Detroit Diesel Engines

Detroit Diesel Allison

SALES TECH DATA BOOK III

VOLUME 1

ENGINE APPLICATION INFORMATION

Detroit Diesel Allison
Division of General Motors Corporation

13400 West Outer Drive
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18SA0090-01

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Division of General Motors Corporation

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Detroit Diesel Allison
Division of General Motors Corporation

FUEL SYSTEMS

FOR

SERIES 53, 71, V-71 & 110 DIESEL ENGINES

by

B. D. Zannoth

and

Z. Zylka

Engineering Technical Data Dept.

18 SA 0201

ENGINEERING BULLETIN No. 21

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FUEL SYSTEMS FOR GM DIESEL ENGINES

The fuel system on all G. M. Diesel 2 cycle engines consists of a unit injector for each cylinder and a fuel transfer pump plus filters, fuel lines and fuel tank (see Figure 1, page 2). The gear type fuel transfer pump "transfers" the fuel from the supply tank through the system to the unit injectors. As the pump works with moderate pressure, no high pressure fuel lines are required. The pump circulates an excess supply of fuel through the injectors. This assists in purging the air and prevents vapor lock when using light fuels. It also fills, lubricates and aids in cooling the injectors. The unused fuel is returned to the supply tank via the return line. The fuel system must be properly installed and maintained in order to assure prompt engine starting and efficient engine operation. As the use of satisfactory fuels is of prime importance, the correct fuels for varying conditions are explained in Service Bulletin 7SE181.

In addition to using the correct fuel, other conditions must be given consideration to insure complete satisfaction in engine performance and trouble-free life. The bulk supply of fuel must be properly handled to prevent contamination by dirt and water before it is placed in the engine fuel tank. When the fuel is dispensed from a bulk supply, a water separator and fuel filter should be used between the bulk supply and its pump. Without these precautions, dirt and water can be introduced into the tank resulting in premature overloading of the fuel strainer or plugging of the fuel lines.

Fuel tanks must be made of the correct material, be properly designed and of adequate size and properly located. Piping, valves, and connections must be correctly chosen and properly installed.

<u>The essential elements of a fuel system are:</u>

1. Fuel tank - large tank with submerged fuel spill line.
2. Fuel piping and connections - must be of adequate size to lessen flow restriction and get maximum fuel filter life.
3. Fuel transfer pump - maximum suction 6 in. hg. measured at pump inlet with clean filters.
4. Fuel strainer and filter - use approved filters to get the most service life and efficiency for trouble-free fuel systems, injector life and maximum horsepower.

TYPICAL FUEL SYSTEM

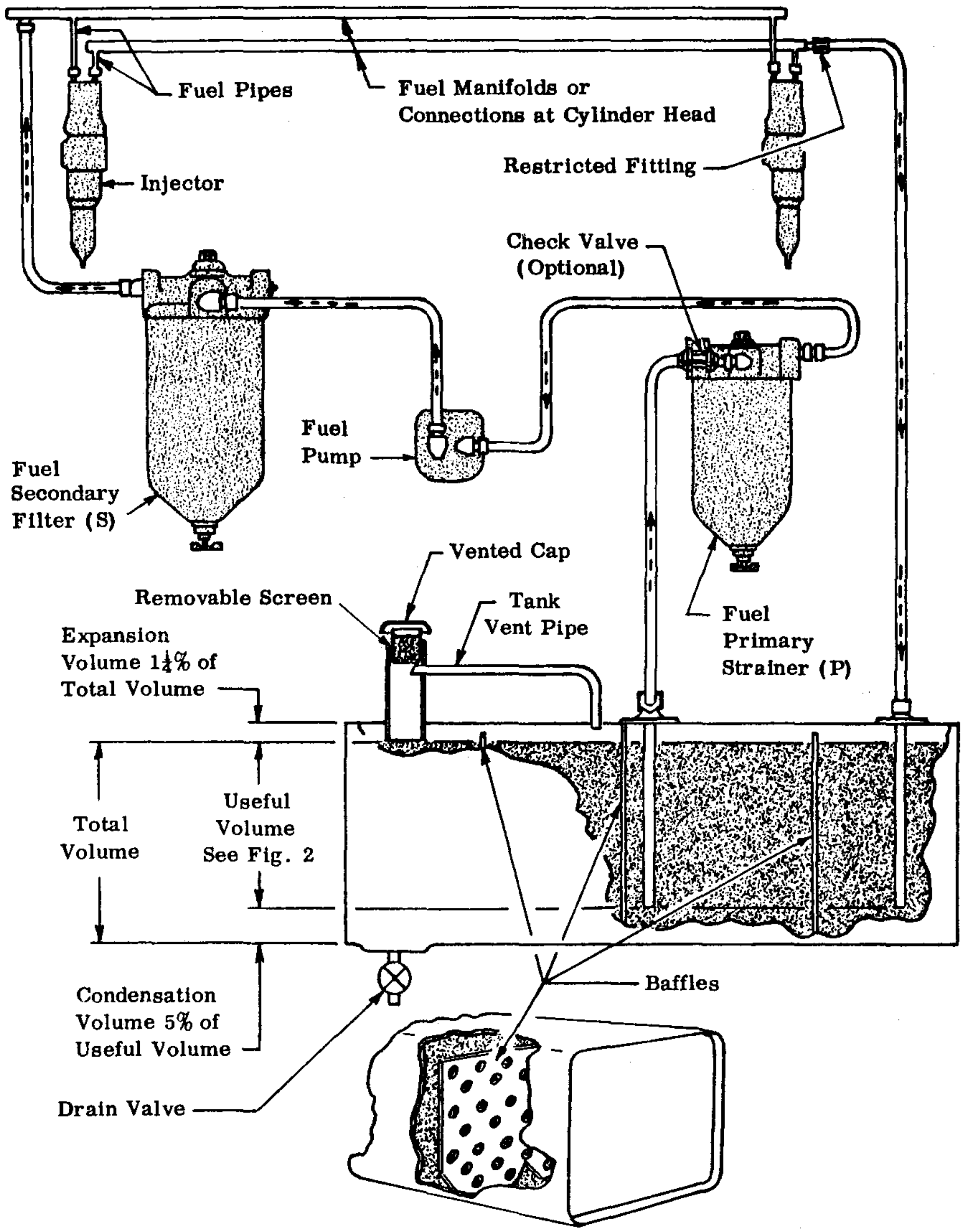


FIG. 1

SUPPLY TANKS

There is no single tank configuration which will be suitable for all installations, the final shape being largely determined by the available space. However, the following basic design points will apply.

The fuel tank should be made of aluminum, monel stainless steel, black iron, welded sheet steel, or reinforced plastic. A galvanized steel tank should never be used as the fuel oil reacts chemically with the zinc coating to form a powdery flaking which quickly clogs the fuel filters and soon causes damage to the fuel transfer pump and injectors. Connections for fuel suction and return lines must be provided and separated as much as possible to prevent recirculation of hot fuel and to allow the separation of any gases entrained in the returned fuel. Approximately 5% of the tank's total volume should be allowed for the accumulation of condensation and contaminants. A low point in the tank should be equipped with a drain valve or plug, in an accessible location, to allow periodic removal of these contaminants. Suitable baffles should be incorporated to reduce excessive surging on mobile units.

The filler neck should be located in a clean, accessible location with sufficient room around it to allow use of an average sized filling can or a hose from a tank truck. A removable wire screen of approximately 1/16 inch mesh should be placed in the filler neck to prevent foreign material from entering the tank.

The filler neck cap should be vented and the tank vented into the neck, preferably from the center top of the tank, to reduce the possibility of air entrapment and fuel blow-back while filling the tank. The cap vent should be approximately 1/8 inch in diameter to maintain atmospheric pressure in the tank at all times and to provide pressure relief in case the fuel expands due to a rise in temperature. An allowance of approximately 1-1/4% of the total fuel volume should be provided for expansion area in the tank. The filler should be such that accidental fuel spillage does not create a fire hazard. By proper venting of the tank into the filler neck it is possible to design a tank which reduces the possibility of being accidentally overfilled. Figure 1 illustrates a tank incorporating the above mentioned features. On Marine installations, when it is desired to use a non-vented filler neck cap, a separate large vent may be incorporated in the fuel tank and piped overboard. The vent pipe must not have any "U" bends which can trap fuel and prevent proper venting of the tank.

The tank should meet all legal requirements and Interstate Commerce Commission regulations. Marine installations should comply with the National Fire Prevention Association Bulletin #302.

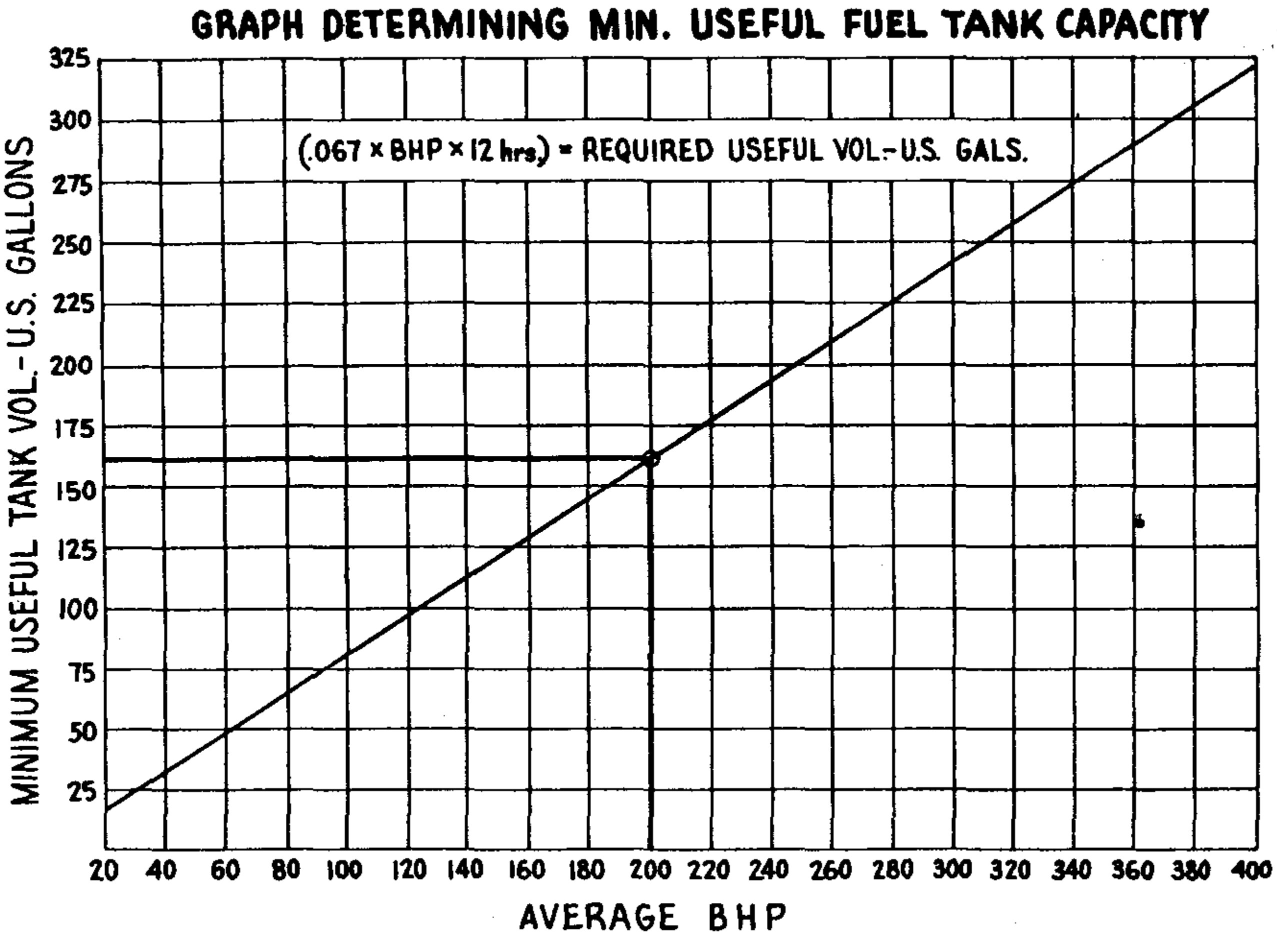


FIG. 2

The tank should be of sufficient useful capacity to allow operation for at least one full working shift under average load conditions. Figure 2 will assist in rapidly determining the minimum useful capacity for which the tank should be designed and is based upon a 12 hour working period. Note, in Figure 2, that for an average bhp of 200, the tank should have a minimum useful capacity of approximately 161 gallons. If the working period is only 8 hours, the fuel capacity will be 2/3 of that shown. If the average bhp, due to use of a larger engine or unit, exceeds 400 bhp but is less than 800 bhp, read 1/2 of the average bhp on the bhp scale and then multiply the useful fuel volume by 2 to arrive at the required fuel volume.

Example: Average bhp - 540 ÷ 2 = 270
270 bhp requires 217 gallons
217 x 2 = 434 useful gallons required for 12 hours

Note: The constant of .067 gallons/BHP is based on an anticipated average fuel consumption.

