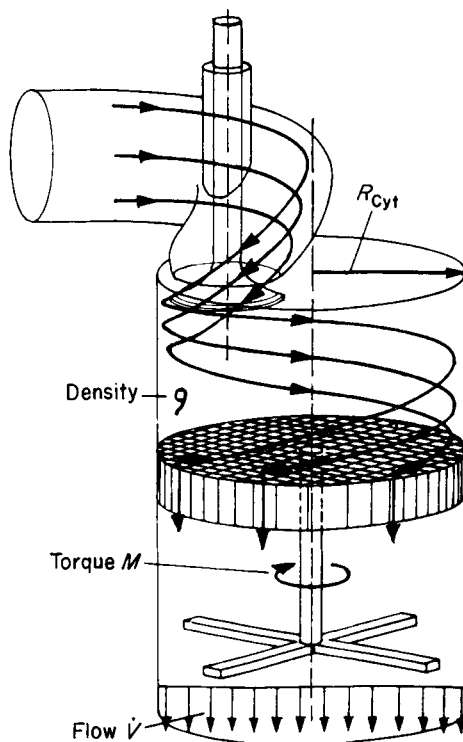


Figure 13 Impulse momentum meter (Tippelman, 1977). Reprinted with permission © 1977 Society of Automotive Engineers, Inc.

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

$$\frac{d}{dt} \left(\frac{\pi D^3}{6} \right) = - \frac{\pi \lambda}{4} D$$

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 naphthalene; 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

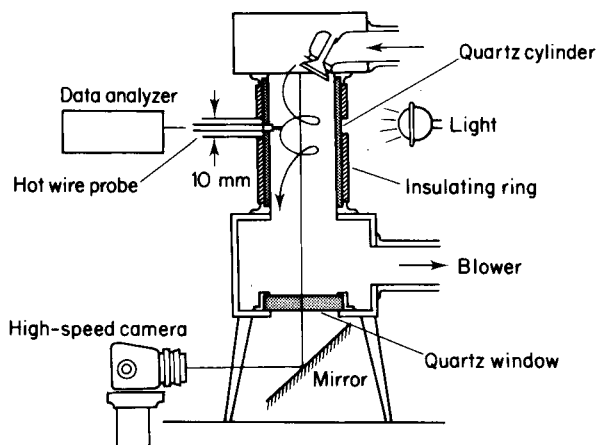


Figure 27 Experimental apparatus for gas flow study in cylinder by high speed photography and hot wire anemometry (Matsushita *et al.*, 1985).

hot wire anemometry 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

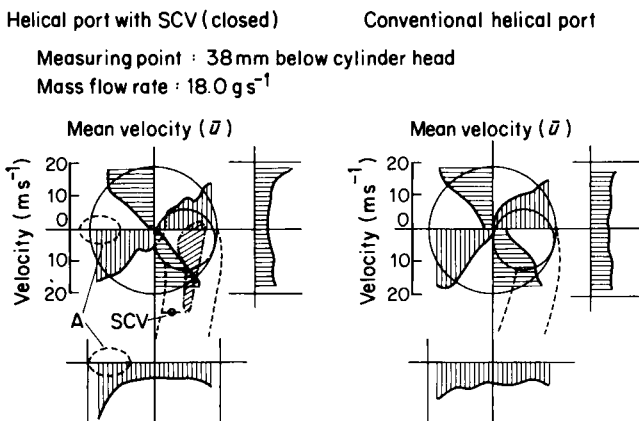


Figure 28 Flow velocity pattern for conventional helical port and SCV valve closed (Matsushita *et al.*, 1985).

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.

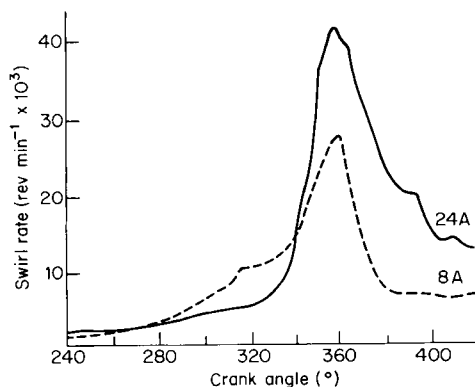


Figure 29 Comparison of swirl rates in toroidal and re-entrant chambers. Case 24a denotes the re-entrant chamber (Williams and Tindal, 1980).

EXHIBIT 1022

Alternative Transportation FUELS

Utilisation in Combustion Engines



M. K. Gajendra Babu
K. A. Subramanian



CRC Press

Taylor & Francis Group

Petitioner MWE Investments, LLC, et al.
Exhibit 1003, Page 1543

Alternative Transportation FUELS

Utilisation in Combustion Engines

**M. K. Gajendra Babu
K. A. Subramanian**



CRC Press

Taylor & Francis Group

Boca Raton London New York

CRC Press is an imprint of the
Taylor & Francis Group, an **informa** business

CRC Press
Taylor & Francis Group
6000 Broken Sound Parkway NW, Suite 300
Boca Raton, FL 33487-2742

© 2013 by Taylor & Francis Group, LLC
CRC Press is an imprint of Taylor & Francis Group, an Informa business

No claim to original U.S. Government works
Version Date: 20130305

International Standard Book Number-13: 978-1-4398-7282-6 (eBook - PDF)

This book contains information obtained from authentic and highly regarded sources. Reasonable efforts have been made to publish reliable data and information, but the author and publisher cannot assume responsibility for the validity of all materials or the consequences of their use. The authors and publishers have attempted to trace the copyright holders of all material reproduced in this publication and apologize to copyright holders if permission to publish in this form has not been obtained. If any copyright material has not been acknowledged please write and let us know so we may rectify in any future reprint.

Except as permitted under U.S. Copyright Law, no part of this book may be reprinted, reproduced, transmitted, or utilized in any form by any electronic, mechanical, or other means, now known or hereafter invented, including photocopying, microfilming, and recording, or in any information storage or retrieval system, without written permission from the publishers.

For permission to photocopy or use material electronically from this work, please access www.copyright.com (<http://www.copyright.com/>) or contact the Copyright Clearance Center, Inc. (CCC), 222 Rosewood Drive, Danvers, MA 01923, 978-750-8400. CCC is a not-for-profit organization that provides licenses and registration for a variety of users. For organizations that have been granted a photocopy license by the CCC, a separate system of payment has been arranged.

Trademark Notice: Product or corporate names may be trademarks or registered trademarks, and are used only for identification and explanation without intent to infringe.

Visit the Taylor & Francis Web site at
<http://www.taylorandfrancis.com>

and the CRC Press Web site at
<http://www.crcpress.com>

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

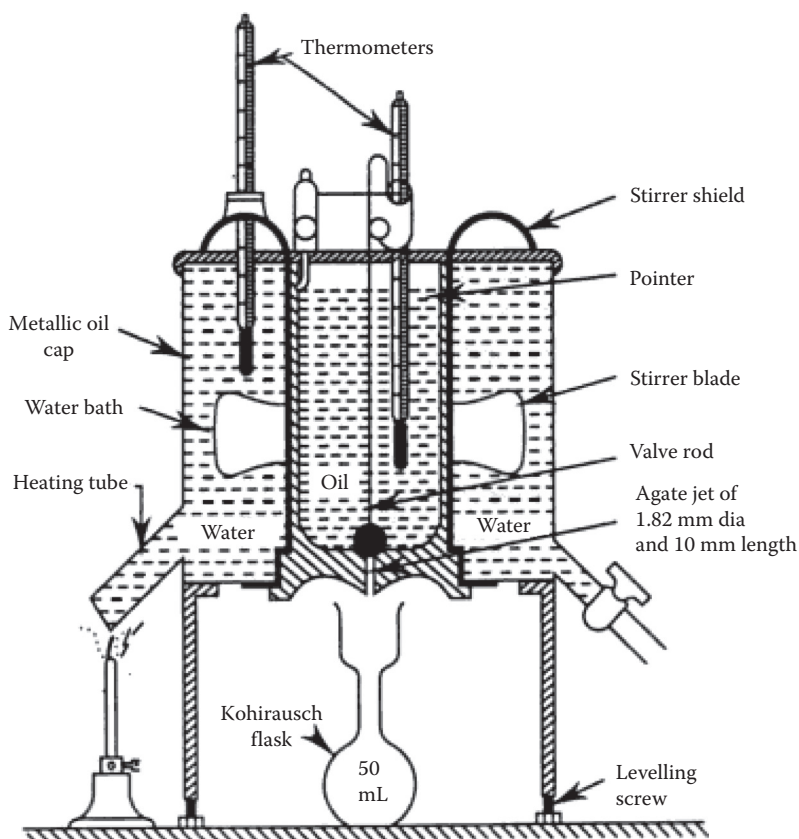


FIGURE 6.2
Redwood viscometer.