FUEL SYSTEM ARRANGEMENTS

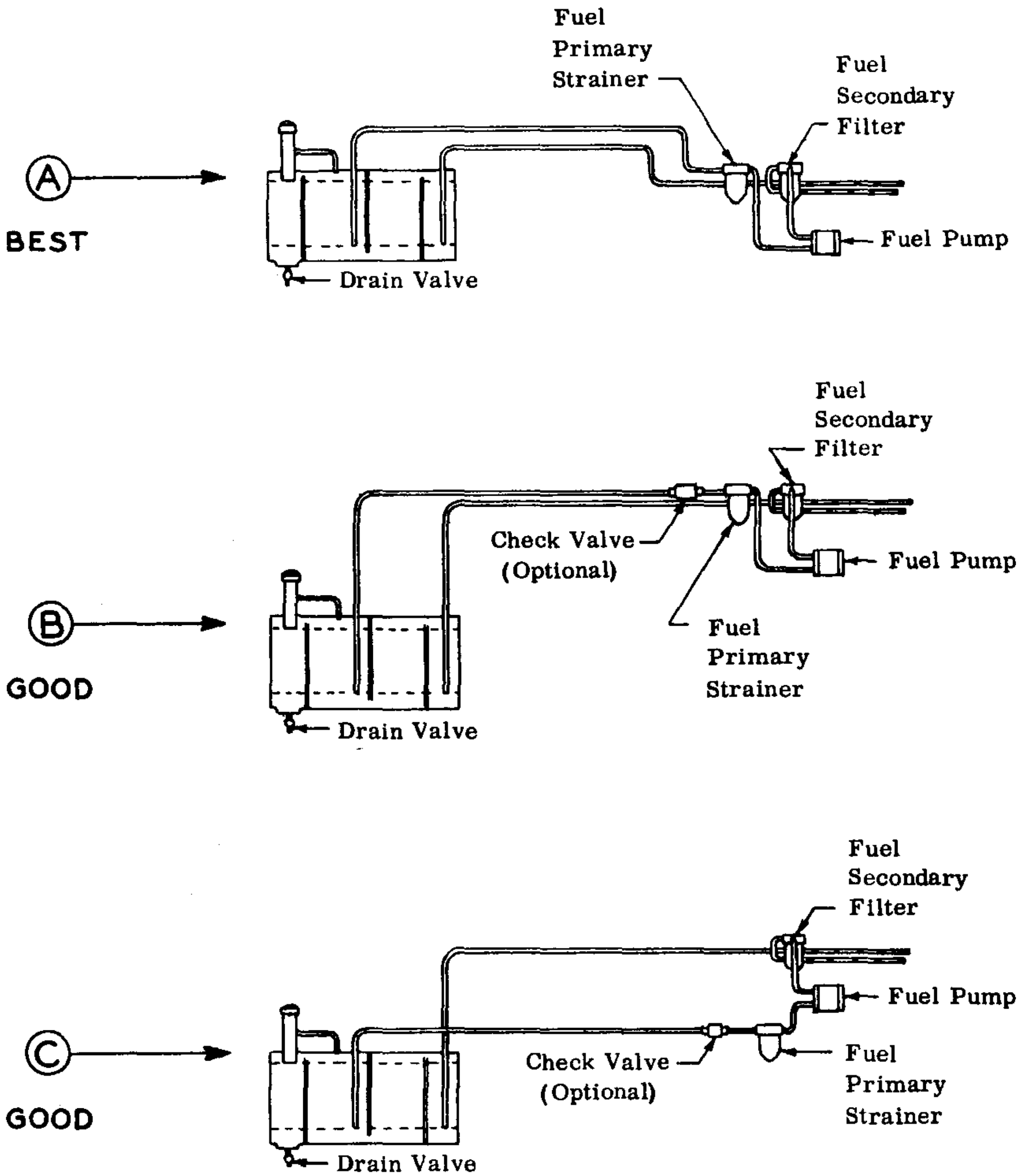


FIG. 3

FUEL SYSTEM ARRANGEMENTS

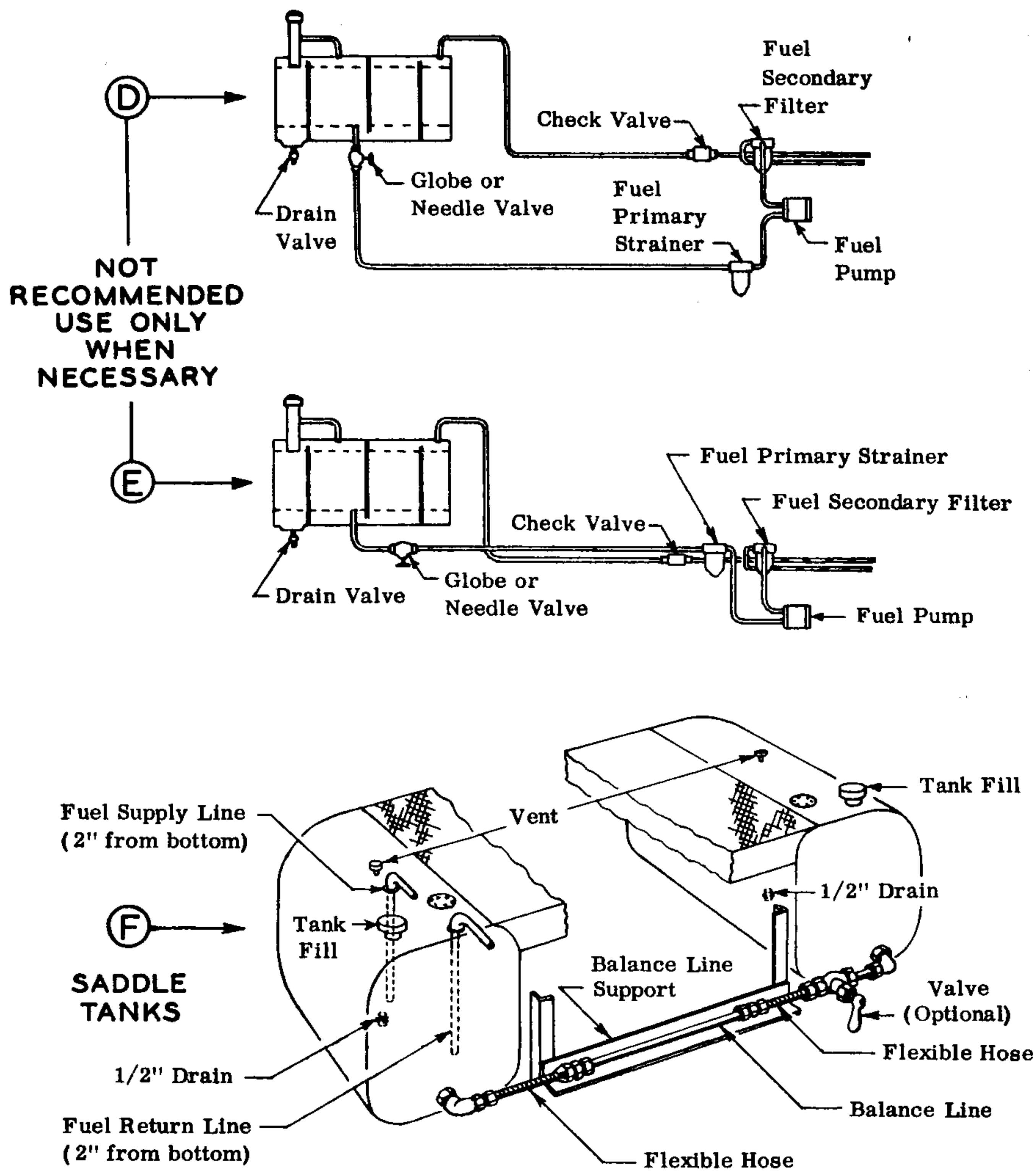


FIG. 4

FUEL SYSTEM ARRANGEMENTS

The position of the fuel tank may be determined by convenience, or standard industry practice, providing that position will meet the conditions of a few typical installations shown in Figures 3 and 4, pages 5 and 6.

View A, Figure 3, illustrates the preferred location of the fuel tank, primary strainer and secondary filter. Note that the minimum level of the useful fuel volume is at the same level as the fuel pump and the fuel suction and return lines are of equal length and extend below the fuel at the minimum level. The lines are widely separated in the tank, with the suction line at the tank center to compensate for angularity operation. Included are baffles, a vent from center top of the tank to the filler neck, and a low point for contaminant drainage.

View B is similar to A, except that the fuel tank is mounted lower than the fuel pump. Here it is imperative that the supply and return line both extend below the fuel at minimum level; otherwise, drain back from the pump and filters will occur during periods when the engine is not operating. An optional but less desirable method of preventing drain back is to install a fuel check valve at the inlet to the primary fuel strainer. Check valves are objectionable because they add restriction to the fuel flow system. If a check valve is used, it must be of the type approved by DDED.

View C is similar to B except that the primary strainer is mounted lower than the fuel pump.

Views D and E illustrate systems with the fuel tank mounted higher than the engine and are not recommended. However, when their use is unavoidable, the fuel return line should not extend into the fuel supply (so that siphoning cannot occur in case a leak occurs in the line) and must incorporate a check valve; the fuel inlet must incorporate a shut-off valve of the needle or globe type construction and must not impose any undue restriction to fuel flow. If these precautions are not observed, there is the possibility of fuel leaking into one or more cylinders, creating a hydraulic lock and resulting in severe damage to the engine upon cranking. The fuel pump with reversed rear seal must be used to prevent seal leakage which will result from the pressure created by the elevated tank. These pumps are identified with a star stamped on the fuel pump cover.

View F illustrates truck mounted saddle tanks. It is desirable to connect the two tanks with at least a 3/4 I.D. crossover line to minimize the problems encountered with plugging and freezing water. The fuel return or spill line should either enter the tank from the bottom or be in the form of a tube extending from the top inlet to the bottom. This is desirable to prevent the fuel from draining from the filters when the engine is not operating. A flexible tube is desirable to prevent breakage caused by vibration.

PIPING, CONNECTIONS AND VALVES

The matter of designing and installing the fuel transfer system should not be taken lightly. It is very important to the operation of the engine and directly affects starting, full power operation, filter life and fuel leakage even when the engine is not in operation. Therefore, considerable care must be exerted in the selection of lines, fittings and their positioning if an acceptable and trouble free fuel system is to be attained.

A desirable fuel system is one with a low restriction to fuel flow on the suction side of the fuel transfer pump. This permits sufficient flow to get full engine power and assures maximum service life from the filters. The maximum suction with clean fuel filters for a complete system at top no load speed is 6 in. hg. measured at the inlet to the fuel pump. This will be with clean fuel filters. Loss of power will occur when the primary fuel filter becomes plugged and the pump inlet restriction approaches 12" hg. It should be apparent that the lower the initial system restriction, the longer will be the primary filter life. Fuel filters and pumps are available with 3/8 N.P.T. holes to minimize restriction.

Fuel lines may be either steel or flexible hose. Copper tubing will become brittle due to "cold working" when subjected to vibration and is not recommended.

If flexible hose is used, it must have a fuel oil resistant synthetic rubber inner tube, reinforcing inner braid, and a cover resistant to fuel, lubricating oils, mildew, and abrasion. As a minimum requirement, it must be suitable to withstand a maximum suction of 20" of mercury without collapsing, a pressure of 100 psi without bursting and temperatures between minus 40°F and plus 200°F.

When installing, the minimum bend radius must not be less than that recommended by the hose manufacturer, and this is particularly true with lines on the suction side of the pump. All lines must be securely clipped to the engine to allow normal engine movement without abrasion.

When steel tubing is used, the connection between the engine and stationary members must be made with a flexible hose. All fuel line connections must be pressure tight and capable of withstanding a pressure of 100 psi. For durability and service, the double flare or sealastic type fitting is preferred. All pipe thread connections should be coated with a sealant that will withstand fuel oil; White lead compound, or others that may contain abrasive, or are of a hardening type, are unsatisfactory. Care must be exercised when applying sealant so that it is not introduced into the fuel system. Apply a limited amount at least two threads back from the male fitting end. Do not apply sealant to female threads.

Sturdy shrouding or guards must be provided in areas where fuel lines are exposed to stones thrown by vehicle wheels or where they can be stepped on by operating personnel.

Change-over valves between multiple fuel tanks, or shut-off valves in the inlet or return lines, must be of the needle or globe type, must not impose undue restriction to fuel flow and must be airtight.

The interconnection of other machinery, such as torque converters, in series with the engine fuel system is not approved.

Minimum fuel line sizes are shown in Figure 8, page 14. These are general and should be used only for short lines. For a more specific analysis, refer to "Determining Optimum Fuel System Performance" on page 12.

FUEL TRANSFER PUMP

The fuel pump furnishes a continuous supply of fuel for engine operation and injector cooling. The pump delivery is in excess of the combustion requirements, the excess being returned to the fuel tank.

The maximum recommended suction at maximum fuel flow when measured at the pump must not exceed 6 in. hg. This maximum restriction includes all inlet restrictions and fuel lift.

Each foot of fuel lift is equivalent to 0.8 in. hg. restriction.

FUEL FILTERS

Good quality fuel oils suitable for high speed diesel engines are practically free from water, sediment, and foreign matter immediately after being refined. However, during subsequent handling of the fuel, a certain amount of contamination is unavoidable.

For this reason, a positive and combined system of fuel filters is used for maximum service life efficiency.

Two stages of filtration are recommended:

1. The first stage is called the primary strainer (marked with the letter "P" cast in the filter head). This is a depth type strainer designed to filter by absorption and to complement the secondary filter The primary strainer is made of a special carded and graded cotton which will absorb varnish, gum, water, emulsions and coarse dirt without rapid plugging and will not mat or pack. The latter condition results in channeling, rendering the strainer useless. The primary element being on the suction side of the pump must, at the same time, cause only a low restriction to flow. The filtering ability should be about 30 microns. Water should be drained from the strainer periodically to prevent the level exceeding half the height of the element at which point the water will be sucked into the system.

   The primary strainer should be installed in an accessible location between the fuel tank and the fuel pump. If located below the fuel tank level, a shut-off valve must be provided between it and the tank so that the element may be changed without draining the tank. Sufficient room must be allowed for periodic removal of the replaceable element without disturbing other items on the installation. If fuel lines, torque converter lines, air cleaner inlet pipes, or other such items must be removed before the filter element can be reached, there is a good possibility the filter will not be serviced as it should be and plugged elements and excessive engine down time will result.

2. The secondary fuel filter (marked with an'S" cast in the head) supplements the primary strainer by filtering all the fine dirt particles that get past the primary. The secondary filter is made of pleated paper with a filtering ability of 10 microns (.0004 in.). It is a surface type filter rather than a depth type, as the primary. This filter must be carefully selected to get high efficiency coupled with long life - a combination not available in all filters.

The filter element must be capable of handling the flow of the fuel pump and be able to withstand a pressure of 75 psi. The filter bowl must contain a sump from which accumulated sediment can be drained at regular intervals. Caution: If the fuel filter is located above wiring harnesses, starters or other electrical equipment, care should be taken to shield this equipment during filter changes, since fuel oil may permanently damage electrical insulation and cause a fire hazard.

The filter should be in an accessible location, close to the fuel manifold. The top of the filter should not project above the level of the fuel connectors in the cylinder head; otherwise, there will be danger of draining a portion of the filter content into the engine when changing injectors. As on the fuel strainer, sufficient room must be allowed for periodic removal of the replaceable element without disturbing other items on the installation.

On installations where the engine must operate beyond the normal element change period, and it is not permissible to stop the engine for filter element changes, two of the fuel strainers and fuel filters may be installed in parallel, by means of three-way valves, and used alternately.

Since plugging of the fuel strainer or fuel filter will interrupt the fuel flow to the engine, with resulting power loss, the elements must be replaced before this occurs. Filter element life depends upon their capacities and the degree of fuel contamination. Some fuel oils may contain sludges which rapidly clog the filter elements necessitating their earlier replacement. A spring loaded by-pass valve between the fuel filter inlet and outlet connections should never be used as the unfiltered fuel will damage the injectors. The 71, 53 and 110 series engines require a minimum fuel inlet pressure (as shown in the chart below) for satisfactory operation. The strainer and filter elements must, therefore, be changed frequently to prevent the pressure dropping below these figures.

MINIMUM FUEL INLET PRESSURES*
*Measured at outlet of secondary filter.

ENGINE	OPERATING SPEED RANGE(RPM)	MINIMUM MANIFOLD PRESSURE-PSI
2-71	1200-2000	40
3,4 & 6-71	1600-2300	45
All V-71	1600-2300	45
All 53	1600-2800	45
6-110	1600-2000	55

FIG. 5

INJECTOR FILTERS

Fine metal filters are used to further protect the injectors. To prevent dirt from entering the injector, these filters should not be removed while the injector is in the engine. Under normal conditions, these filter elements need no attention, but should be inspected and replaced whenever the injector is serviced. If these elements are cleaned and reused, they must be reinstalled in the same relative position from which they were removed; that is, the element from the fuel return side of the injector must never be reinstalled in the fuel inlet side or the resulting reversal of fuel flow will wash accumulated dirt particles into the injector with resulting damage.

Injectors should never be interchanged between series 71 engines having the external fuel manifolds and those having the fuel manifolds integral with the cylinder head unless new injector filters are installed. Likewise, reversal of fuel flow will result when interchanging injectors between the 2-71 engine and all other series 71 engines having the integral fuel manifold. This reversal of fuel flow is not likely to happen on the series 53 engines, unless the fuel system is deliberately piped in reverse as these engines are built only with integral fuel manifolds and plainly marked as to which is the "in" and"out" connection. The series 110 engine has external fuel manifolds only and no fuel flow reversal should result unless the fuel flow through the manifolds is reversed.

SPECIAL PRECAUTIONS

An increase in fuel inlet temperature above 90°F will result in a bhp loss of approximately 2% per 20°F increment fuel temperature rise.

On applications where available space limits the size of the fuel tank and/or where the tank is exposed to high ambient temperature, a fuel cooler may be necessary to maintain a reasonable fuel supply temperature. A typical fuel cooler installation for marine engines is shown in (Fig. 6, page 12).

On applications which are subjected to a wide range of ambient temperature, a fuel must be selected which has a"cloud point" (the temperature at which wax crystals begin to form in the fuel) below the lowest expected fuel temperature, to prevent clogging of the fuel filters by the wax crystals. If the proper selection of fuel can't be assured, it may be necessary to provide a heater which will maintain the fuel temperature above the "cloud point".

All fuel systems must be checked to determine that the flow per engine of the return fuel to the tank is not less than that shown in Fig. 7, page 12, and also that the fuel return line restriction is as recommended.

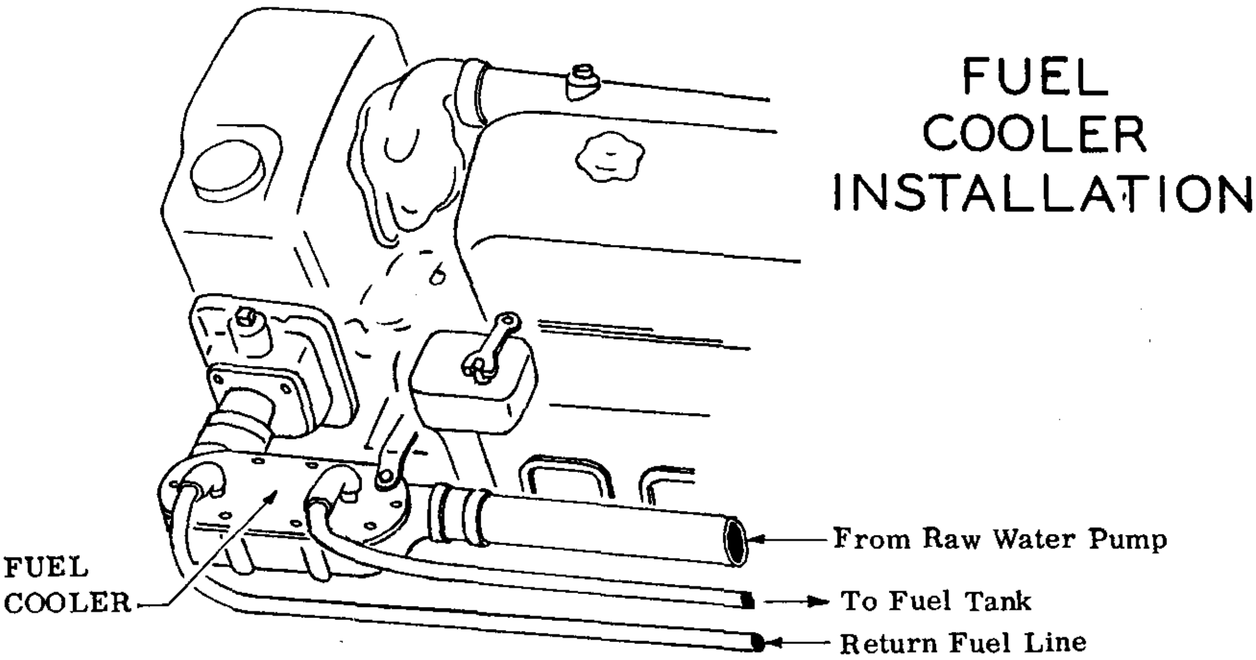


FIG. 6

ENGINE SERIES	SPILL FLOW AT 1200 RPM U.S. GAL/MIN. MINIMUM	RECOMMENDED SPILL ORIFICE SIZE
53	0.6	.070"
2-71 2-71 (Reefer)	0.5 0.4	.055" .040"
6V, 8V & 12V-71 3, 4, 6-71 Twin 71 & Quad 71	0.8	.080"
12V-71	1.2	.106"
16V-71	1.2	.070"
Turbocharged 71 with 90 mm^3 Injectors and All 6-110	0.8	.106"

FIG. 7

DETERMINING OPTIMUM FUEL SYSTEM PERFORMANCE

For optimum fuel system performance, the fuel line piping should be flow designed (designed with minimum resistance to fuel flow) for the specific engine application. Therefore, considerable fuel line planning is required if the fuel pump inlet restriction of 6" hg. maximum for a new system is to be achieved. The desired piping and components can be determined through trial and error methods or can be calculated to an approximate value if appropriate flow and restriction data are available. The latter is more practical. Fuel flow rates and flow restrictions for common components in the fuel inlet systems are provided in Figures 9, 10, 11 and 12 to assist in calculating the total restriction at the pump inlet.

The calculated fuel pump inlet restriction for a typical engine application, with the fuel pump located above the fuel tank (similar to View B, page 5), would be determined as follows:

Maximum fuel flow (6V-71 engine, Figure 9, page 14) 90 gal. per hr.

Maximum fuel pump inlet restriction, 6 in. hg.

Each foot of fuel lift is equivalent to 0.8 in. hg. restriction.

Minimum recommended fuel inlet line size (Figure 8, page 14), 1/2"

BREAKDOWN OF INLET SYSTEM BY INDIVIDUAL COMPONENTS			
COMPONENT DESCRIPTION	RESTRICTION (In. Hg.)	TOTAL UNITS	TOTAL RESTRICTION (In. Hg.)
1/2" steel tubing (.444" I.D.) (Fig. 10, page 15)	.134 in. hg/ft	18 ft.	2.41
137425 90° elbow (1/2" tube, 3/8" NPTF) (Fig. 11, page 16)	.43 ea.	7	3.01
137409 Connector (1/2" tube, 3/8" NPTF) (Fig. 11, page 16)	.03 ea.	2	.06
6435875 Primary strainer (8-1/2" element) (Fig. 12, page 17)	1.32	1	1.32 ea.
Fuel Lift (1.0 ft.) (Page 9)			.80
Total Restriction at Fuel Pump Inlet			7.60 in. hg.
Maximum Allowable Restriction at Fuel Pump Inlet			6.00 in. hg.
Excessive Restriction (unsatisfactory)			1.60 in. hg.

SOLUTION 1 Replace 90° elbows with connectors and additional tube bends.	Remove 5 elbows and decrease restriction	2.15 in. hg.
	Add 5 connectors and increase restriction	.15 in. hg.
	Add 5 tube bends and increase restriction	.20 in. hg.
	Total decrease in system restriction	1.80 in. hg.
	Revised total restriction at fuel pump inlet	5.80 in. hg.
	A restriction of 5.80 in. hg. is satisfactory (below 6 in. hg.).	
SOLUTION 2 Increase tubing size to 5/8".	A similar result, but most likely more expensive, could be accomplished by changing to a 5/8" steel tube.	

As indicated above, the most significant restrictions are generally due to fittings, valves and piping. Therefore, the maximum restriction of 6 in. hg. can be achieved, in most cases, by varying fittings, pipe sizes, or both.

MINIMUM FUEL LINE SIZES			
	I-53, 2,3,& 4-71 & 6-110	6V & 8V-53, 6-71 Twin I-71 6V, 8V & 12V-71	Quad I-71 16V-71
Supply Line (Min. Tube Size)	3/8	1/2	5/8
Return Line (Min. Tube Size)	5/16	5/16	3/8

FIG. 8

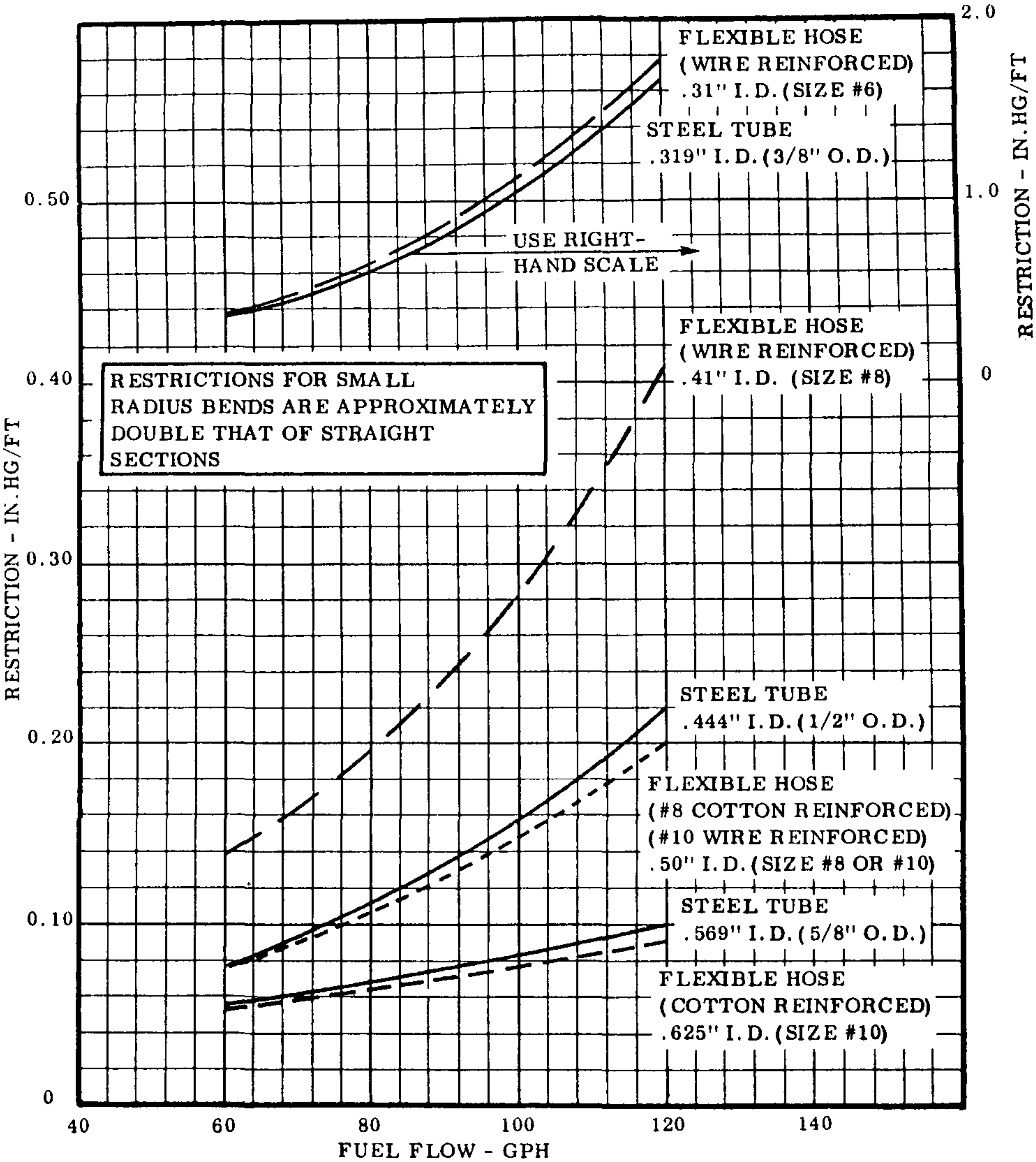
ENGINE FUEL FLOW			
ENGINE	FUEL RETURN ORIFICE	ENGINE SPEED RPM	FUEL FLOW GPH
All 53	.070"	2500-2800	60
2-71 3,4 & 6-71	.055" .080"	1800-2000 1800-2300	60 90
6V, 8V, 12V-71 16V-71	.080" .070"	1800-2300 1800-2100	90 120
12V-71	.106"	1800-2300	120
6-110	.106"	1600-2000	90

FIG. 9

DETROIT DIESEL ENGINE DIVISION

GENERAL MOTORS CORPORATION

HOSE & TUBE RESTRICTIONS
(NO. 2 DIESEL FUEL)



ENGINEERING-TECHNICAL DATA DEPT.

FIG. 10

ELBOW & CONNECTOR RESTRICTION
(NO. 2 DIESEL FUEL)

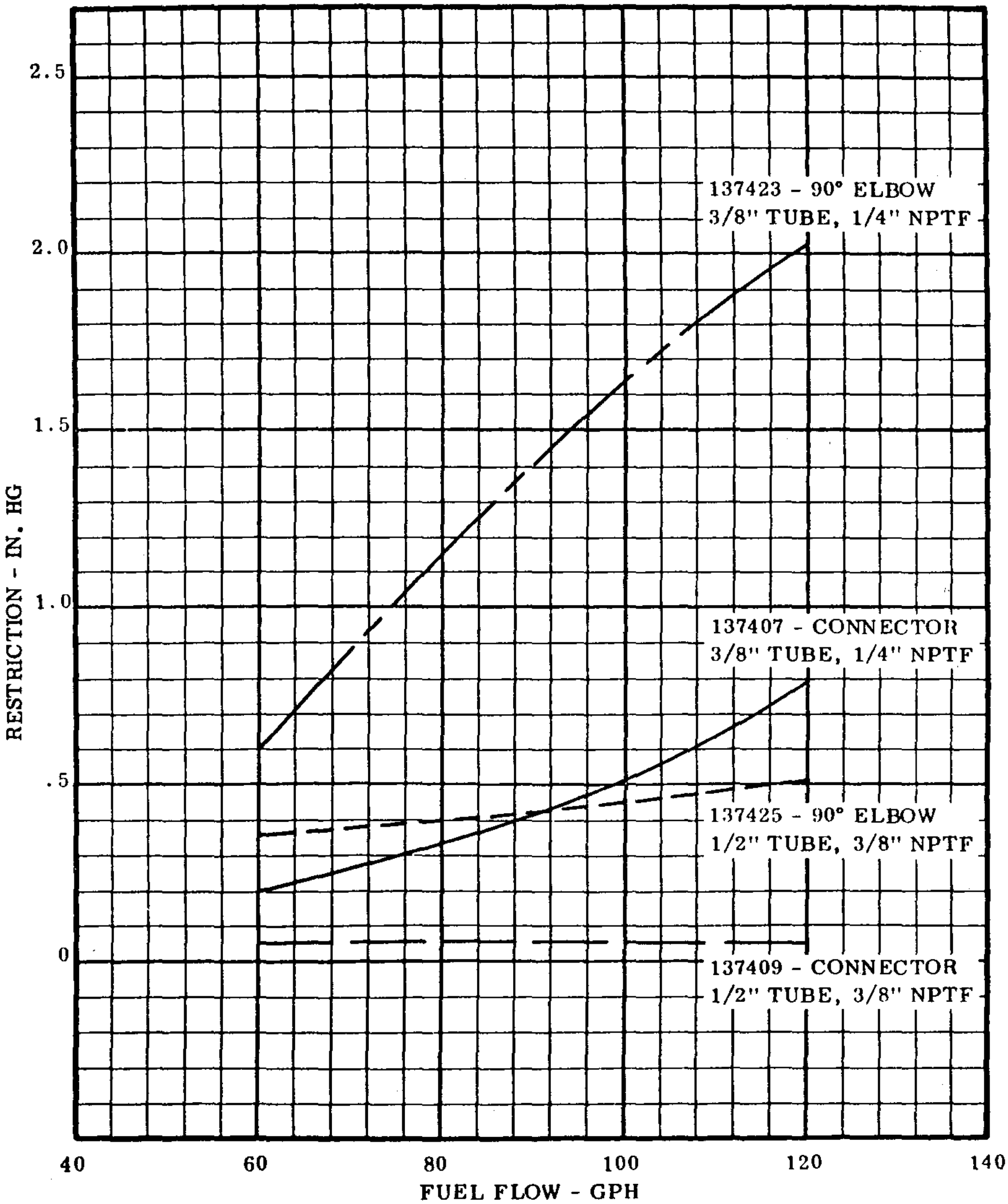


FIG. 11

DETROIT DIESEL ENGINE DIVISION

GENERAL MOTORS CORPORATION

FUEL STRAINER RESTRICTIONS
(NO. 2 DIESEL FUEL)