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

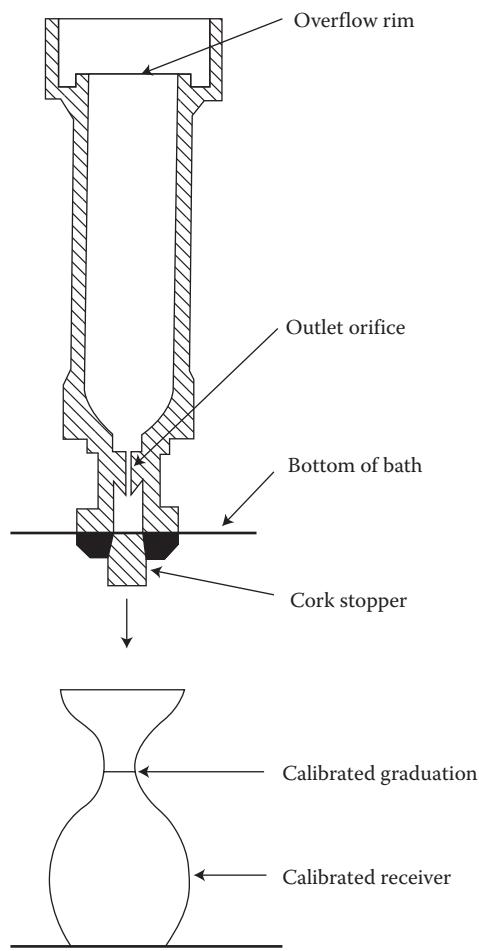


FIGURE 6.3
Saybolt viscometer.

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

$$\text{Kinematic viscosity } (\nu) = (A \times t - (B/t)) \times 10^{-6} \text{ in m}^2/\text{s} \tag{6.8}$$

where

$A = 0.264$, $B = 190$, when $t = 40\text{--}85$ s

$A = 0.247$, $B = 65$, when $t = 85\text{--}2000$ s

The *density* of the given oil is

$$\rho = (w_2 - w_1) \times (10^3/50) \text{ in kg/m}^3 \quad (6.9)$$

where

w_1 = weight of the measuring jar in grams

w_2 = weight of 50 cc of oil in grams

$$\text{Absolute viscosity } (\mu) = \rho \times v \text{ in Pa s or N s/m}^2 \quad (6.10)$$

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

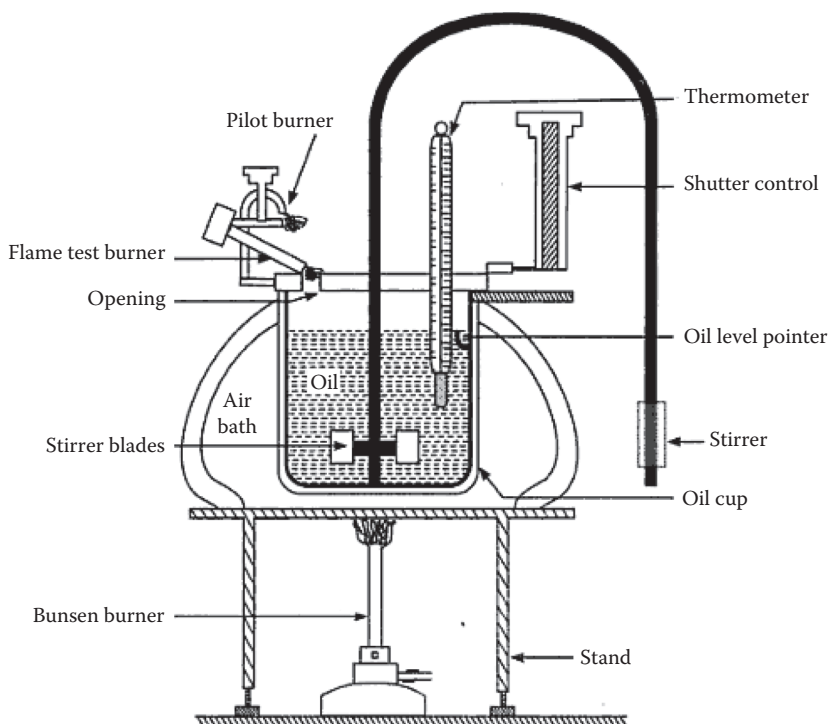


FIGURE 6.4
Ables apparatus.

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 $2^{\circ}\text{C}/\text{min}$ 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 $15\text{--}17^{\circ}\text{C}$ 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

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

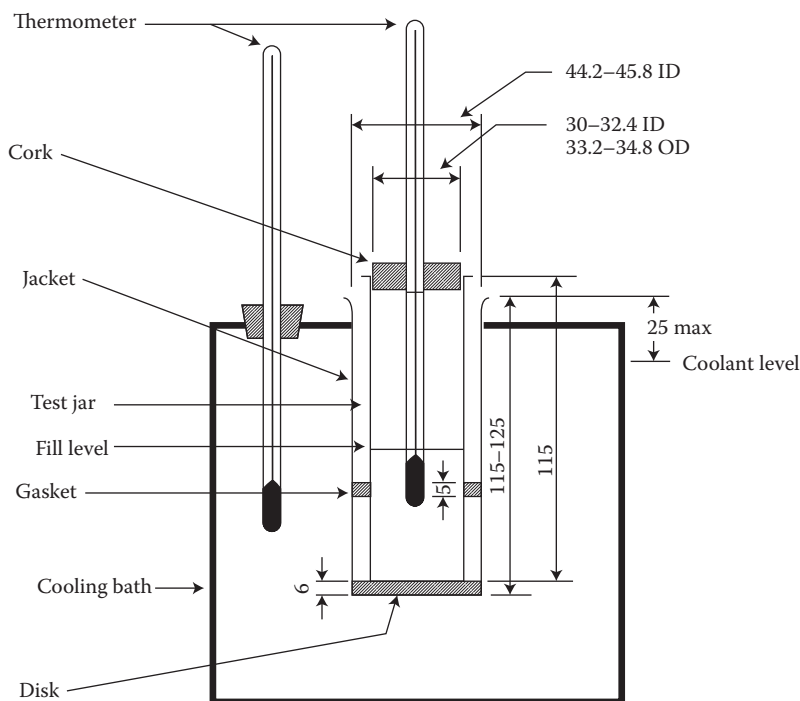


FIGURE 6.5

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 $0 \pm 1.5^{\circ}\text{C}$.

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 $-186 \pm 1.5^{\circ}\text{C}$ (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 $-336 \pm 1.5^{\circ}\text{C}$. 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

EXHIBIT 1023

PREFACE

MECHANICAL ENGINEER'S REFERENCE BOOK

Edited by

A. PARRISH, M.B.E., C.Eng., M.I.Mech.E.

With specialist contributors

BUTTERWORTHS

LONDON

—

BOSTON

Sydney — Wellington — Durban — Toronto

The Butterworth Group

- United Kingdom** **Butterworth & Co (Publishers) Ltd**
London: 88 Kingsway, WC2B 6AB
- Australia** **Butterworths Pty Ltd**
Sydney: 586 Pacific Highway, Chatswood,
NSW 2067
Also at Melbourne, Brisbane, Adelaide and Perth
- Canada** **Butterworth & Co. (Canada) Ltd**
Toronto: 2265 Midland Avenue, Scarborough,
Ontario M1P 4S1
- New Zealand** **Butterworths of New Zealand Ltd**
Wellington: T & W Young Building,
77-85 Customhouse Quay, 1, CPO Box 472
- South Africa** **Butterworth & Co (South Africa) (Pty) Ltd**
Durban: 152-154 Gale Street
- USA** **Butterworth (Publishers) Inc**
Boston: 10 Tower Office Park, Woburn, Mass. 01801

First published as *Newnes Engineers Reference Book* 1946
Eleventh edition (completely revised and reset) 1973
Reprinted 1977, 1978, 1980

© Butterworth & Co (Publishers) Ltd, 1973

All rights reserved. No part of this publication may be reproduced or transmitted in any form or by any means, including photocopying and recording, without the written permission of the copyright holder, application for which should be addressed to the Publishers. Such written permission must also be obtained before any part of this publication is stored in a retrieval system of any nature.

This book is sold subject to the Standard Conditions of Sale of Net Books and may not be re-sold in the UK below the net price given by the Publishers in their current price list.

ISBN 0 408 00083 X

Printed in the United States of America

UNIVERSITY OF ALBERTA

PREFACE

The introduction of SI Metric Units has necessitated a change to British Standards and other National Standards. Technical literature should reflect or anticipate this change. The principal objective of the eleventh edition of this Reference Book which has been completely revised is to give information in SI units only, even though in some instances information is given in the inch or MKSA systems.

It is hoped that this book will meet the needs of management personnel in the Mechanical, Chemical and Engineering disciplines. It should also prove valuable to Engineers and College Lecturers and Students.

The contributors who are drawn from Industrial Associations and Educational bodies have done so with sufficient depth to give a thorough coverage of the subject readily understood by the reader. Further information is given in numerous tables and illustrations. An extensive bibliography is given in each section which gives reference to national and international literature and standards covering the subject matter.

Much of the technology is common to many countries. The work of the International Standards Bodies and the efforts for the removal of barriers to trade. Many of the terms referred to in the text may therefore be compared with the National bodies.

Absolute units at first present difficulty to those who are used to inch and metric systems and it takes time to become used to a newton or a pascal as a force or pressure, because the gravitational systems do not differ from the metric loads (mass) and forces. The expressions, however, are the same when the units are in the coherent (SI) system. Many special names used in the older systems. It is not possible to convert empirical data to another system of units without doing this unilaterally. A case in point is gearing. The units are not yet fully converted to SI. Some of the problems on the former unit systems as, for example, pipe sizes, the inch nominal sizes in the metric standards and the units which are based on the material stresses in kgf/cm² prevent the use of SI units for expressing the properties of materials.