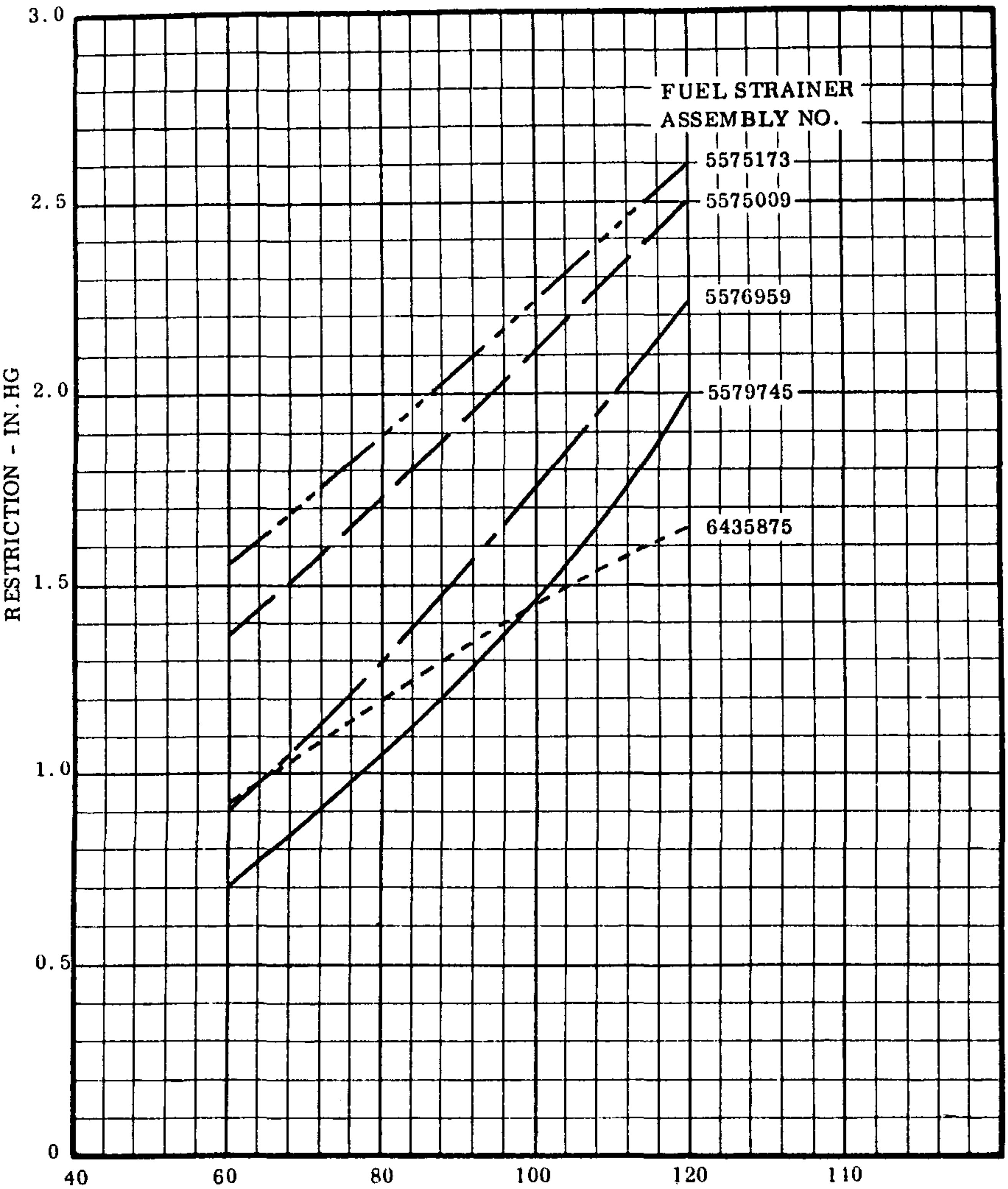


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FIG. 12

FUEL AND LUBRICATING OILS FOR DETROIT DIESEL ENGINES

Giant strides in industry and construction are being met with dependable diesel power under increasing loads, higher speeds, and with temperature ranges extended in both directions. Fuels and lubricants which were satisfactory a few years ago no longer serve today's needs. The intention of this bulletin is to set forth guide lines for the fuels and lubricants best suited to good performance and long life for Detroit Diesel engines.

DIESEL FUEL OILS

Detroit Diesel engines will burn today's quality fuels as clean as or cleaner than competitive engines. Quality is emphasized because the quality of fuel oil used for high-speed diesel engine operation is a very important factor in obtaining satisfactory engine performance, long engine life and satisfactory exhaust.

Fuel selected should be completely distilled material blended from components of minimum aromatic hydrocarbon content. Fuels of high aromatic content usually manifest lower cetane numbers which indicate poorer combustibility of the fuel. In addition, such fuels will exhibit a lower API gravity. Where fuel with API gravity below 34 (at 60°F.) is used and sludging, valve sticking, or other related cold engine operating troubles are encountered, fuel may be the major factor.

Residual fuels and domestic furnace oils are not considered satisfactory for Detroit Diesel engines.

All diesel fuel oil contains an amount of sulphur, but too high a sulphur content results in excessive cylinder wear and harmful acid build-up in the lubricating oil. For most satisfactory engine life, fuels containing less than 0.5% sulphur should be used.

Fuel oil should be from a supply which has been kept clean and free of contamination. Storage tanks should be inspected regularly for dirt, water or water-emulsion sludge, and cleaned if contaminated. Storage instability of the fuel can lead to the formation of varnish or sludge in the tank. The presence of sludge may require sample analysis to determine if the condition is due to bacteria or fungus. In this instance, the storage tank and fuel could be treated with a biocide until the condition is corrected. The presence of varnish or sludge from storage instability must be resolved with the fuel supplier.

DIESEL LUBRICATING OILS

All Diesel engines require heavy duty lubricating oils. Basic requirements of such oils are:

Lubricating Quality
High Heat Resistance
Control of Contaminants

LUBRICATING QUALITY. The reduction of friction and wear by maintaining an oil film between moving parts is a very important requisite of a lubricant. Film thickness, and its ability to prevent metal to metal contact of moving parts, is proportional to oil viscosity. The optimum for Detroit Diesel engines is SAE 30 weight.

HIGH HEAT RESISTANCE. Temperature is the most important factor in determining the rate at which deterioration or oxidation of the lubricating oil will occur. The advent of high speed engines, higher loads, and the added use of oil as a piston coolant have raised oil operating temperatures considerably. The oil should have adequate thermal stability at elevated temperatures thereby precluding formation of harmful carbonaceous and/or ash deposits.

CONTROL OF CONTAMINANTS. The piston and compression rings must ride on a film of oil to minimize wear and prevent seizure with the cylinder wall. At normal rates of consumption, oils reach a temperature zone at the upper part of the piston where rapid oxidation and carbonization can occur. In addition, as oil circulates through the engine it is continuously contaminated by soot, acids and water originating from combustion. Until they are exhausted, detergent and dispersant additives prevent the deposit of the sludges formed by these contaminants. Detergent and dispersant additives aid in keeping sludge and varnish from depositing on engine parts. But, such additives in excessive quantities can of themselves result in detrimental ash deposits. If abnormal amounts of insoluble deposits form, particularly on the piston in the compression ring area, early engine failure may result.

Oil that is carried up the cylinder liner wall is normally consumed during engine operation. The oil and additives leave carbonaceous and/or ash deposits when subjected to the elevated temperatures of the combustion chamber. The amount of deposits is influenced by the oil composition, additive content, engine temperatures, and oil consumption rate.

OPERATION AND MAINTENANCE

In addition to the use of quality fuels and lubricants, two more factors contribute to satisfactory performance and long life in service. They are: Good Operating Practices and Proper Maintenance. These subjects are treated fully in the operators and maintenance manuals. Specifically, an effort should be made to avoid engine idling and low operating temperatures.

Also, a qualified serviceman should check the condition of the pistons, rings, and liners periodically through the AIR BOX COVERS. Rings must float freely in the piston ring grooves. A dark discolored ring should be checked for sticking. A slight pressure applied through a port in the liner can test the freeness of the ring in the groove. A blunt nosed brass rod should be used rather than a sharp instrument which might mar the ring and cause scoring. If a ring appears to be scuffed or discolored, or there is little or no movement of the ring in the groove, your serviceman should check the compression and follow the maintenance manual for corrective measures.

Generally, running conditions will vary for each operator even with comparable mileage or hours and, therefore, maintenance schedules can vary. A good rule of thumb for piston, ring and liner inspections, however, would be at 45,000 miles or 1500 hours for the first such inspection and at 30,000 miles or 1000 hours intervals thereafter.

Another excellent preventive maintenance practice is a regularly scheduled testing of fuel and lubricating oils by either the oil supplier or an independent testing laboratory.

Since the oil supplier knows the physical properties of his products best and maintains laboratories to determine practical oil drain intervals, take advantage of this service and request him to check drained oil samples frequently and report the results to you.

DIESEL FUEL OIL SPECIFICATIONS

The quality of the fuel oil used for high-speed diesel engine operation is a major factor in satisfactory engine performance and life. The fuel oils selected must be clean, completely distilled, stable, and non-corrosive. Enlist the aid of your supplier in obtaining proper fuel oil. The responsibility for clean fuel lies with the fuel supplier as well as with the operator.

DISTILLATION RANGE, CETANE NUMBER, AND SULFUR CONTENT are three of the most important properties in the selection of diesel fuels for optimum combustion and minimum wear. Engine speed, load, and atmospheric temperature influence the selection of the fuels with respect to distillation range and cetane number. THE SULFUR CONTENT OF THE FUEL MUST BE AS LOW AS POSSIBLE, to avoid excessive deposit formation and premature wear.

Diesel fuels are generally marketed according to ASTM DESIGNATION D975 and only distillate fuels No. 1D and 2D are considered satisfactory for Detroit Diesel engines. Residual fuels and furnace oils, generally, are not considered satisfactory for Detroit Diesel engines. In some regions, however, fuel suppliers may distribute only one fuel that is marketed as either diesel fuel (ASTM D-975) or domestic heating fuel (ASTM D-396). In this case, the fuel should be investigated to determine whether the physical properties conform with those shown in the Fuel Oil Selection Chart.

As a guide to the selection of the proper fuel oil for various applications refer to the ASTM Classification Chart and the Fuel Oil Selection Chart.

ASTM Classification of Diesel Fuel Oils

	No. 1-D	No. 2-D
Flash Pt.; °F Min.	100	125
Carbon Residue; %	0.15	0.35
Water and Sediment; (% by Volume) Max.	Trace	0.10
Ash; % by Wt.; Max.	0.01	0.02
Distillation, °F 90% Pt.; Max.	550	640
Min.	-	540
Viscosity at 100°F; centistokes Min.	1.4	2.0
Max.	2.5	4.3
Sulfur; % Max.	0.5	0.5
Cetane No; Min.	40	40

Engine operation at altitudes above 5000 feet requires use of next lighter class of fuel oil than would normally be used.

During cold weather engine operation, the "cloud point" (the temperature at which wax crystals begin to form in the fuel oil) should be 10°F. below the lowest expected fuel temperature to prevent clogging of the fuel filters by wax crystals.

At temperatures below -20°F. consult an authorized Detroit Diesel Service Outlet, since particular attention must be given the cooling system, lubricating system, fuel system, electrical system, and cold weather starting aids for efficient engine operation.

FUEL OIL SELECTION CHART

Type of Engine Service	Typical Application	General Fuel Classification	Final Boiling Point (Max.)	Cetane Number (Min.)	Sulfur Content (Max.)
Light load and speed with considerable idling.	City Buses	No. 1-D	550°F	45	0.30%
Light load and speed.	Generator Sets, Industrial and Automotive Equipment in city and suburban operation.	Winter No. 1-D	550°F	45	0.30%
		Summer No. 1-D	600°F	40	0.50%
Medium load and speed.	Marine Pleasure Craft, Tractors, Industrial Equipment.	Winter No. 1-D	600°F	45	0.50%
		Summer No. 2-D*	675°F	40	0.50%
Heavy load and high speed with idling.	Highway Trucks	Winter No. 2-D*	675°F	45	0.50%
		Summer No. 2-D*	675°F	40	0.50%
Heavy load and high speed.	Heavy Duty Off-the-Road Equipment, Trucks, Tractors	No. 2-D*	675°F	45	0.50%

*NOTE: For most satisfactory engine life, use only those No. 2-D diesel fuel oils containing 0.50% or less sulfur. Where minimum exhaust smoke is required or where long periods of idling or cold weather conditions below 32°F. are encountered, the more volatile or light distillate fuels are recommended.

BREAK-IN OILS AND ADDITIVES MARKETED FOR FUELS AND LUBRICANTS

The use of proprietary blends of supplementary additive or concentrates such as engine oil supplements, break-in oils, tune-up compounds and friction reducing compounds is not recommended in lubricating oils used in Detroit Diesel engines unless given official Detroit Diesel part numbers and made available for use in appropriate service applications. This also applies to the use of metal containing diesel fuel additives.

DIESEL LUBRICATING OIL SPECIFICATIONS

OIL QUALITY

There are hundreds of commercial crankcase oils marketed today. Lubricants marketed for heavy duty diesel service consist of refined crude oil to which has been added additives compounded to meet the desired engine performance levels. Oil additive selection is based on evaluations conducted by the oil supplier; therefore, satisfactory OIL QUALITY is the responsibility of the oil supplier. (The term oil supplier is applicable to refiners, blenders and rebranders of petroleum products, and does not include distributors of such products.) Experience has shown that oil performance in commercial heavy duty diesel service applications varies from brand to brand.

Obviously engine manufacturers or users cannot completely evaluate the hundreds of commercial oils; therefore, the selection of a suitable lubricant in consultation with a reliable oil supplier, strict observance of his oil change recommendations (used oil sample analysis can be of value), and proper filter maintenance will provide your best assurance of satisfactory oil performance.

Detroit Diesel lubricant recommendations are based on general experience with current lubricants of various types and give consideration to the commercial lubricants presently available.