This book describes a wide variety of engineering practices and technology in metric units at the International level so far as it is at present practicable.

Extracts from British Standard publications are given. The British Standards Institution, 2 Park Street, London, has copies of the complete publication may be obtained.

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.

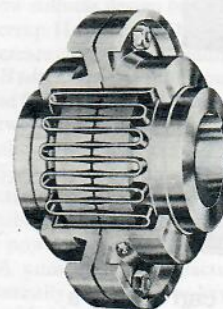
A *flexible* coupling specification should differentiate between the ability to tolerate misalignment and the inherent torsional flexibility. The specification should also clearly define the degree of flexibility required, whether this be radial (torsional) or in respect of an axial or end float motion. Data to be assembled should list the nature of torque

The BIBBY GEAR COUPLING

The Gear Coupling that has all the features you want. Forged steel hubs and sleeves, fully crowned teeth, interchangeability.

BIBBY COUPLING

Renowned for 50 years for absolute dependability and outstanding performance.



The famous BIBBY RESILIENT COUPLING

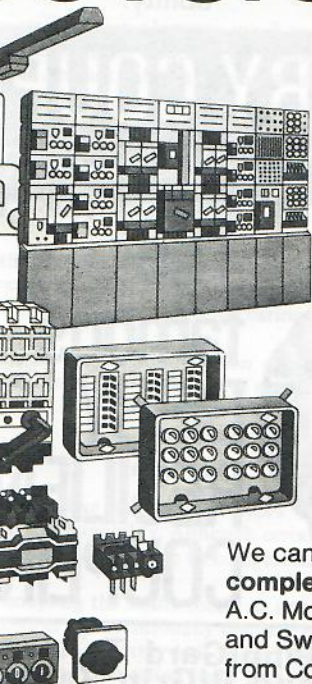
ALSO BibbiGard Torque Coupling and BiGrip Taper Coupling

Technical brochures available



THE WELLMAN
Davis House, High Street
Croydon, CR9 3JL
Tel: 01-688 7104

coverage



We can supply a **complete range of A.C. Motor Control and Switchgear from Components to Rewired Load and Control Centres.**

-MOELLER

Head Office, P.O. Box 35,
Weybridge, Bucks.
Tel: 0181-5. Telex: 83224.

Write today for your copy of our
Main Catalogue.

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

EXHIBIT 1024

Carsten Baumgarten

Mixture Formation in Internal Combustion Engines

With 180 Figures and 9 Tables

 Springer

Dr.-Ing. Carsten Baumgarten,
MTU Friedrichshafen GmbH
Maybachplatz 1
88045 Friedrichshafen
Germany

Series Editors

Prof. Dr.-Ing. Dieter Mewes
Universität Hannover
Institut für Verfahrenstechnik
Callinstr. 36
30167 Hannover, Germany

Prof. em. Dr.-Ing. E.h. Franz Mayinger
Technische Universität München
Lehrstuhl für Thermodynamik
Boltzmannstr. 15
85748 Garching, Germany

Library of Congress Control Number: 2005937086

ISSN - 1860-4846
ISBN-10 3-540-30835-0 Springer-Verlag Berlin Heidelberg New York
ISBN-13 978-3-540-30835-5

This work is subject to copyright. All rights are reserved, whether the whole or part of the material is concerned, specifically the rights of translation, reprinting, reuse of illustrations, recitations, broadcasting, reproduction on microfilm or in any other way, and storage in data banks. Duplication of this publication or parts thereof is permitted only under the provisions of the German copyright Law of September 9, 1965, in its current version, and permission for use must always be obtained from Springer-Verlag. Violations are liable for prosecution under the German Copyright Law.

Springer-Verlag Berlin Heidelberg New York
a member of BertelsmannSpringer Science+Business Media GmbH
springer.com

© Springer-Verlag Berlin Heidelberg 2006
Printed in Germany

The use of general descriptive names, registered names trademarks, etc. in this publication does not imply, even in the absence of a specific statement, that such names are exempt from the relevant protective laws and regulations and therefore free for general use.

Typesetting: Digital data supplied by editor
Cover design: deblik Berlin
Printed on acid free paper 62/3020/SPI Publisher Services - 5 4 3 2 1 0

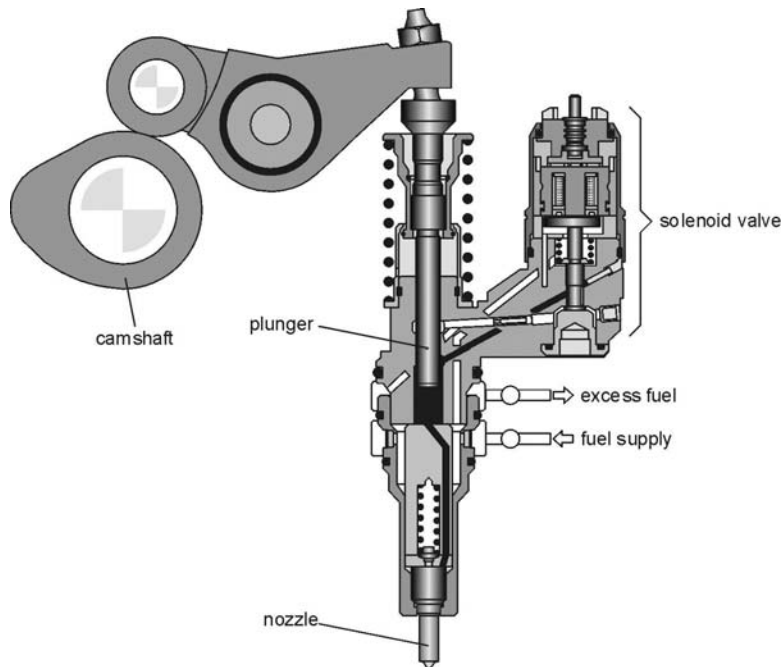


Fig. 2.27. Unit Injector [46]

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

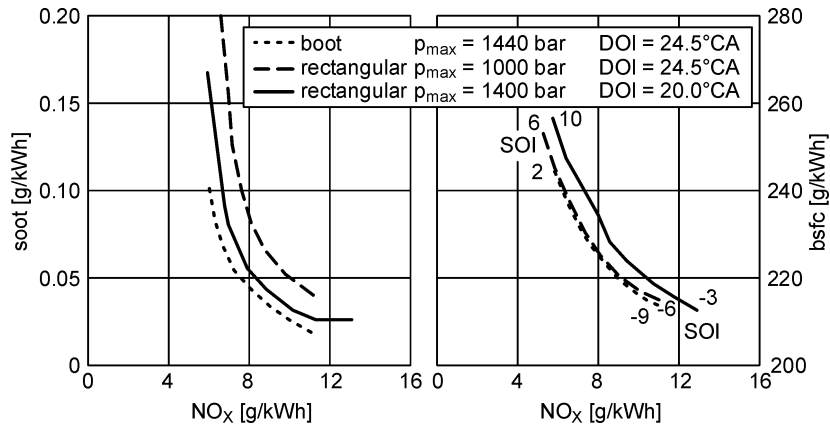


Fig. 6.7. Comparison of rectangular and boot-type injection, single cylinder engine, displacement: 1 liter, 1400 rpm, 100% load, no EGR [52]. SOI and DOI: start and duration of injection

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

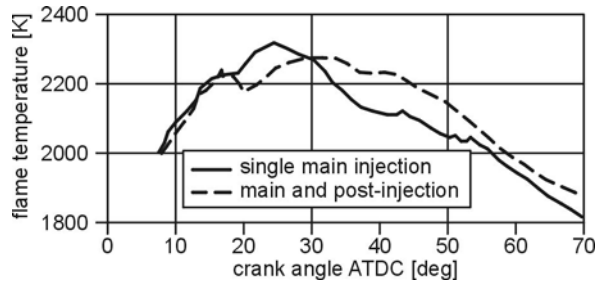


Fig. 6.8. Increase of temperature due to an early post-injection [12]

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 decreases because more fuel leaves

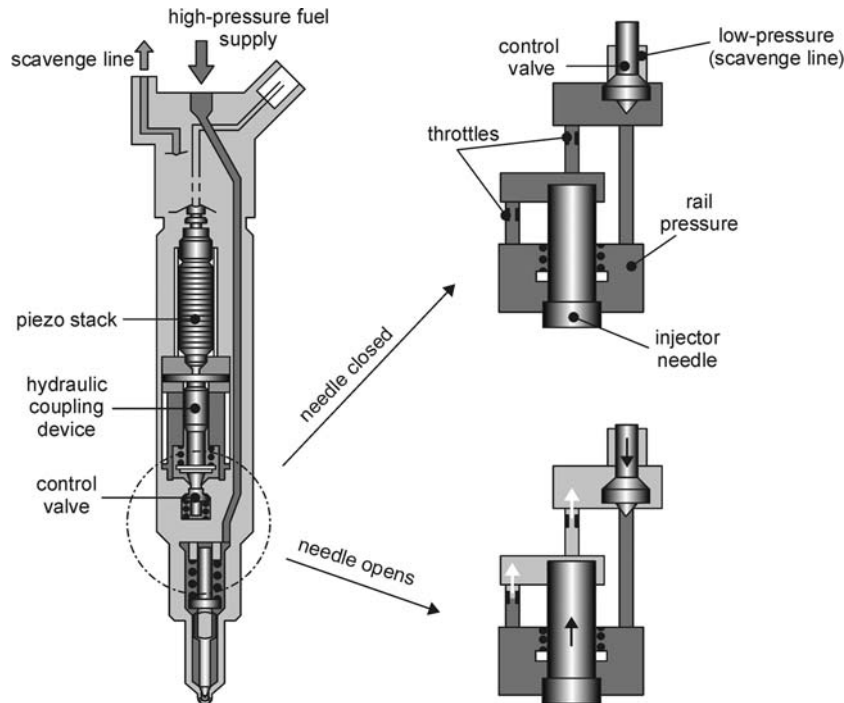


Fig. 6.9. Piezo in-line injector from Bosch [70]

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.

EXHIBIT 1025

Günter P. Merker
Christian Schwarz
Rüdiger Teichmann
Editors

Combustion Engines Development

Mixture Formation, Combustion,
Emissions and Simulation



Springer

Petitioner MWE Investments, LLC, et al.
Exhibit 1003, Page 1566

Editors

Univ.-Prof. Dr.-Ing. habil.
Günter P. Merker
Schöneckstr. 30
88060 Tettnang
Germany
g.p.merker@t-online.de

Apl. Prof. Dr.-Ing. habil.
Christian Schwarz
BMW-Group
EA 31
Hufelandstr. 8a
80788 München
Germany
christian.schwarz@bmw.de

Dr.-Ing. Rüdiger Teichmann
AVL List GmbH
Hans-List-Platz 1
8020 Graz
Austria
ruediger.teichmann@avl.com

Original german edition published by Vieweg & Teubner, Wiesbaden, 2009.

ISBN 978-3-642-02951-6 e-ISBN 978-3-642-14094-5
DOI 10.1007/978-3-642-14094-5
Springer Heidelberg Dordrecht London New York

Library of Congress Control Number: 2011936001

© Springer-Verlag Berlin Heidelberg 2012

This work is subject to copyright. All rights are reserved, whether the whole or part of the material is concerned, specifically the rights of translation, reprinting, reuse of illustrations, recitation, broadcasting, reproduction on microfilm or in any other way, and storage in data banks. Duplication of this publication or parts thereof is permitted only under the provisions of the German Copyright Law of September 9, 1965, in its current version, and permission for use must always be obtained from Springer. Violations are liable to prosecution under the German Copyright Law.

The use of general descriptive names, registered names, trademarks, etc. in this publication does not imply, even in the absence of a specific statement, that such names are exempt from the relevant protective laws and regulations and therefore free for general use.

Printed on acid-free paper

Springer is part of Springer Science+Business Media (www.springer.com)

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/cm² 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.

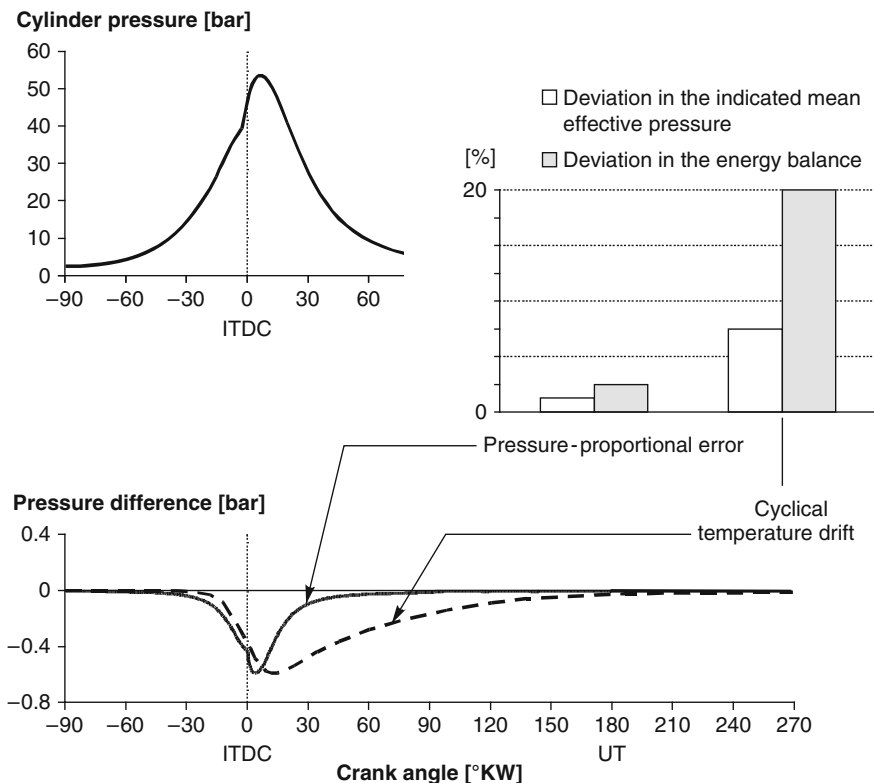


Fig. 3.13 Deviations in the indicated mean pressure and in the energy balance due to a pressure-proportionate error and cyclical temperature drift

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.

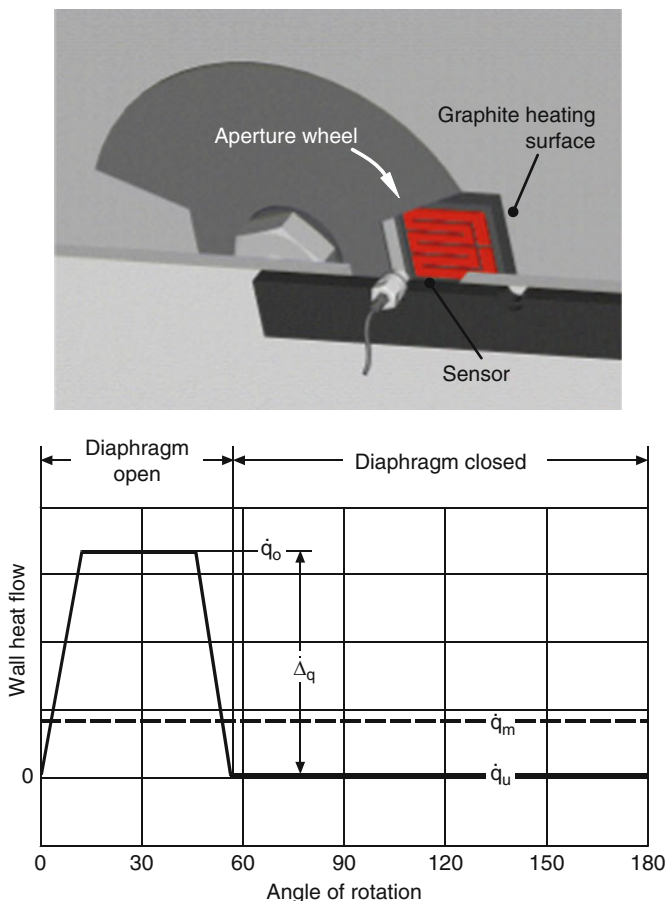


Fig. 3.14 Dynamic test apparatus to determine cyclical temperature drift

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

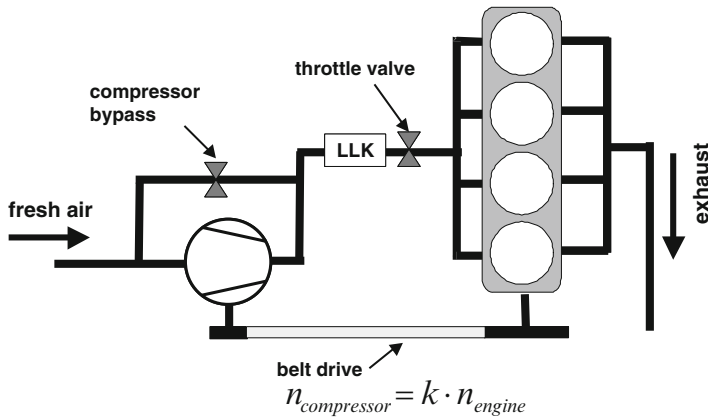


Fig. 8.5 Mechanical supercharging: block diagram

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.

The advantages and disadvantages of mechanical supercharging are summarized in the list below.

- Advantages:
 - Spontaneous response behavior thanks to immediately available charge air in case of a sudden load change when a reduction of the compressor power can be quickly eliminated at partial load.
 - High boost pressures already available at low engine speeds (limitation: surge limit in the case of the mechanically powered turbo compressor).
 - Response behavior is hardly affected by exhaust gas aftertreatment systems.
 - No changes are required in the exhaust apparatus since only the inlet side is affected.
 - The exhaust back pressure does not influence the gas exchange/lower gas exchange work.
 - No adverse effects on heating and thus on the response behavior of exhaust gas aftertreatment systems (catalytic converters, soot filters etc.) in case of a cold start or unsteady processes (no heat extraction through the turbines).
 - No heating of the fresh gas via heat transfer from the exhaust (as in the case of the turbocharger).
 - No influence on the bearings or materials of the compressor due to exhaust heat.
 - There is no metal mass in the engine room which can heat up and radiate heat to the environment.
- Disadvantages:
 - Specific fuel consumption at partial load without corresponding control can be higher than that of the natural aspirated engine.
 - High costs for design and control is required to lower the compressor power (coupling, speed shifting and valve control).
 - Under certain conditions, response time is delayed because of shifting or positioning delays in case of a sudden load requirement.
 - High costs due to complex charging aggregates, transmissions with sometimes their own belt drives and difficult controls.
 - The smaller engine installation volume with the same power is compensated by the installation space for the charging aggregate and the drive.
 - Higher noise properties in the case of positive displacement chargers due to the unsteady operation.
 - In the case of the mechanically coupled compressor, the interaction of the engine and the charging aggregate also causes a reduction of the boost pressure at low engine speeds and of the useful engine torque due to the high compressor drive power at full engine load and due to the often high leakage losses of the charging aggregate at low speeds.
 - In the case of positive displacement chargers with abrasive sealing gaskets, there are high friction losses and a resultantly low overall efficiency and often a large amount of wear.
 - To obtain small gap widths, a considerable amount of manufacturing cost is required for positive displacement chargers.