RECOMMENDATION

MIL-L-2104B Lubricants

Detroit Diesel engines have given optimum performance, and experienced the longest service life with MIL-L-2104B, SAE 30 oils. However, the additive concentration of some MIL-L-2104B oils has been substantially increased in order to meet 1968-1969 MS performance requirements. Some of these 1968-1969 MS/MIL-L-2104B oils have given unsatisfactory performance because of excessive exhaust valve and ring-belt ash deposits. For these reasons our primary lubricant recommendations are MIL-L-2104B and Supplement 1 oils with the following limitations:

1. Zinc, as zinc diorganodithiophosphate, between a minimum of 0.07 and a maximum of 0.10 percent by weight.
2. Sulfated ash (ASTM D-874) of 1.00 percent maximum by weight, except lubricants that contain only barium detergent - dispersants where 1.50 percent by weight is allowed.

Contact a reliable oil supplier and obtain his assurance that his product has been tested and given good performance in Detroit Diesel engines. An SAE 30 oil is recommended for year-round use. The use of lower viscosity oils or multigrade products will usually result in less than normal engine life.

MIL-L-45199B (Series 3) Lubricants

The use of Low Ash Series 3 oils (sulphated ash less than 1.65 percent by weight - ASTM designation D-874) may be necessary if the continued use of high sulfur fuel (sulfur greater than 0.5 percent by weight - ASTM D-129) is unavoidable. Consult a reliable oil supplier, obtain assurance that his products have been tested in Detroit Diesel engines, and select the best performer for optimum engine life.

Low ash Series 3 oils do NOT have to meet any specific military low temperature performance requirements; therefore, they may NOT perform as well as MIL-L-2104B lubricants in cold climates.

The older high ash Series 3 oils should NOT be used in Detroit Diesel engines as they tend to deposit heavy ash on valve faces and head inserts resulting in channelling, guttering, and short engine life.

Supplement 1 Lubricating Oils

See MIL-L-2104B limitations under Recommendation.

Multigrade Lubricating Oils

Multigrade oils are NOT recommended. The use of an SAE-30 grade is desirable for year-round use when cold starting can be accomplished. Multigrade oils should be considered only as the "last resort" to facilitate starting when prolonged exposure to temperatures below freezing is unavoidable and adequate starting aids are unavailable.

Experience clearly indicates that multigrade oils are NOT comparable to SAE-30 lubricants for heavy duty diesel service. Cylinder liner scuffing, liner port and ring groove deposit levels are all greater using multigrade lubricants. This results in shortened engine life.

COLD WEATHER OPERATION

Cold weather starting will be facilitated when immersion type electrical coolant heaters can be used. Other practical considerations, such as the use of batteries, cables and connectors of adequate size, generators or alternators of ample capacity, proper setting of voltage regulators, ether starting aids, oil and coolant heater systems, and proper fuel selection will accomplish starting with the use of SAE-30 oil. For complete cold weather starting information, consult an Authorized Detroit Diesel Service Outlet.

OIL CHANGES

It is recommended that new engines be started with 100 hour oil change periods. For highway vehicles this corresponds to approximately 3,000 miles, and for "city-service" vehicles approximately 1,000-2,000 miles. The drain interval may then be gradually increased, or decreased with experience on a specific lubricant while also considering the recommendations of the oil supplier (analysis of the drained oil can be helpful here) until the most practical oil change period for the particular service has been established.

Solvents should not be used as flushing oils in running engines. Dilution of the fresh refill oil supply can occur which may be detrimental.

OIL FILTRATION

Heavy sludge deposits found on the oil filter elements at the time of an oil change must be taken as an indication that the detergency of the oil has been exhausted. When this occurs, the oil drain interval should be shortened. Since abrasive dust, metal particles and carbon material accumulate in the lubricating oil during engine operation, the oil filter elements must be replaced each time the oil is changed.

> NOTE: The manufacturer's warranty applicable to Detroit Diesel engines provides in part that the provisions of such warranty shall not apply to any engine unit which has been subject to misuse, negligence or accident. Accordingly, malfunctions attributable to neglect or failure to follow the manufacturer's lubricating recommendations indicated above may not be within the coverage of the warranty.

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Cooling of Detroit Diesel On-Highway Vehicle Engines

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INTRODUCTION

Detroit Diesel Allison engines, like all other internal combustion engines, reject heat during operation. The heat rejected to the coolant, which is ultimately dissipated to the environment by a radiator, is the subject of this bulletin. Because the design and function of the cooling system directly affects engine performance and life, DDA powered on-highway and on-off highway vehicle cooling systems must comply with the following requirements.

1. COOLING SYSTEM PERFORMANCE REQUIREMENTS

1.1 Cooling Index

The heat dissipation capacity of the cooling system must be sufficient to assure that the engine coolant-out temperature will not exceed 210°F in actual operation.

The ATB indexes, below, have proven to be adequate for the majority of vehicles operated in the United States and Canada. Operation in areas having more severe ambient temperatures require a higher cooling index.

112°F ATB - Line haul, on-off highway, refuse and dump trucks.

104°F ATB - City and suburban pickup and delivery trucks.

For details see Sections 2.4 through 2.4.5

1.2 Deaeration

Every cooling system must have the capability to remove entrained air from the coolant. The system must remove air at the rate specified below. For details see section 9.5.

DDA Specified Induced Air
(Automotive Engines Only)

Engine Model	Test Air To Be Induced CFM
4-53 & 8.2	0.3
6V-53	0.4
6-71	0.5
6V-92	0.6
8V-71 & 8V-92	0.8

1.3 Drawdown Capacity

The cooling system design must be such that a reasonable loss of coolant from the normal full level will not result in aeration of engine coolant. The minimum drawdown for systems having a total capacity between 30 and 120 quarts is 10 per cent of the capacity. Minimum drawdown capacities for other systems are shown on page A-4 in the Appendix. For details see section 9.4.

1.4 Coolant Expansion

There must be provision for at least a 6 per cent expansion of the total system coolant volume; i.e., engine, radiator, and accessory volumes. For details see section 4.1.

1.5 Coolant Pump Suction

The system must not impose more than 3 in. Hg. suction at the water pump inlet with the radiator fill cap removed and the thermostats fully open. For details see section 2.3.2.

1.6 System Fill

The system must be vented so that it can be filled to 90 per cent of the total system coolant volume at a minimum rate of 3 g.p.m. For details see section 9.0.

1.7 System Pressure Regulation

It is recommended that a minimum 9 p.s.i. separate relief valve be used to regulate the cooling system pressure. For details see section 4.8.

2. COOLING SYSTEM DESIGN PARAMETERS

2.1 Heat Rejection

The engine is the major source of heat which must be dissipated by the cooling system. Other heat sources include accessory coolers connected in the engine coolant circuit, such as an automatic transmission oil cooler and accessory radiators which utilize cooling system air flow such as an air conditioning condenser.

2.1.1 Engine Heat Rejection

The heat rejection values for various DDA automotive engines are shown on page A-10 in the Appendix.

2.1.2 Automatic Transmission Heat Rejection

The heat load of an automatic transmission must be considered in the design of an engine cooling system when using either an oil-to-water cooler connected to the engine cooling system, or an oil-to-air cooler using the engine cooling system air flow. The heat load for HT & AT Allison Automatic Transmissions for radiator sizing is 500 BTU/MIN. The transmission installation manual should be referenced for sizing transmission coolers.

2.1.2.1 Oil-To-Water Cooling

When torque converter oil is cooled by engine coolant, the total heat load to the cooling system is the sum of the engine and the converter heat rejection.

1) Total heat rejected (BTU/MIN.) = 42.5 BTU/MIN./HP × (net engine HP) (100-% converter eff.) + engine heat load (BTU/MIN).
2) Total heat rejected (watts) = 1000 watts/KW × (net engine KW) (100-% converter eff.) + engine heat load (watts).

This calculation is normally made at 80% converter efficiency for on-highway applications which is considered to be the worst sustained operating condition normally encountered. Additional oil cooling is required if it is expected that the converter will be operated in a stall condition for more than one minute or if the converter is operated for sustained periods of time at less than 80% efficiency.

Full converter stall heat load may be as much as three times the heat load at 80% converter efficiency. If vehicle wheel slip can occur prior to reaching 80% converter efficiency, then the calculation can be made at the efficiency corresponding to wheel slip.

2.1.2.2 Oil-ToAir Cooling

The use of an oil-to-air cooler in conjunction with the engine radiator will have an adverse effect on the performance of the radiator because of increased air flow restriction. If it is mounted in front of the radiator, it also increases the temperature of the air entering the radiator core. This should be considered when making a radiator core selection.

2.1.3 Hydraulic Retarders

The heat rejected by a hydraulic retarder cooler varies with gross vehicle weight, percent grade, and other particulars of vehicle design and operation. If a hydraulic retarder cooler which utilizes engine coolant is required, consult the factory.

2.1.4 Engine Accessories

Consideration must be given to accessories which might add significant heat to the cooling system. Some accessories, like an automatic transmission or an air conditioning condenser, add a large heat load, while others like a water cooled air compressor add very little heat.

2.2 Engine Coolant Capacity

The approximate engine coolant capacity in quarts (liters) for Detroit Diesel automotive engines is shown on A-1 in the Appendix.

2.3 Engine Coolant Flow

A centrifugal type pump circulates coolant through the engine and radiator. To aid in selecting the proper radiator size, the coolant flow for DDA automotive engines is shown on A-11 and 12 in the Appendix. The curves indicate the coolant flow with the various types of thermostats which are presently being used. Nominal curves are shown; actual flow values may differ due to differences in coolant temperature and system restriction.

2.3.1 Engine Inlet and Outlet Hoses

Connections between the radiator and the engine should be made with the largest practical hose size; unnecessary bends should be avoided. (See Appendix, pg. A-1). Hoses conforming to S.A.E. 20R1 are recommended.

2.3.2 Coolant Pump Suction

To insure proper coolant pump flow, the cooling system restriction must not be more than 3 in. Hg. at the water pump inlet with the radiator fill cap removed and the thermostats fully open.

2.3.3 Temperature Cavitation

Operating above 180 °F (82.2C) coolant out temperature, with a non-pressurized system, will have a tendency to decrease pump flow due to pump cavitation. This factor becomes more critical when the engine is operated at altitude; the figure on A-5 in the Appendix indicates the change in boiling point with an increase in altitude. The temperature at which the coolant begins to boil also varies with the type of coolant and system pressure valve as shown below.

Press. Relief Valve Rating PSI (kPa)		**Boiling Point - °F (°C)** Water	44% Glycol Mix
0 (0.0)	Note A	212 (100.0)	224 (106.7)
4 (27.6)	Note A	222 (105.6)	234 (112.2)
7 (48.3)	Note A	230 (110.0)	242 (116.7)
9 (62.1)	Note B	235 (112.8)	247 (119.4)
12 (82.7)	Note C	242 (116.7)	254 (123.3)
14 (96.6)	Note C	247 (119.4)	259 (126.1)

Note A - Not recommended
Note B - Recommended
Note C - Recommended for altitude

2.3.4 Thermostat Types and Temperature Ranges

Two basic types of thermostats are used in Detroit Diesel engines–full blocking (top bypass) and choke (poppet type).

The full blocking type thermostat simultaneously controls the flow of coolant to the radiator and the bypass circuit. Bypass flow discharges through the top of the thermostat. Radiator flow discharges at the base of the valve sleeve when the thermostat is open. During the engine warm-up, all engine coolant flows through the bypass circuit. As the thermostat opens, increasing amounts of coolant flow to the radiator and the bypass flow is correspondingly reduced. Full blocking thermostats are generally used on 2 cycle DDA "V" engines.

The choke (poppet) type thermostat controls the flow of coolant only to the radiator while the continuously open bypass sends coolant directly back to the engine block. Coolant flow to the radiator discharges through the top of the thermostat valve. Choke type thermostats are generally used in Detroit Diesel in-line engines.

The partial blocking (side bypass) thermostat is a variation of the choke type unit. It has a circular sleeve attached below the valve which moves with the valve. When the thermostat opens to permit coolant flow to the radiator, the sleeve moves to partially block the bypass. Partial blocking thermostats are used on both Detroit Diesel in-line and "V" engines, primarily automotive models.

The standard thermostats used in Detroit Diesel automotive engines begin to open at 180 °F (82.2 °C) and are fully open at approximately 195-197 °F (90.6-91.7 °C).

NOTICE: None of the thermostats or thermostat housings used on Detroit Diesel automotive engines have air vents or bleed holes. *An external deaeration (vent) line must be used in order to fill and deaerate the cooling system.* See 4.1 Top Tank Design.

2.4 Maximum Coolant Temperature

The cooling system must be designed so that the 210 °F (98.9 °C) maximum specified coolant out temperature (see 2.4.2) will not be exceeded under any vehicle operating condition. A system which will not maintain the temperature below this maximum is considered unsatisfactory even though coolant boiling may not occur, since corresponding oil temperatures may be excessive.

2.4.1 High Coolant Temperature Alarm

If a high coolant temperature alarm or warning device is used, its recommended nominal setting is 5 °F (2.8 °C) above the 210°F (98.9 °C) maximum specified engine coolant out temperature.

2.4.2 Cooling Index

The cooling index is a numerical value which expresses the cooling capability of a given system. It can be stated either as air-to-boil (ATB) or air-to-water (ATW). The approved cooling indexes for various Detroit Diesel automotive applications are tabulated in the following chart. These indexes are based on tests conducted with a maximum of 15 MPH ram air applied to the front of the vehicle.

Cooling Index*

500 ft. (152.4m) Altitude

Service	Min. ATB Temp.	Max. ATW-Temp Diff.	Max. Water Out Temp.
Line haul, on/off highway, refuse & dump trucks	112 F (44.4C)	100 F (55.6C)	210 F (98.9C)
City & suburban pickup & delivery	104 F (40.0C)	108 F (60.0C)	210 F (98.9C)

* These cooling indexes apply in the U.S. and Canada. In other countries consult the Detroit Diesel Allison Distributor or Zone office. (See 9.6 for Test Procedure. See 2.4.5 for Affect of Altitude.)

2.4.3 Air-to-Boil (ATB)

The ATB index is the ambient temperature at which the engine coolant out temperature would reach 212 °F (100 °C) (the sea level boiling point of water) under full fuel conditions.

As an example, with an ambient temperature of 100 °F (37.8 °C) and a stabilized coolant out temperature of 197 °F (91.7 °C), the resultant ATB index is 115 °F (46.1 °C).

100 °F ambient + (212 °F-197 °F coolant) = 115 °F

37.8 °C ambient + (100 °C-91.7 °C coolant) = 46.1 °C

The resultant ATB index is *greater than* the 112 °F (44.4 °C) requirement specified in 2.4.2 and is satisfactory. To convert from Air-to-Boil to Air-to-Water differential:

ATB = 212° - ATW

2.4.4 Air-to-Water (ATW)

The ATW differential is the difference between the stabliized engine coolant out temperature and the ambient air temperature under full fuel conditions.

As an example, with an ambient air temperature of 100 °F (37.8 °C) and stabilized coolant out temperature of 197 °F (91.7 °C), the resultant ATW differential is 97 °F (53.9 °C).

197 °F - 100 °F = 97 °F ATW

91.7 °C - 37.8 °C = 53.9 °C ATW

The calculated ATW differential is *less than* the 100 °F (55.5 °C) requirement specified in 2.4.2 and is satisfactory. To convert from Air-to-Water differential to Air-to-Boil:

ATW = 212 °F - ATB

2.4.5 Effect of Altitude

The results of cooling index tests conducted above 500 feet (152m) altitude are affected by the lower ambient air density. Test results will vary by 3.6 °F (2 °C) per 1000 feet (305m) change in altitude and should be corrected to determine if the cooling system is adequate. Tests at 500 feet (152m) altitude, or below, need not be corrected.

For example, a test conducted at a site having an elevation of 9000 feet (2743m) above sea level resulted in an ATB of 97 °F (36.1 °C) with the equivalent ATW of 115 °F (63.9 °C). These values, as they are, do not meet the required ATB or ATW differential. However, at 9000 feet (2743m) above sea level, a correction of 32.4 °F (3.6 °F per 1000 feet) or 18 °C (i.e. 2 °C per 305m) should be applied as follows:

ATB index correction:

97 °F + 32.4 °F = 129.4 °F ATB

36.1 °F + 18 °C = 54.1 °C ATB

The corrected ATB index is *greater than* the 112 °F (44.4 °C) minimum ATB requirement for a highway vehicle application and is satisfactory.

ATW differential correction:

115 °F - 32.4 °F = 82.6 °F ATW

63.9 °C - 18 °C = 45.9 °C ATW

The corrected ATW differential is *less than* the 100 °F (55.6 °C) maximum ATW requirement for a highway vehicle application and is satisfactory.

3. COOLING SYSTEM DESIGN

A cooling system must provide the following under all engine operating conditions:

1. Regulation of engine coolant temperature.
2. Constant flow of air-free coolant to the engine with minimum water pump suction.
3. Regulation of system pressure.

The above can be provided by using a continuous deaeration or rapid warm-up system. A typical cooling system with an integral top/surge tank radiator is shown on A-2 in the appendix. A system with a remote surge tank is shown on A-3 in the appendix.

NOTICE: There are no air vents or bleed holes in the thermostats or thermostat housings used on DDA automotive engines. A deaeration line is required to provide air venting during initial system fill and engine operation (see 4.1 top tank design).

Continuous deaeration systems function as follows:

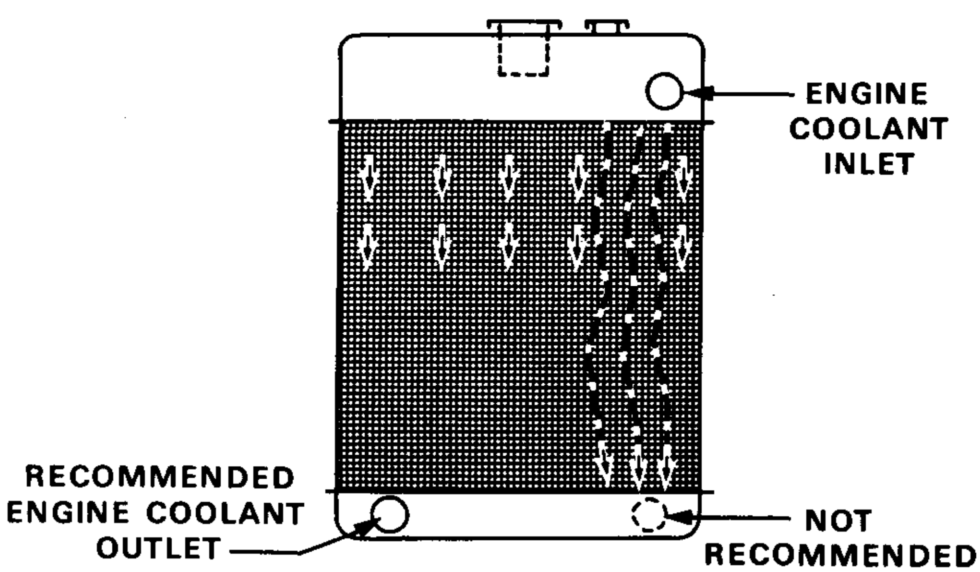
When the thermostats are closed, a sampling of coolant and any entrained air flows from the top of the thermostat housing through the deaeration line to the deaeration area in the top tank or surge tank. An equivalent amount of solid coolant is returned from the deaeration area through the fill line to the inlet side of the engine water pump.

When the thermostats are open, coolant and any entrained air flows through the engine outlets to the radiator top tank. Some coolant and entrained air flows through the standpipe or vent line into the deaeration area. An equivalent amount of solid coolant flows through the fill line to the inlet side of the engine water pump.

4. RADIATOR DESIGN

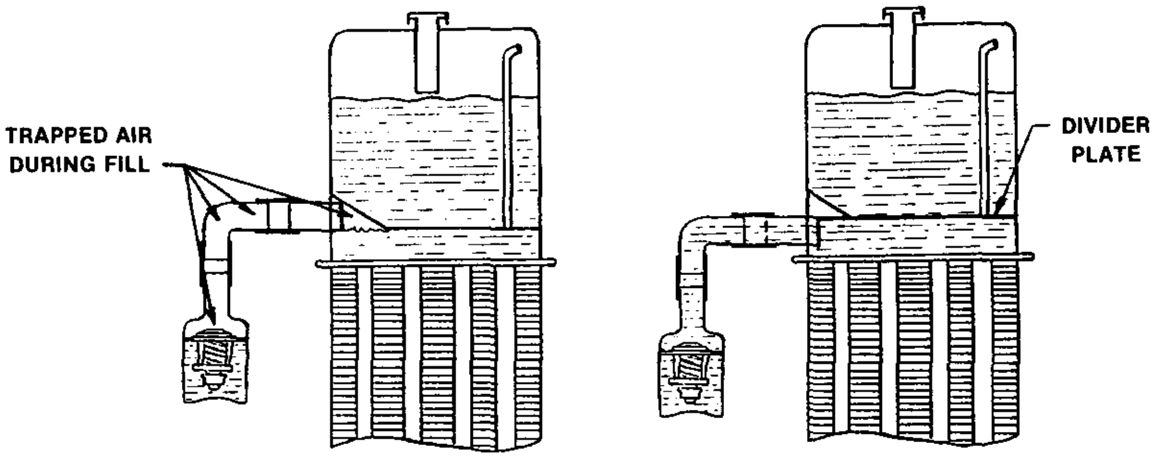
4.1 Top Tank Design

Regardless of the type of top tank, the top tank inlet diameter should be at least equal to the engine coolant outlet diameter. Dual inlets are preferred for 2 cycle vee engines. When only one top tank inlet is used, it should be located as far as possible from the radiator outlet to assure an even distribution of the coolant over the core. A baffle may be required to distribute the coolant.



There are no air vents or bleed holes in the DDA automotive engine thermostat housings and the radiator inlet hoses must be filled from the radiator at the time of intitial system fill. Therefore, the radiator inlet connections should be below the top tank divider to assure filling with coolant.

The pressure relief valve should be installed in the top of the deaeration area near its center to minimize the loss of coolant when the valve opens. A separate relief valve is preferred.



The integral top/surge tank or remote surge tank must be of sufficient capacity to provide for coolant expansion, deaeration, and drawdown. In general, a tank having a volume of 20%-25% of the total system capacity can be made to satisfy these requirements:

The expansion volume is the amount which the coolant expands during engine warm-up. Usually 6% of the total system capacity is considered adequate for expansion since a 50% water-ethylene glycol solution expands 6% over a temperature increase of 180°F (100°C). See A-4 in the appendix.

2% of the system capacity is required to provide space for deaeration.

The drawdown capacity is that amount of coolant which may be lost from the system before coolant aeration occurs. The volume required for drawdown is shown on A-4 in the appendix.

The above quantities are in additon to the quantity of coolant required to prevent aeration of coolant returning to the engine through the fill line. In some cases this amount may be decreased by adding a vortex baffle in the tank at the fill line connection.

The filler neck should be extended into the tank to provide a reference level for cold filling the coolant system. A 1/8″ diameter hole at the top of the filler neck is required to allow venting of air from the system without loss of coolant. It also provides for relief of system pressure, with a minimum loss of coolant, when the filler cap is removed from a hot system. See A-2 & 3 in the appendix

Typical deaeration line sizes are shown on A-1 in the appendix. On vee engines the deaeration line should be installed low in the deaeration area to prevent the possibility of air being taken from the tank to the engine when the thermostats are open; the deaeration line senses pump suction under this conditon. The fill line to the suction side of the pump should be 1" I. D. minimum and be connected to the lowest point of the deaeration area. To prevent its returning aerated coolant to the engine, it should not be near the deaeration line, standpipe or vent line.

4.2 Top Tank Design With Integral Top/Surge Tank

The top tank is divided into two chambers by a divider plate sealed along its periphery. The upper chamber is the deaeration area and the lower chamber is the core top tank. The distance between the divider plate and the radiator core, generally 1" or more, must be great enough to permit entrained gasses to separate and exit up the standpipe.

The standpipe is required to vent the core during system filling and for continuous open thermostat deaeration. Its discharged coolant should be directed away from the fill line and the filler neck. The top of the standpipe should be as high as practical and *must* be above the coolant level under all engine operating conditions. If it is not, warm coolant will flow to the radiator core when the thermostats are closed and the engine may not properly warm up, shown on A-2 in the appendix.

4.3 Remote Surge Tank

If conditions do not permit the use of an integral top/surge tank, as described in 4.2, a remote tank is required. The bottom of the surge tank should be located above all other components of the cooling system, ie., the engine, thermostat housing, and radiator.

The tank must be of sufficient capacity to provide for coolant expansion, deaeration, drawdown, etc., as described in 4.1. In addition to the deaeration and fill lines described in 4.1, a vent line from the radiator top tank to the surge tank is required to vent the cooling system during filling and for continuous open thermostat deaeration. Its discharged coolant should be directed away from the fill line and filler neck. The connection should be above the coolant level under all engine operating conditions. If it is not, warm coolant will flow to the radiator core when the thermostats are closed and the engine may not warm up, shown on A-3 in the appendix.

4.4 Top Tank Design For Use With Remote Surge Tank

A top tank for use with a remote surge tank should be large enough to permit entrained gasses to separate and exit up the vent hose connected to the top of the surge tank above the maximum coolant level, shown on A-3 in the appendix.

4.5 Deaeration and Fill Lines

Schematics of cooling systems with connections for a continuous deaeration system are shown on A-2 & 3 in the appendix. Hose sizes are as shown on A-1 in the appendix.

For correct location of deaeration and fill line connections see section 4.1. The deaeration and fill line hose sizes shown are typical and will not necessarily be adequate for all cooling systems. A cooling system must be checked by the method specified in section 9.2 to determine if the system is satisfactory. It may be necessary to make the lines larger or smaller depending upon the conditions encountered.

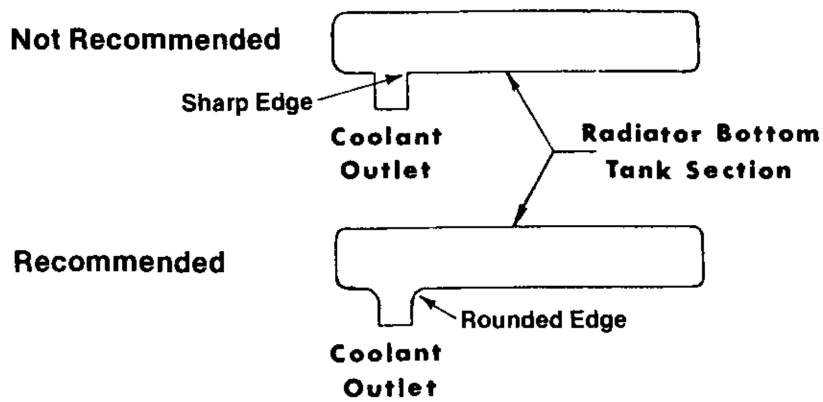
These hoses must rise continuously from the engine to the radiator so that air traps are not formed. Air traps can cause a false system fill condition and poor performance.

If the deaeration hose size used is less than the minimum specified, it will be necessary to conduct a deaeration test with the engine thermostats closed. See A-1 in the appendix.

If the fill line is smaller than that specified, the system fill rate may be too slow. It can also cause coolant to flow down the stand pipe or hose to the radiator resulting in poor engine warm-up. See A-1 in the appendix.

4.6 Bottom Tank Design

The bottom tank and coolant out connection should be designed to have minimum restriction to coolant flow. Sharp edged outlet connections should be avoided.



4.7 Core Design

Two heavy duty types of cores are in common use, the tube-and-fin type, and the tube-and-sepentine fin type. Cellular or honeycomb types are not suitable for heavy duty use and are not recommended. It is advisable to provide for as large a core frontal area as possible. This will usually result in the most efficient core and reduce the amount of fan horsepower required to cool the engine.

However, the heat dissipating capacity of a core cannot be determined by the frontal area alone. There are many varieties of fins, tubes, fin spacing, and tube spacing avaliable; these factors affect core cooling performance. Consult the radiator manufacturer for details.

4.8 Pressure System, Caps and Relief Valves

It is recommended that a minimum 9psi separate pressure relief valve be used for all DDA automotive cooling systems. The maximum allowable pressure is a function of radiator and/or heater core design.

The purpose of a pressurized system is to prevent coolant pump cavitation by maintaining pressure at the pump inlet during periods of operation at high coolant out temperature or at high altitude. It also prevents afterboil and coolant loss when a hot engine is idled or shutdown. See A-5 in the appendix.

4.9 Cross Flow Radiator

A crossflow radiator may be used if required but its use may involve problems of deaeration and filling. If a crossflow radiator must be used, *CONSULT THE FACTORY.*

4.10 Remote Radiator

Where remote mounting of a radiator is necessary, *CONSULT THE FACTORY* as there are many variables that affect the performance of such a cooling system.

5. FAN RECOMMENDATIONS

Proper fan selection is necessary to insure adequate cooling at the lowest possible parasitic horsepower and noise level.

5.1 Fan Selection

The selection of an engine cooling fan depends upon the following:

1. Cooling air flow required by the radiator core.
2. Cooling air system pressure drop.
3. Space available.
4. Noise level limit.
5. Horsepower capacity of the fan drive.
6. Fan speed limit.

5.2 Fan Performance

Three functions need to be considered when analyzing fan performance: speed, static pressure capability, and horsepower.