In the case of engines with high efficiency requirements (slow and average-speed large diesel engines), the turbocharger has become preferred to the supercharger due to the high boost ratio and the better overall efficiency. For commercial and diesel passenger applications as well, mechanical supercharging has not become widespread despite its advantages in unsteady conditions due to its complexity and lower efficiency. Only in the field of SI passenger cars has mechanical charging been applied in some niche applications without cost pressure or in combination with turbocharging in order to improve unsteady behavior in the case of extreme downsizing concepts. Here too, the overall efficiency of mechanical charging is at most comparable with turbocharging with moderate boost grades.

8.1.3 Turbocharging

In turbocharging, a turbo compressor is powered with an exhaust turbine using energy from the exhaust gas, i.e. the reciprocating engine is only fluidically connected to the turbo machine “turbocharger” (Fig. 8.8). The exhaust gas of the internal combustion engine slows through the exhaust manifold to the turbine and powers it. A compressor sitting on the main shaft of the turbine, which can in fixed operation convert the drive power of the turbine into compressor power (minus bearing friction), itself compresses fresh air, which must be cooled by a charge air cooler. Via the manifold, the compressed air enters the cylinder. In the case of highly unsteady applications (e.g. vehicle operation), the power of the turbine must be controlled. This is done in the schematic shown here with a so-called wastegate.

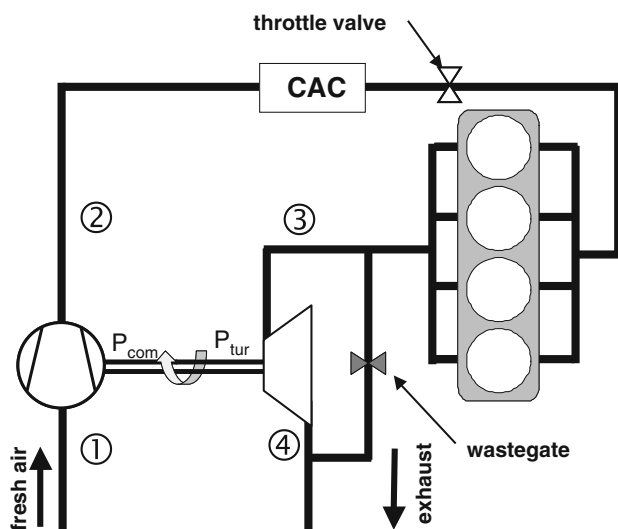


Fig. 8.8 A reciprocating engine and turbo machine. Indexes: 1: b.com, 2: a.com, 3: b.tur, 4: a.tur

This is the “second fundamental equation of turbocharging”

$$\dot{m}_{tur} \frac{\sqrt{T_{b,tur.}}}{p_{b,tur.}} = A_{is,tur} f\left(\frac{p_{a,tur.}}{p_{b,tur.}}, \kappa_{tur}\right). \quad (8.15)$$

With the help of this relation, we can form reference or dimensionless variables. This will be dealt with in more detail in Sect. 8.2.

The isentropic turbine substitute cross section is approximately constant and depends only on the geometrical turbine cross section; we thereby ignore its dependence on the speed of the turbocharger and on the pressure ratio $p_{b,tur}/p_{a,tur}$.

8.1.3.3 Interaction with the Exhaust Dynamics of the Internal Combustion Engine

Strictly speaking, turbocharging is classified as either constant-pressure turbocharging or pulse turbocharging.

In constant-pressure turbocharging, only the thermal energy of the exhaust is utilized, since the exhaust of a single cylinder of an internal combustion engine is guided into a large pipe volume between the engine and the turbine, in which it loses its momentum. This momentum results from the discontinuous working process of the internal combustion engine, in which the exhaust abruptly escapes into the manifold when the outlet valves open. There is thus a constant pressure before the turbine in this type of turbocharging, which also has an effect on the individual cylinders. For this reason, a free selection of the ignition sequence of the combustion engine is possible in the case of ideal constant-pressure turbocharging from the thermodynamic perspective. Also, there is no limitation on the cylinder number of the engine joined to the outlet volume. The constant pressure before the turbine has, on the one hand, a positive effect on the average efficiency and suction behavior of the turbine. On the other hand, the velocity contained in the individual impulses of exhaust gas is first converted into thermal energy and thus into static pressure. This energy must then again be converted in the turbine into the more valuable kinetic energy (nozzle, guide apparatus). Since each process of energy conversion is bound with losses however, corresponding losses arise both in the uniformization of the flow and the reacceleration of the flow. Besides the high demands on the package, which can only be met for a stationary application, the exhaust also loses a considerable amount of energy due to the large surface of the volume connected between the engine and the turbine. A large volume with highly unstable engine operation is especially problematic, e.g. as in an airplane, since filling the volume with exhaust in order to obtain a corresponding pressure and thus a corresponding amount of turbine power takes a long time. This is especially true at low speed and loads, since in this case there is already a low pressure level with low mass flows.

In pulse turbocharging on the other hand, the aim is to direct the exhaust to the turbine using as short a path as possible with as little loss as possible. Both the

thermal and kinetic energy of the exhaust is exploited. The exhaust system of a pulse-turbocharged engine is therefore characterized by its compactness. The flow of the exhaust is however very inconsistent, since the pressure waves in the exhaust system propagate with sonic speed and, due to the compactness of the pipe system, quickly affect the exhaust counterpressure of the other cylinders joined to the pipe system. This has adverse effects on the turbine on the one hand, which is applied with highly erratic mass and energy flows. The turbine blades are not optimally impacted with flow at every moment in the process. This in turn has a negative effect on efficiency and eliminates part of the benefits. The latter also has a negative effect on the mechanical and thermal strength of the turbine.

On the other hand, the high fluctuations in pressure also have a great effect on the engine gas exchange. Depending on the ignition sequence and the outlet valve control times of the cylinders combined in the exhaust system, it is possible that gas exchange during the push-out phase can be considerably impaired by the previous or subsequent cylinder. The result is that either the pushing-out work becomes very high or that residual gas remains in the cylinder and the combustion of the next working cycle is negatively affected. In the case of a four-stroke engine, three cylinders can be joined assuming a control time of max. 260°CA for one outlet valve. This means that in the case of three or six-cylinder engines (or all cylinder numbers dividable by 3), an aggregation of cylinders that is optimal with respect to gas exchange and turbocharging is possible in principle.

The six-cylinder engine requires however an arrangement of three cylinders each on either two turbines or on one double-flow turbine (twin-scroll), which makes flow separation possible (see below). In the case of the four-cylinder engine, there is an ignition distance of 180°CA , which is why only two cylinders can be joined optimally, which again means two exhaust flows (twin-scroll turbine). Nevertheless, four-cylinder engines in particular also have single-flow aggregates, but to significant disadvantage at low speeds and high loads and in their response behavior. In the V8 engine, the usual ignition sequence necessitates an ignition distance of only 90°CA on every bank, which can only be poorly compensated by flow combination. In this case, usually only combination of four cylinders on one bank remains.

Since complete reduction of the pressure pulse itself is not possible in the case of slowly running engines, while on the other hand an ideal utilization of the exhaust pulses is tied to a real volume of the exhaust system, no pure form of constant-pressure or pulse turbocharging exists.

8.1.3.4 Flow Turbine Designs

In the previous section, the necessity of different flow turbine designs was already pointed out. The basic in this section will be on radial turbines, which can be subdivided into monoscroll – and twin-scroll – or twin turbines. As the name already suggests, monoscroll turbines have only one entry snail which directs the exhaust to the rotor. Figure 8.15 (left) shows a cross-section through a monoscroll

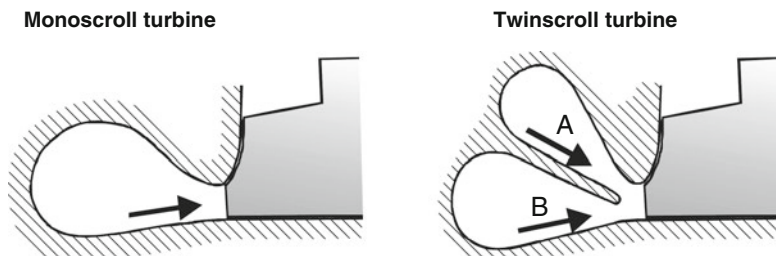


Fig. 8.15 Cross-section through a monoscroll and a twin-scroll turbine

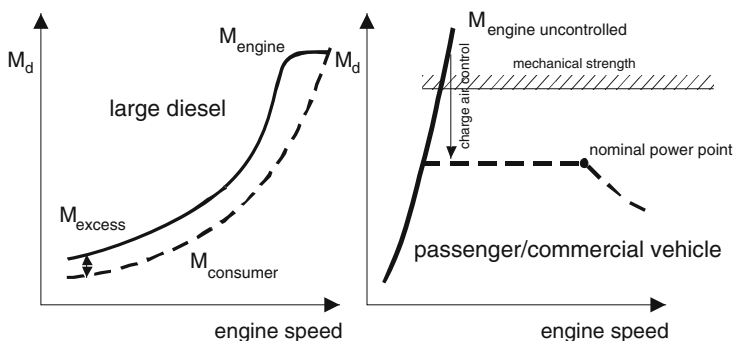


Fig. 8.16 Engine torque and load torque for large diesel and vehicle engines

turbine. On the right is a twin-scroll turbine. We can recognize the two parallel snails that direct the exhaust flows of each cylinder aggregate completely separately to the rotor of the turbine. To fully exploit the functionality of twin-scroll turbines, a complete and consistent separation of the flows until entering the rotor must be assured in order to avoid crosstalk of the gas pulses.

8.1.3.5 Charge Pressure Control Measures

Due to the characteristics of turbo machines, a turbocharger designed for full load without any external control has an approximately quadratic dependence of the boost pressure on the engine speed. At low engine speeds, only a small boost pressure and thus only a small engine torque can be created as a rule. Figure 8.16 (left) clarifies this behavior. This tendency is not as critical for medium speed diesel engines, whose time response is usually not decisive because of the stationary operation near the nominal power point, than for small, high-speed engines, which require high torque already at very low engine speeds due to their dynamic operation. For this reason, measures must be taken in such engines to increase the boost pressure and thus the torque availability at low engine speeds. In the basic design of

the turbine, value is placed on high pile-up pressure and thus on high turbine performance already with low mass flows. That means that the size of the turbine is not optimally designed for the nominal power point, but for a much lower engine speed ($< 2,000$ rpm). With a smaller turbine size, the pressures and turbocharger speeds would become very high with increasing engine speeds due to the almost quadratic increase of mass flow through the compressor and turbine, which would exceed the mechanical strength of the charging system and the combustion engine. Moreover, the charge pressure would increase dramatically, which would lead to serious knocking problems in SI engines due to the resultant high temperatures. For these reasons, boost pressure, exhaust counterpressure and the speed of the turbocharger must be limited by suitable measures in the charging system.

Until now, two methods have become established on the turbine side of the exhaust turbocharger. On the one hand, it is possible to control the exhaust mass flow through the turbine with a wastegate (Fig. 8.8). In this case, part of the exhaust mass flow is guided past the turbine and is thus no longer available for the production of drive power, which is why the compressor conveys a smaller mass flow. In this way, the boost pressure before the engine can be controlled and thus the engine torque as well. For wastegates, the control elements are either pressure sensors that are applied with boost pressure or negative pressure, or electric control elements.

It is also possible to limit the power of the turbine by means of variability in the guide vanes by changing the cross-section by turning the guide vanes. When the cross-section is reduced, this causes a higher pile-up pressure and thus larger turbine power, and if the cross-section is enlarged, it is possible to reduce the pile-up pressure and turbine power.

Nearly all modern turbocharged car diesel engines have charge pressure control using an adjustable guide apparatus. Turbocharged SI engines as a rule require a wastegate because of their much higher exhaust temperatures, since adjustable guide vanes would represent a much larger function risk.

Moreover, SI engines with conventional air-mass load control require, for acoustic reasons, a “diverter valve”. By means of reverse flow at the compressor, this valve makes possible a rapid reduction of the charge pressure in case of a sudden intrusion of mass flow due to load reduction. Without a diverter valve, the operating point of the compressor in the characteristic map would shift to the left over the surge limit (see Fig. 8.17) due to the inertia of the exhaust turbocharger. If one keeps the mass flow through the compressor to a higher level with the diverter valve, it is possible not to exceed the surge limit until the speed of the compressor has been reduced.

Figure 8.17 shows the characteristic map of a turbo compressor with its limits (surge, speed and choke limit). The structure of the map is described in more detail in Sect. 8.2.

The full load operating line is drawn in the characteristic map, which runs almost quadratically in the left part. We can clearly see the kink in the operating line, which is caused by the opening of the wastegate due to the charge pressure control described above.

8.1.3.7 Interaction with the Internal Combustion Engine

Figure 8.17 shows the interaction of a internal combustion engine and a turbocharger with the help of the stationary full load operating line in the compressor characteristic map. This operating line is drawn in Fig. 8.20 in the torque/speed diagram in comparison to a natural aspirated engine of the same capacity as well as a mechanically charged engine. We can clearly see that in the case of the steady-state operation of a turbocharged engine a high torque can be reached already at very low speeds (“low end torque” LET). The torque plateau can be held constant by the wastegate control described above until the desired power is reached (power hyperbola). The early position of LET is made possible by the variable control times in the SI engine caused by the inlet and outlet camshaft phasing. In the case of a positive pressure difference at the turbocharger, in which case the charge pressure is larger than the exhaust counterpressure, in combination with a large overlap of outlet and inlet valve opening during gas exchange, a complete purging of the residual gas from the cylinder and a filling of the cylinder with cold fresh air is possible. In combination with direct injection, the air fuel ratio in the cylinder can be set slightly rich, which causes very high combustion velocities. The result is a largely knock-free operation, even at low engine speeds along with high torque values.

Figure 8.21 shows the unsteady behavior of a turbocharged SI engine at a constant speed of 1,500 rpm and a load step from partial load to full load, from which we can see that the flow-type coupling of both sub-components results in a significantly delayed torque build-up compared with the steady-state behavior of the engine. Section 10.5 presents some simulation calculations for improving unsteady behavior and for explaining individual effects.

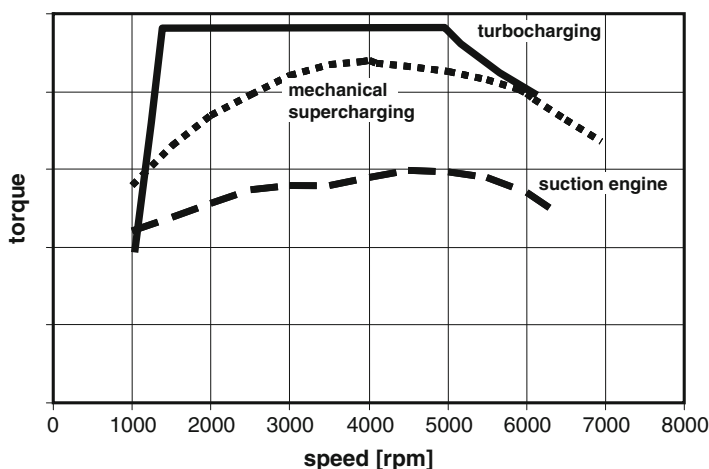
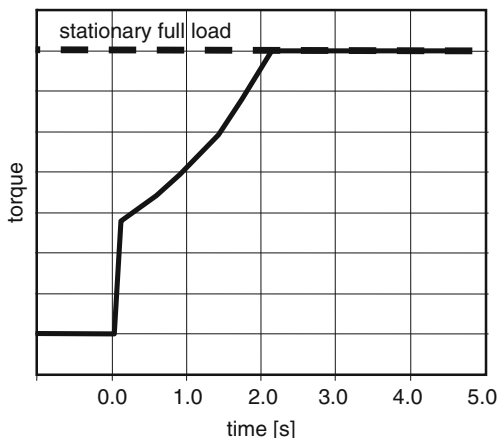


Fig. 8.20 The torque profile of various charging concepts

Fig. 8.21 Unsteady torque build-up in a turbocharged engine



The following summarizes the advantages and disadvantages of exhaust turbocharging.

- Advantages:
 - The exhaust counterpressure is reduced in case of partial load. A so-called “self-regulation effect” of the turbocharger: the specific fuel consumption is reduced with it as well.
 - Small turbine cross-sectional surfaces, small polar moments of inertia and high potential turbocharger speeds make for a fast, almost delay-free response time (no noticeable turbo lag).
 - No external drive necessary.
 - Economical manufacture and low manufacturing costs for the turbocharger with high production numbers, since they can be adjusted to the engine on the side of both the turbine and the compressor by means of a change of trim.
 - The small construction size of the turbocharger unit. Integration of the turbine into the exhaust manifold results in additional performance advantages and good response behavior.
 - Utilization of exhaust energy improves engine power and, under some conditions, even the indicated efficiency of the engine.
 - The continuous supply of the turbo compressor.
 - Lower thermal stress on the turbocharging system due to temperature reduction via the turbine.
- Disadvantages:
 - The surge limit restricts the charge pressure at low engine speeds.
 - In the case of large turbine housings (cross-sectional areas), the unsteady response time is clearly delayed (turbo lag).
 - High gas exchange losses and poor gas exchange success of the engine due to the high pile-up pressure before the turbine, especially in the case of a negative purge slope and small turbocharger designs.