The following relationships are useful when interpreting basic fan curves.

$$CFM_2 = CFM_1 \times \frac{(RPM_2)}{(RPM_1)}$$

$$Ps_2 = Ps_1 \times \frac{(RPM_2)^2}{(RPM_1)^2}$$

$$HP_2 = HP_1 \times \frac{(RPM_2)^3}{(RPM_1)^3}$$

Some of the factors which affect the installed performance of a fan are:

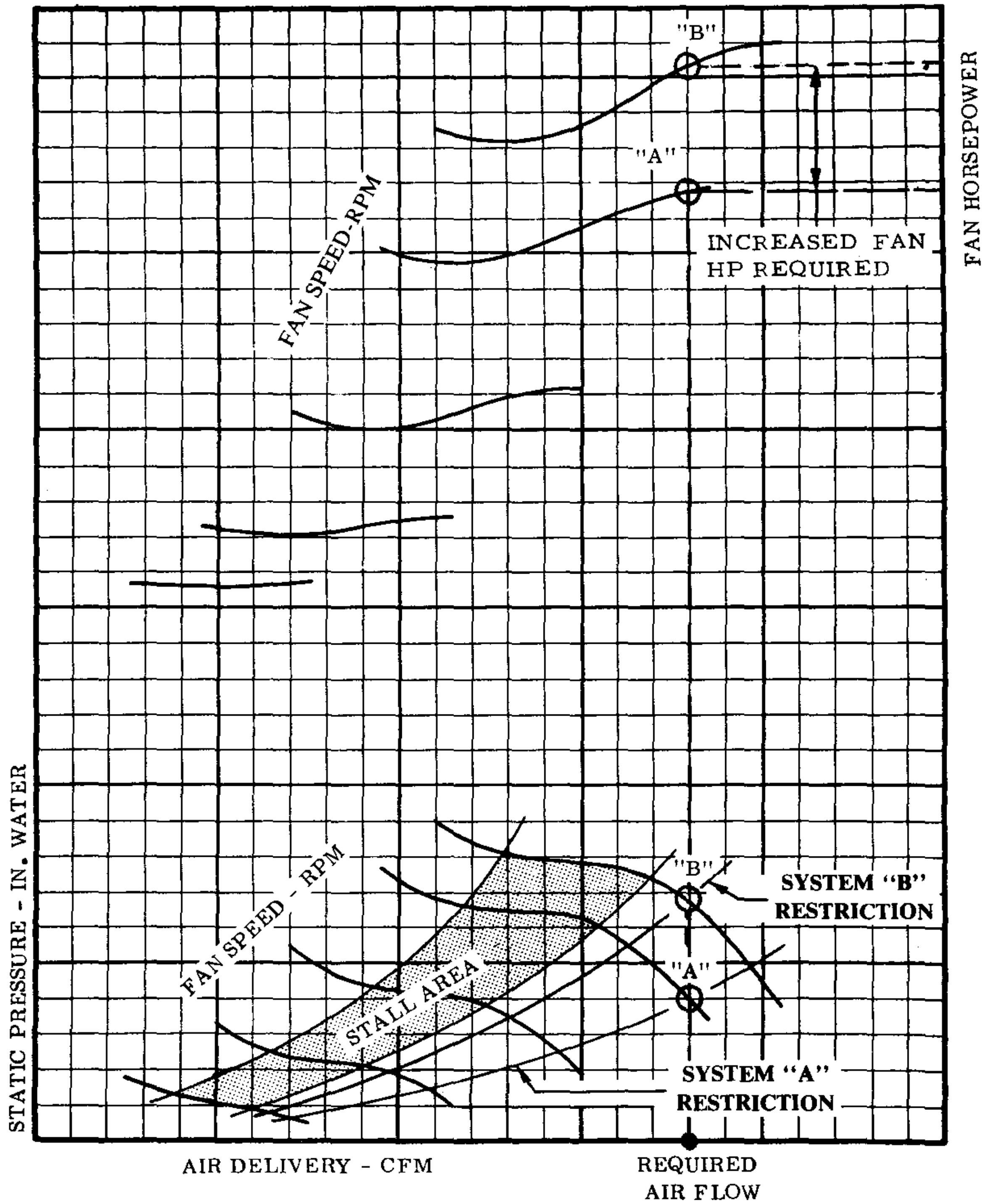
1. Fan position:
   - Fan to core distance
   - Fan to engine distance
2. Air flow restriction:
   - Radiator core
   - Engine compartment configuration
   - Engine
   - Grill
   - Bumper
   - Air conditioning condenser
   - Air-to-oil coolers
   - Shutters and bug screens
3. Shroud:
   - Shroud to fan tip clearance
   - Shroud to fan position
   - Shroud type, i.e. ring, box, venturi
   - Shroud to core seal

The most efficient fan installation will have the largest practical diameter fan turning at the slowest possible speed to produce the required radiator core air flow. In order to achieve this goal, the overall air system restriction must be kept to a minimum.

The parasitic horsepower penalty of high overall static pressure restriction is shown on the following typical fan performance curve.

Detroit Diesel Allison
Division of General Motors Corporation

RADIATOR - FAN MATCH



ENGINEERING-TECHNICAL DATA DEPT.

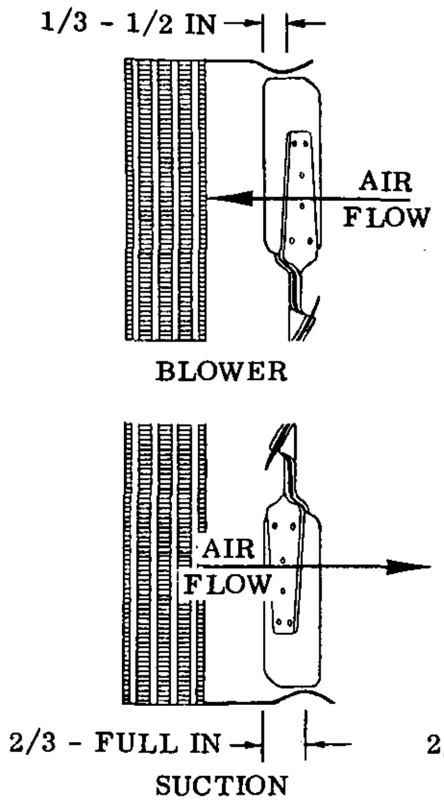
The system static pressure of installation "B" is higher than that of installation "A". Since the system air flow at pressure "B" must be the same as at "A" to achieve equal cooling, the fan RPM must be increased as shown on the curve. This RPM increase will result in an increased fan parasitic horsepower loss as indicated.

In no case should the fan operate in the stall area. In this area, small increases in restriction, such as foreign material in the fins of the core, result in a large loss of air flow.

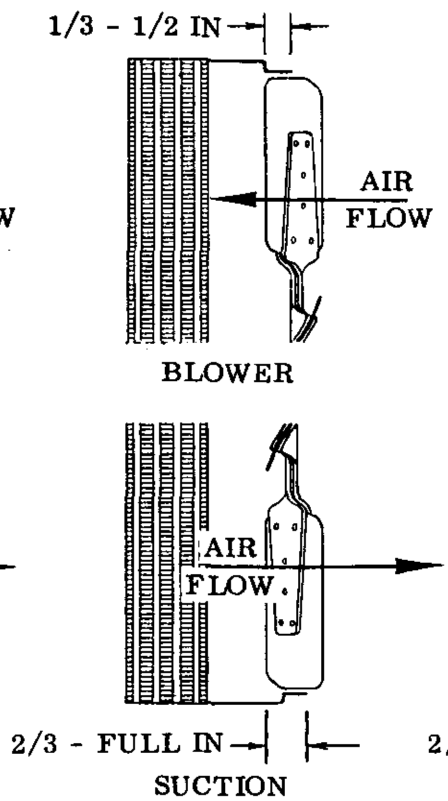
5.3 Fan Shrouds

The use of a shroud is required to achieve efficient fan performance. A good shroud will increase core air flow and reduce the recirculation of air. The shroud attachment to the core must be air tight to assure that all fan air passes through the radiator core.

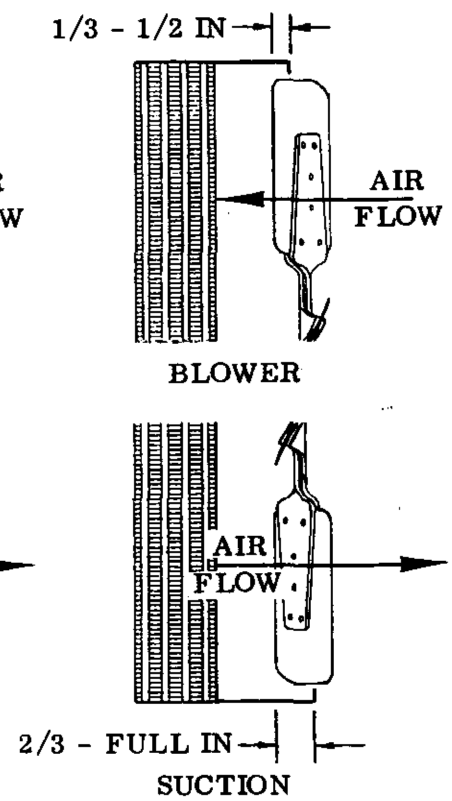
There are three basic types of shrouds, a well rounded entrance venturi type, a ring type, and a box type. Although ring or box type shrouds are more common, maximum air delivery is obtained with a venturi type.



Venturi Type



Ring Type



Box Type

Fan-to-shroud clearance (tip clearance) influences air flow and noise level significantly and should be kept to a minimum; this is best achieved by using a shroud with a round opening. If the fan pulley is adjustable for drive belt tightening, an adjustable fan shroud is recommended. The use of a fixed shroud with an oval opening will be less efficient and should be avoided.

The above figures also show the typical position of the fan blade with respect to the shroud. However, cooling system tests may reveal that a different relationship results in improved cooling system performance.

6. ACCESSORIES

6.1 Temperature Controlled Fan Drives

Temperature controlled on-off fan drives are recommended for use on DDA automotive truck engines and especially on constant horsepower models.

The radiator core air flow required to cool an engine is independent of the fan drive used. Fan drive clutches which are capable of locking-up can utilize the same fan as used with a solid drive. When drives having no lock-up capability are used, the fan size and/or drive ratio must be increased to compensate for the slippage in the drive and the resulting loss of air flow.

Some fan drives utilize coolant temperature sensors to control the on/off modes. Such sensors must be located so that they are fully submerged in flowing coolant at all times. It is recommended that the fan control be set to engage the fan when the engine coolant out temperature reaches 195 °F (90.6 °C)-200 °F (93.3 °C).

6.2 Shutters and Controls

Under some climatic conditions, the use of radiator shutters may be desirable.

If shutters are used, it is recommended that the control be calibrated to open the shutters before the engine thermostats open and be installed on the engine side of the engine thermostat. This calibration provides for the use of thermostatically controlled fans, which should engage at 195 °F (90.6 °C)-200 °F (93.3 °C) engine coolant out, without overlapping the temperature at which the various coolant controls operate.

For example:

Shutters open:
170 °F (76.7 °C)-175 °F (79.4 °C)

Thermostat starts to open:
180 °F (82.2 °C)

Fan engages:
195 °F (90.60 °C)-200 °F (93.3 °C)

If it is necessary to have the shutters open after the engine thermostat, which starts to open at 180 °F (82.2 °C), temperature controlled fan drives should not be used. The shutter control should be set to function at 10 °F (5.5 °C)-15 °F (8.3 °C) above the thermostat opening temperature when mounted to sense engine coolant out and at the same setting as the thermostat when mounted to sense coolant leaving the radiator.

The shutter control element must be fully submerged in flowing coolant at all times.

7. HEATER CONNECTIONS

The following recommended heater connection locations will provide the highest coolant pressure differential and maximum coolant flow for best heater core performance.

"71" Inline Engines

Heater Supply:
1. Cylinder block
2. Water manifold

Heater Return:
1. Water pump cover
2. Oil cooler housing

"V-71 and "V-92" Engines

Heater Supply:

1. Water pump to oil cooler elbow
2. Oil cooler to block connection
3. Cylinder block

Heater Return:

1. Water pump at bypass connection or main inlet connection
2. Thermostat housing

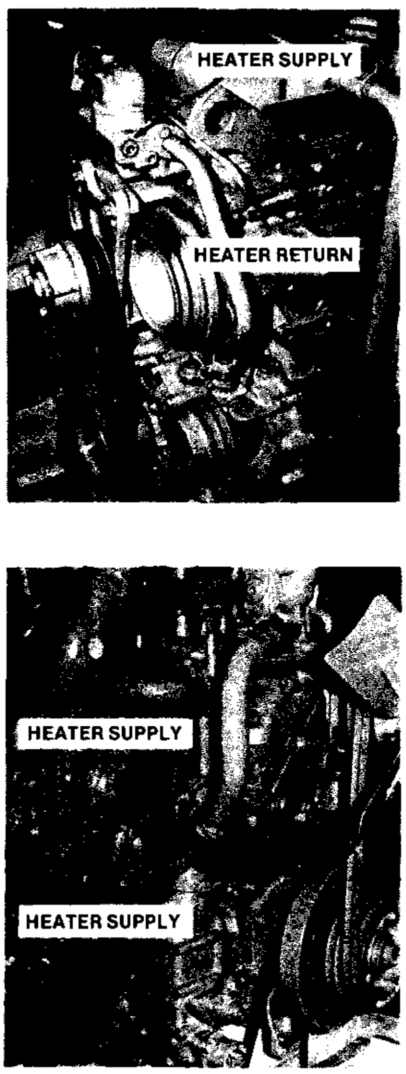
"53" Inline and Vee

Heater Supply:

1. Cylinder block
2. Cylinder head at rear
3. Thermostat housing
4. Cooler to block elbow

Heater Return:

1. Top of water pump



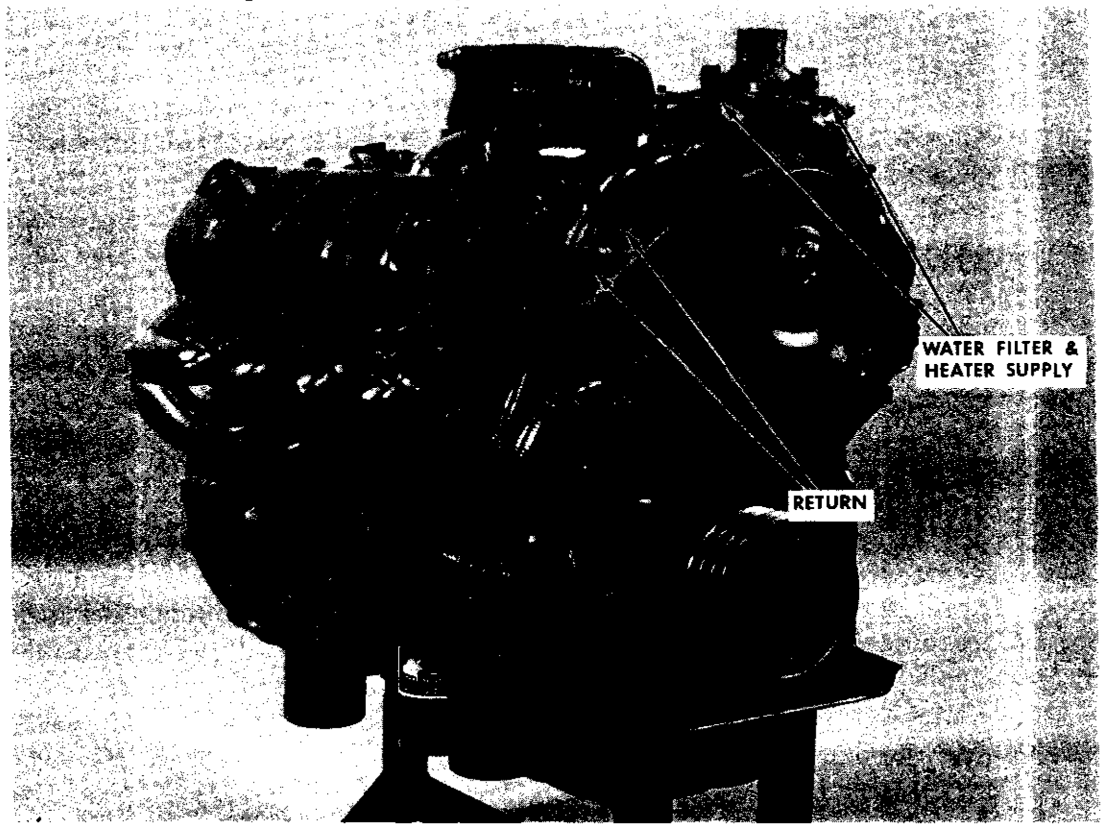
"71" In-Line Engine

"V-71, V-92" Engine

"8.2" Engines

Heater Supply: Thermostat housing
Heater Return: Top of Water Pump

See figure below.