EXHIBIT 1026



Introduction to Internal Combustion Engines



Introduction to Internal Combustion Engines

RICHARD STONE
*Department of Engineering Science
University of Oxford*

Third Edition





© Richard Stone 1985, 1992, 1999

All rights reserved. No reproduction, copy or transmission of this publication may be made without written permission.

No paragraph of this publication may be reproduced, copied or transmitted save with written permission or in accordance with the provisions of the Copyright, Designs and Patents Act 1988, or under the terms of any licence permitting limited copying issued by the Copyright Licensing Agency, 90 Tottenham Court Road, London W1P 9HE.

Any person who does any unauthorised act in relation to this publication may be liable to criminal prosecution and civil claims for damages.

The author has asserted his right to be identified as the author of this work in accordance with the Copyright, Designs and Patents Act 1988.

First published 1985
Second edition 1992
Reprinted six times
Third edition 1999
MACMILLAN PRESS LTD
Houndmills, Basingstoke, Hampshire RG21 6XS
and London
Companies and representatives throughout the world

ISBN 978-0-333-74013-2 ISBN 978-1-349-14916-2 (eBook)
DOI 10.1007/978-1-349-14916-2

A catalogue record for this book is available from the British Library.

This book is printed on paper suitable for recycling and made from fully managed and sustained forest sources.

10 9 8 7 6 5 4 3 2 1
08 07 06 05 04 03 02 01 00 99

Typeset by EXPO Holdings, Malaysia

and this will reduce the ignition delay period. The modelling of the ignition delay and the subsequent combustion are treated in section 10.2.

To obtain maximum output the peak pressure should occur about $10\text{--}20^\circ$ after top dead centre. Sometimes the injection is later, in order to retard and to reduce the peak pressure. The fuel ignites spontaneously at many sites, and produces an intense flame. There is significant radiation from the flame front, and this is important for vaporising the fuel. Since the combustion occurs from many sites, compression ignition engines are not susceptible to cyclic variation or cyclic dispersion.

Compression ignition engines can operate over a wide range of air/fuel mixtures with equivalence ratios in the range 0.14–0.90. The power output of the engine is controlled by the amount of fuel injected, as in the combustion region the mixture is always approximately stoichiometric. This ensures good part load fuel economy as there are no throttling losses. The fuel/air mixture is always weaker than stoichiometric, as it is not possible to utilise all the air. At a given speed the power output of an engine is usually limited by the amount of fuel that causes the exhaust to become smoky.

3.7 Fuels and additives

The performance and in particular fuel economy of internal combustion engines should not be considered in isolation, but also in the context of the oil refinery. Oil refining and distribution currently has an overall efficiency of about 88 per cent (Francis and Woollacott, 1981). This efficiency would be changed if the demand for the current balance of products was changed. Oil refining can be compared to the work of a butcher; each process has a raw material (a barrel of oil or a carcass) that has to be cut in such a way that all the products can be sold at a competitive price. Just as there are different animals, there are also different types of crude oil, depending on the source. However, the oil refinery can change the type of product in additional ways, such as cracking, although there is an energy cost associated with this.

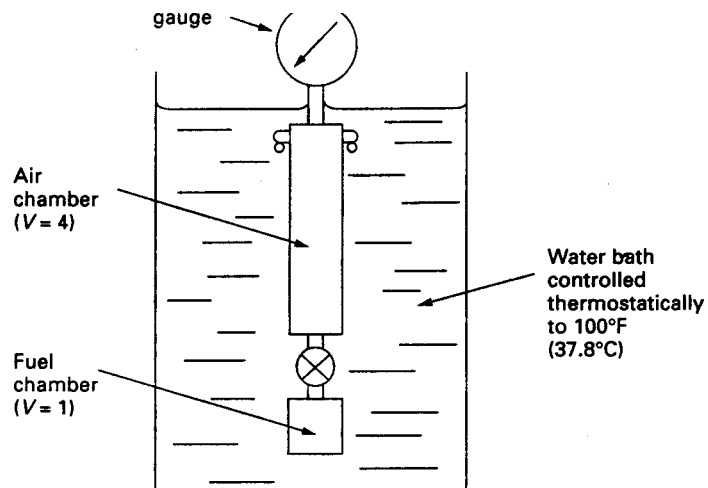
The energy content of a typical petrol is 44 MJ/kg or 31.8 MJ/litre and of a typical diesel fuel 42 MJ/kg or 38.15 MJ/litre; associated with these is an energy content at the refinery of typically 2.7 MJ/kg and 1.6 MJ/kg (Francis and Woollacott, 1981). This gives an effective primary energy of 46.7 MJ/kg or 35.5 MJ/litre for petrol and 43.65 MJ/kg or 39.95 MJ/litre for diesel fuel. There may be circumstances in which it is more appropriate to use primary energy density than to use calorific value. These figures also highlight the difference in energy content of unit volume of fuel. This is often overlooked when comparing the fuel economy of vehicles on a volumetric basis.

3.7.1 Characteristics of petrol

The properties of petrol are discussed thoroughly by Blackmore and Thomas (1977). The two most important characteristics of petrol are its volatility and octane number (its resistance to self-ignition).

Volatility is expressed in terms of the volume percentage that is distilled at or below fixed temperatures. If a petrol is too volatile, when it is used at high ambient temperatures the petrol is liable to vaporise in the fuel lines of carburetted engines

Reid vapour pressure apparatus (from Goodger, 1975).



and form vapour locks. This problem is most pronounced in vehicles that are being restarted, since under these conditions the engine compartment is hottest. If the fuel is not sufficiently volatile the engine will be difficult to start, especially at low ambient temperatures. The volatility also influences the cold start fuel economy. Spark ignition engines are started on very rich mixtures, and continue to operate on rich mixtures until they reach their normal operating temperature; this is to ensure adequate vaporisation of fuel. Increasing the volatility of the petrol at low temperatures will evidently improve the fuel economy during and after starting. Blackmore and Thomas (1977) point out that in the USA as much as 50 per cent of all petrol is consumed on trips of 10 miles or less. Short journeys have a profound effect on vehicle fuel economy, yet fuel consumption figures are invariably quoted for steady-state conditions.

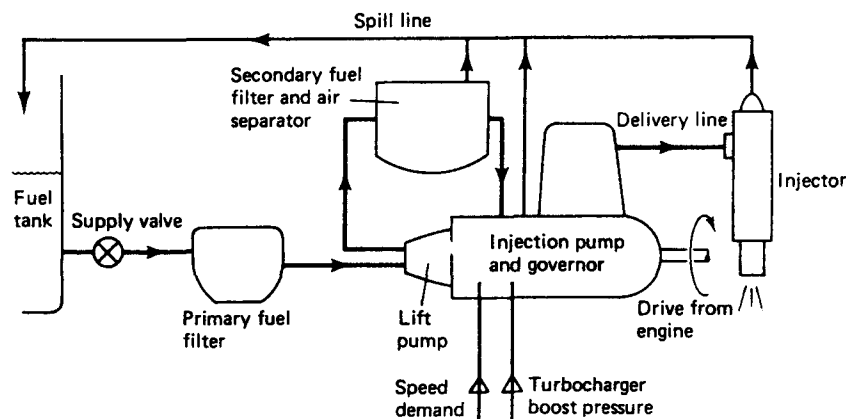
The European Standards Organisation (CEN) has defined a common standard for its 18 member countries, and for unleaded gasoline this standard has the designation EN 228: 1993, and this includes eight different volatility specifications. Members of CEN can then use their national standards (BS EN 228 in the case of the UK), to specify particular volatilities for different seasons. The volatility is also expressed in terms of the Reid Vapour Pressure (RVP). The Reid apparatus is shown in figure 3.12, and it consists of a fuel chamber connected to an air chamber of four times its volume. The fuel chamber is filled with chilled fuel, connected to the air chamber, and then immersed in a water bath at 37.8°C and shaken. The method is defined in ASTM Procedure D323. The final pressure has to be corrected for atmospheric pressure and the effect of heating the air in the air chamber, and the result is an absolute value.

The fuel volatility specified in BS EN 228: 1993 is compared with typical fuel specifications from Blackmore and Thomas (1977) in table 3.3. This is plotted with further data in figure 3.13.

Table 3.3 shows how the specification of petrol varies to suit climatic conditions. Petrol stored for a long time in vented tanks is said to go stale; this refers to the loss of the more volatile components that are necessary for easy engine starting.

The octane number of a fuel is a measure of its anti-knock performance. A scale of 0–100 is devised by assigning a value of 0 to n-heptane (a fuel prone to knock),

Figure 5.11
Fuel injection system for a
compression ignition
engine.



injection systems have a high-pressure fuel pump which produces a controlled and steady pressure, and the injector has to control the start and end of injection. Charlton (1998) provides a comprehensive overview of all types of electronically controlled fuel injection equipment.

After the overview in section 5.5.1 there is a discussion of the fuel injection nozzles (section 5.5.2), since these are common to all types of fuel injection system.

5.5.1 Injection system overview

A traditional pump-line-injector (PLI) system is shown in figure 5.11. Before discussing this, it is important to realise that the low-pressure part of the system is common to all diesel engines.

In general the fuel tank is below the injector pump level, and the lift pump provides a constant-pressure fuel supply (at about 0.75 bar) to the injector pump. The secondary fuel filter contains the pressure-regulating valve, and the fuel bleed also removes any air from the fuel. If air is drawn into the injection pump it cannot provide the correctly metered amount of fuel. It is essential to remove any water or other impurities from the fuel because of the fine clearances in the injection pump and injector. The injection pump contains a governor to control the engine speed. Without a governor the idle speed would vary and the engine could over-speed when the load on the engine is reduced.

The injection pump is directly coupled to the engine (half engine speed for a four-stroke engine) and the pump controls the quantity and timing of the fuel injection (figure 5.12). The quantity of fuel injected will depend on the engine load (figure 5.13). The maximum quantity of fuel that can be injected before the exhaust becomes smoky will vary with speed, and in the case of a turbocharged engine it will also vary with the boost pressure. The injection timing should vary with engine speed, and also load under some circumstances. As the engine speed increases, injection timing should be advanced to allow for the nearly constant ignition delay. As load increases, the ignition delay would reduce for a fixed injection timing, or for the same ignition delay the fuel can be injected earlier. This ensures that injection and combustion are complete earlier in the expansion stroke. In engines that have the injection advance limited by the maximum permissible cylinder pressure, the injection timing can be advanced as the load reduces. Injection timing should be accurate to 1° crank angle.

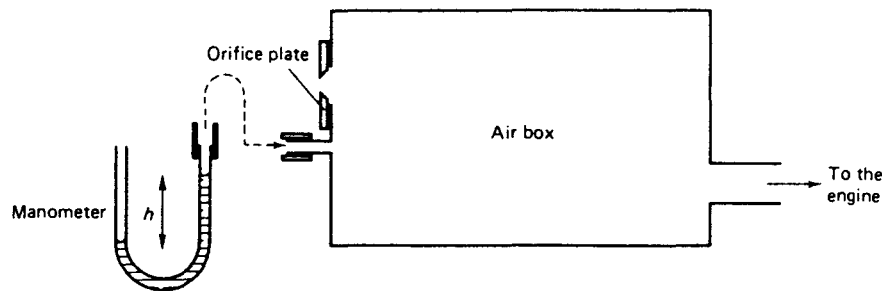


Figure 13.11
Air flow rate tank.

where $\dot{m}_a$ = mass flow rate of air (kg/s)
 C_d = discharge coefficient of the orifice
 A_o = cross-sectional area of the orifice (m)
 g = acceleration due to gravity (m/s²)
 h = height difference between liquid levels in the manometer (m)
 ρ_f = density of manometer fluid (kg/m³)
 ρ_a = density of air = $(p/R_a T)$ (kg/m³).

The accuracy depends on knowing the discharge coefficient for the orifice; this should be checked against a known standard.

In practice, it can be very difficult to make the air box work satisfactorily. Apart from making the air box act as a Helmholtz resonator (chapter 6, section 6.6.2), coupling a long tube to the inlet of an engine can have a significant effect on the engine performance. It should be remembered that even without a tuned induction system, there can be flow reversals occurring at the air inlet to the engine. A way of minimising any coupling effects and checking for their effect is shown in figure 13.12. If the flexible pipe is of large enough diameter, and the air entry protrudes far enough into the pipe, then the unsteady flow sensor will show no change to the air flow pattern when the flexible pipe is connected. The unsteady flow sensor is most likely to be a form of hot wire anemometer (see chapter 8, section 8.2.2).

An alternative approach is the viscous flowmeter. As no damping is required it is much more compact than an air box. To obtain viscous flow a matrix of passages is used in which the length is much greater than a typical diameter. Since the flow is viscous the pressure drop is proportional to the velocity or volumetric flow rate. The flowmeter has to be calibrated against a known standard.

However, in its usual form the viscous flowmeter is not well suited to pulsating flows. The instantaneous pressure drop will be a function of the unsteady flow,

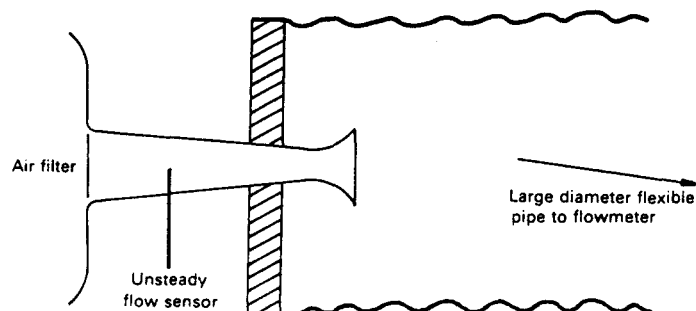


Figure 13.12
Connection of an air
flowmeter to minimise any
effects on engine
performance.

For example, if 300 cycles of a four-stroke engine are to be recorded from two channels, with readings every $\frac{1}{2}^\circ$ and 2 bytes are required for each reading (for a 12-bit reading), then about 3.4 MB of memory will be required.

The trade-offs between the sampling rate and the storage capacity have been illustrated by table 13.4 for a particular system. In this case, the 3.4 MB of memory would require the use of the RAM or the hard disk, and the engine speeds corresponding to the maximum sampling rates would be 8333 or 4166 rpm. These two examples illustrate that the system described in table 13.4 is capable of acquiring large amounts of data very quickly. This can then lead to analysis and archiving problems unless the combustion analysis software is quick to run.

There are other issues to be considered with computer-based data acquisition systems. Firstly, when channels are being multiplexed, the channels are not all being read at the same time. For example, table 13.4 shows that the Computerscope system has a delay of 1 microsecond between reading successive channels. This becomes significant with high sampling rates, but it can be corrected for in software that either assigns the correct angle to the reading or interpolates between readings to give the value at a specific crank angle. Secondly, it is essential that the ADC is coupled to a sample/hold circuit. If this is not the case, slight changes in the signal during the analogue-to-digital conversion process can lead to large errors.

When the data are being analysed, it must be remembered that they were originally a continuous analogue signal that has been assigned to a progression of digital levels. For example, when such a signal is differentiated numerically by looking at the difference in successive values, the result is very noisy. A more satisfactory result can be obtained by using a higher-order finite difference approach and smoothing the signal.

Taylor series can be used to represent the signal, and the following example calculates the first derivative, but includes the terms up to the fourth derivative. Consider a function $f(a)$ in which the interval between values is h :

$$\begin{aligned} f(a_{n+2}) &= f(a_n) + \frac{2hf'(a_n)}{1!} + \frac{2^2h^2f''(a_n)}{2!} + \frac{2^3h^3f'''(a_n)}{3!} + \frac{2^4h^4f^{(4)}(a_n)}{4!} + \dots \\ f(a_{n+1}) &= f(a_n) + \frac{hf'(a_n)}{1!} + \frac{h^2f''(a_n)}{2!} + \frac{h^3f'''(a_n)}{3!} + \frac{h^4f^{(4)}(a_n)}{4!} + \dots \\ f(a_n) &= f(a_n) \\ f(a_{n-1}) &= f(a_n) - \frac{hf'(a_n)}{1!} + \frac{h^2f''(a_n)}{2!} - \frac{h^3f'''(a_n)}{3!} + \frac{h^4f^{(4)}(a_n)}{4!} - \dots \\ f(a_{n-2}) &= f(a_n) - \frac{2hf'(a_n)}{1!} + \frac{2^2h^2f''(a_n)}{2!} - \frac{2^3h^3f'''(a_n)}{3!} + \frac{2^4h^4f^{(4)}(a_n)}{4!} - \dots \end{aligned} \quad (13.44)$$

These equations can be combined to eliminate the second, third and fourth derivatives ($f''(a_n)$, $f'''(a_n)$ and $f^{(4)}(a_n)$), to give

$$f'(a_n) = (f(a_{n-2}) - 8f(a_{n-1}) + 8f(a_{n+1}) - f(a_{n+2}))/12h \quad (13.45)$$

However, figure 13.34 shows that the derivative will still be noisy, and it is advantageous to apply smoothing. The following smoothing algorithm for $(2b + 1)$ values is widely used:

$$\begin{aligned} a_n &= \frac{1}{b^2} [a_{n-(b-1)} + 2a_{n-(b-2)} + 3a_{n-(b-3)} + \dots + ba_n \\ &\quad + \dots + 3a_{n+(b-3)} + 2a_{n+(b-2)} + a_{n+(b-1)}] \end{aligned} \quad (13.46)$$

EXHIBIT 1023

UNIVERSITY OF TORONTO



3 1761 01348136 1

UNIVERSITY
OF TORONTO



Digitized by the Internet Archive
in 2007 with funding from
Microsoft Corporation

INTERNAL COMBUSTION ENGINES

TEM
C

INTERNAL COMBUSTION ENGINES

THEIR THEORY, CONSTRUCTION
AND OPERATION

BY
ROLLA C. CARPENTER, M.M.E., LL.D.
AND
H. DIEDERICHS, M.E.

PROFESSORS OF EXPERIMENTAL ENGINEERING, SIBLEY COLLEGE,
CORNELL UNIVERSITY

WITH 379 ILLUSTRATIONS

SECOND EDITION REVISED



NEW YORK
D. VAN NOSTRAND COMPANY

23 MURRAY AND

1909

27 WARREN STS.

95230
16/3/09



Copyright, 1908, by
D. VAN NOSTRAND CO.

The Plimpton Press Norwood Mass. U.S.A.