"8.2" Engine

8. COOLANT AND COOLING SYSTEM INHIBITOR

Proper functioning of the coolant is basic to the successful operation of an engine and it must be carefully selected and properly maintained.

An adequate coolant inhibitor system is mandatory for all DDA automotive engines.

A suitable coolant solution must meet the following five basic requirements:

1. Provide adequate heat transfer.
2. Provide a corrosion resistant environment.
3. Prevent formation of scale and sludge deposits.
4. Be compatible with the cooling system hose and seal materials.
5. Provide adequate freeze protection.

Requirements one through four are satisfied by combining a suitable water with a reliable inhibitor system. When operating conditions dictate the need for freeze protection, a solution of suitable water and an ethylene glycol type antifreeze containing adequate inhibitors will provide a satisfactory coolant.

An adequate coolant inhibitor system is considered a combination of chemical compounds which provide corrosion protection, pH control, and water softening ability. A corrosion inhibitor protects the metallic surfaces of the cooling system against corrosive attack. pH control maintains an acid free solution. Water softening ability deters the formation of mineral deposits.

Inhibitor systems are available as liquid or dry bulk additives, coolant filter elements, and as an integral part of permanent antifreeze. Whichever system is used, additive must be periodically added to the coolant in order to maintain an adequate level of protection.

8.1 Bulk Inhibitor Additives:

An adequate coolant inhibitor system is *mandatory* for all DDA automotive engines. Bulk inhibitor additives are added directly to the cooling system. *Only a non-chromate inhibitor system is recommended* for use in Detroit Diesel engines because it is compatible with either water or ethylene glycol antifreeze solutions.

Some of the approved non-chromate inhibitor systems offer the additional advantage of a simple on site test to determine the protection level.

Chromate inhibitors are not recommended because they are not compatible with many permanent type antifreezes. Incompatibility results in the formation of chromium hydroxide, "green slime", which deposits in cooling system passages and reduces heat transfer which results in internal engine overheating.

Soluble oil is not recommended as a corrosion inhibitor because of its poor heat transfer characteristics. Water pump lubricants of any type are not needed or recommended.

8.2 Coolant Filters

Although an adequate coolant inhibitor system is mandatory, coolant filters are not required or recommended because of the added number of components required for mounting and plumbing and which carry with them the potential for mechanical failure and/or coolant leakage. Additionally, some filter elements contain magnesium plates which are *not recommended* for DDA automotive engines. The magnesium plate will be attacked by the coolant and dissolved magnesium will be deposited in the hottest zones of the engine where heat transfer is most critical.

Filter elements containing *chromate inhibitors are not recommended* because they are not compatible with many permanent type antifreezes. If coolant filters must be used in place of bulk inhibitors, they should be connected to the engine the same as a cab heater. See 7.0.

8.3 Antifreeze

Ethylene glycol base permanent antifreeze is recommended for freeze protection. An inhibitor system is included in this type of antifreeze and no additional inhibitors are required at initial fill if a minimum concentration of 30% by volume is used. Solutions of less than 30% do not provide adequate protection. Concentrations over 67% adversely affect freeze protection and heat transfer rates.

Antifreeze with a sealer additive can cause plugging problems throughout various areas of the cooling system and is *not recommended* for use in DDA automotive engines. (See A-9 in appendix).

Methyl Alcohol base antifreeze is *not recommended* because of its effect on the non-metallic components of the cooling system and because of its low boiling point.

Methoxy Propanol base antifreeze must not be used because it is not compatible with the fluoroelastomer (Viton®) seals which are used in DDA automotive engine cooling systems.

9. COOLING SYSTEM TESTS

Tests are to be run on all new installations and whenever a change that might affect the cooling system performance is made. When the tests show the cooling system performs satisfactorily, a Detroit Diesel End Product Questionnaire (E.P.Q.) should be completed for future reference. (See A-7 in appendix). Cooling test data may be recorded on the Cooling Test Data Sheet. (See A-6 in the appendix).

The cooling system tests include the following:

1. Continuous deaeration circuit, see paragraph 9.2.1.
2. Coolant flow and pump suction, see paragraph 9.3.
3. Drawdown, see paragraph 9.4.
4. Deaeration capacity, see paragraph 9.5.
5. Cooling Index, see paragraph 9.6.
6. System Fill Rate, see paragraph 9.0.

9.0 System Capacity and Fill Rate Test

Measure the quantity of water required to fill the cooling system to the bottom edge of the filler neck. Run the engine at various speeds to purge the system of any trapped air. If needed, add a measured amount of water to again fill the system to bottom of the filler neck and record the total system capacity.

The system must vent via the deaeration line so that it can be filled to at least 90% of the total system coolant capacity at a minimum rate of 3 gpm.

The test should be performed with cold water and the engine thermostats should be closed (not blocked open). After filling the system to the bottom of the filler neck at the 3 gpm rate, run the engine at varying speeds to purge any trapped air. Having purged the system, the quantity of water required to bring the coolant level back up to the bottom of the filler neck must not be greater than 10% of the total system capacity as determined above.

If the fill rate is too slow, the vent system should be modified and the test repeated.

9.1 Test Preparation and Instrumentation

The vehicle should be prepared for tests as follows:

9.1.1

Install an accurate tachometer calibrated to ±2%.

9.1.2

The maximum engine governor no load rpm setting should be adjusted to approximately 100 rpm above the rated no load speed to assure full fuel input at the specified test rpm.

9.1.3

Completely drain all of the coolant from the cooling system.

9.1.4

If shutters are used, replace the shutter control with a quick acting valve so that the shutter may be manually opened or closed.

9.1.5

Some tests require blocked open thermostats.

If closed thermostats are required, operating thermostats may be used provided there is no debris between the valve and its seat and the tests are completed before the thermostats start to open.

9.1.6

If a temperature controlled fan drive is used, its control must be modified so that the unit is actuated to its normal maximum drive capability. For example, an on-off clutch type drive would be in the full "on" mode. If the drive has no lockup capability, it should be in its minimum slip mode.

9.1.7

Install a pressure gauge 0-25 psi (0-344.76 kPa) or mercury manometer at the discharge side of the water pump to measure pump pressure.

9.1.8

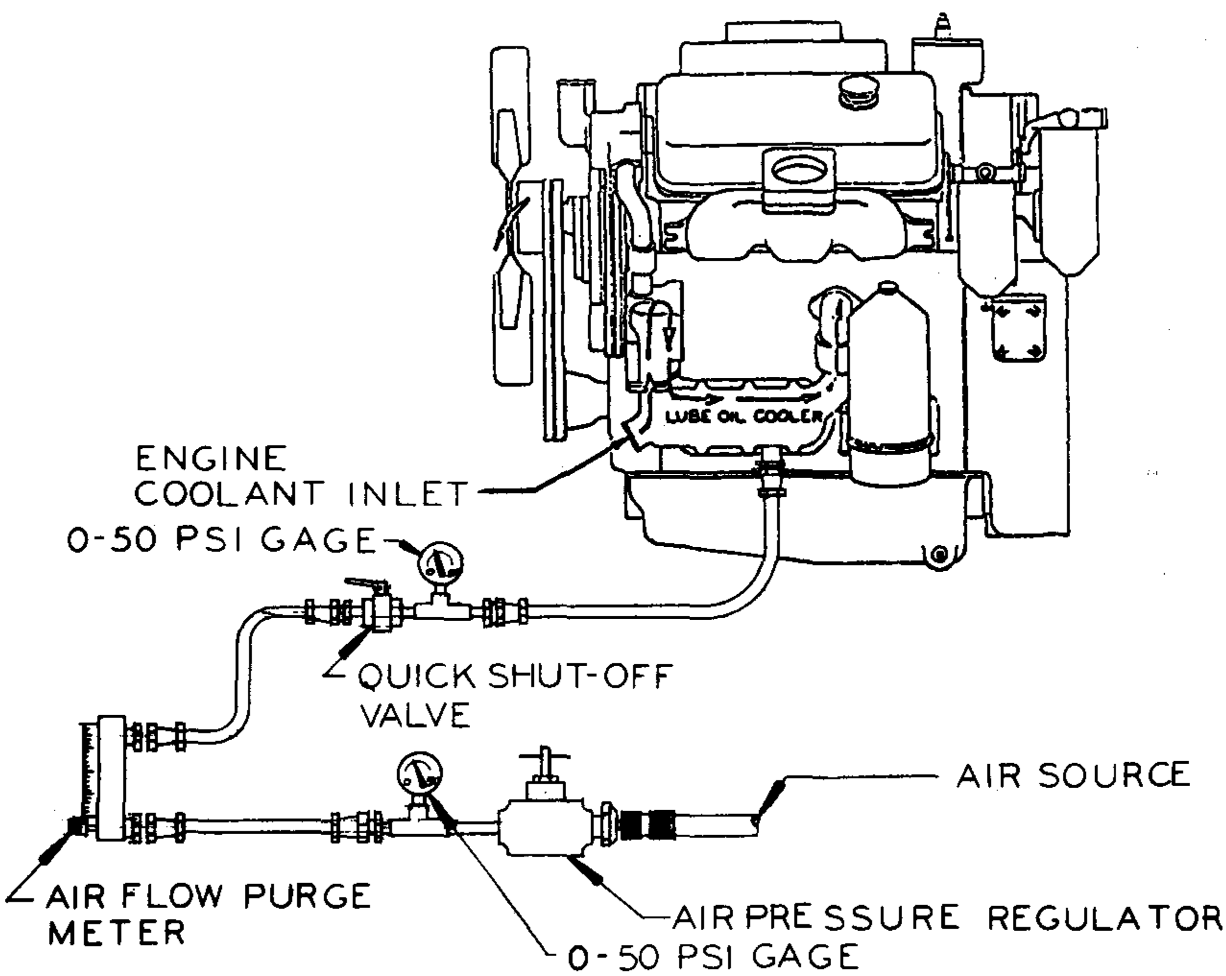
Install a mercury manometer at the inlet side of the water pump to measure the pump suction. It should be removed when running the deaeration test because the engine water pressure may exceed the manometer capacity.

9.1.9

Install a flow meter to measure radiator water flow. Venturi or turbine type meters work equally well on the suction or discharge side of the water pump. When using a venturi type meter, connect the legs of a "U" tube mercury manometer to the two taps on the flow meter. In order to obtain correct differential pressure readings, fill the manometer above the mercury zero line with water and bleed all air from the hoses.

9.1.10

Connect an air flow meter, gauges and regulator to the lowest point on the cylinder block or to the discharge side of the water pump, see figure.



9.1.11

Install a sight glass in the engine water outlet connections to the radiator.

9.1.12

Install thermocouples as follows:

(1) Engine water out: Install in water outlet elbows or thermostat housings. On vee engines install one thermocouple in each bank and average the readings.

(2) Engine water inlet: Install in the bottom of the radiator or in the radiator to engine connection.

(3) Radiator air: Install a 5 thermocouple harness in front of the radiator, one at each corner and one in the center, to determine if air recirculation occurs. If an oil-to-air cooler is mounted in front of the radiator, the harness should be installed between the two cores so that the engine radiator air inlet temperature is measured.

Install a thermocouple in the air stream on the engine side of the fan to measure radiator discharge air temperature. NOTE: If shutters are used, they should not be removed as they are a normal restriction to air flow.

(4) Ambient air: Locate two shaded thermometers or thermocouples to obtain an accurate temperature measurement of the ambient air approaching the radiator.

9.2 Deaeration Circuit and Deaeration Tests:

These tests are designed to determine that the cooling system functions to provide continuous deaeration and prevents coolant flow to the radiator when the thermostats are closed. Test results will be influenced by the cooling system full coolant level, standpipe height or remote surge tank radiator deaeration line connection location, deaeration line size (restriction), and fill line size (restriction). These items are inter-related and must function correctly to achieve the desired results.

9.2.1 Deaeration Circuit Test-With Integral Top/Surge Tank

This test is performed with *closed thermostats* and fill cap *off.*

1. Fill the system to the bottom of the filler neck (See A-2 & A-3 in appendix).
2. Run the engine at governed N.L. rpm. (Constant horsepower (TT) engines are to be run at full T engine N.L. rpm.)
3. With the engine running, pinch off the fill line. The water level should rise in the fill neck. The stabilized level will be at, and indicate, the top of the standpipe.
4. If the water level does not rise, the possible causes are:
   (1) The standpipe is too short.
   (2) The filler neck is too short.
   (3) The fill line circuit is too restrictive.
   (4) The deaeration line circuit does not have sufficient restriction.
5. With the engine stopped and the coolant at the normal full level (bottom of filler-neck), add an amount of water equal to 6% of the total system capacity, as determined in paragraph 9.0. The water level must not be above the top of the standpipe as previously determined. Run the engine at governed N.L. rpm. (Constant horsepower (TT) engines are to be run at full T engine N.L. rpm.) The water level must not rise above the top of the standpipe as previously determined.
6. If the level is too high, the possible causes are:
   (1) Standpipe is too short.
   (2) Filler neck is too short.

   Review design and make corrections as required. NOTE: Deaeration capacity and drawdown tests must be repeated if any changes are made as a result of the above tests.

9.2.2 Deaeration Circuit Test-With Remote Surge Tank

This test is performed with *closed thermostats* and fill cap *off.*

1. Replace the radiator deaeration line with a clear plastic hose of the same I.D.
2. Fill the system to the bottom of the filler neck.

3. Add an amount of water equal to 6% of the total system volume as determined in paragraph 9.0. The level of the coolant in the clear tube must be below its connection to the surge tank. If it is not, the possible causes are:

   (1) The filler neck is too short.

   (2) The tube connection is too low on the tank.

4. With the same quantity of water as in #3 above, run the engine at governed N.L. rpm. (Constant horsepower (TT) engines are to be run at full T engine N.L. rpm.)

   The water level in the clear tube will drop but no water should flow from the surge tank to the radiator. If it does, the possible causes are:

   (1) The filler neck is too short.

   (2) The tube connection is too low on the tank.

   (3) The fill line circuit is too restrictive.

   (4) The deaeration line circuit does not have sufficient restriction.

   Check design and make corrections as required. NOTE: Deaeration capacity and drawdown tests must be repeated if any changes are made as a result of the above changes.

9.3 Coolant Pump Pressure and Flow Tests

These tests are preformed with blocked *open thermostats*, fill cap *off*, engine coolant temperature above 150°F (65.6°C), and the engine running at *rated speed. (Constant horsepower (TT) engines are to be checked at the full T engine rated speed.)

Measure and record:

Water flow.
Water pump inlet pressure (suction).
Water pump pressure.

Water flows for DDA automotive engines are shown on A-11 & 12 in the appendix. The curves indicate the coolant flow with the various types of thermostats which are presently being used. Because of the effects of coolant temperature and system restriction, these flows are to be considered nominal.

Water pump suction should not exceed 3 in. Hg (10.31 kPa). If the suction is excessive, the system must be modified to meet the specification.

*Tests may be run at maximum governed speed instead of rated speed, for convenience. The higher speed, however, will make the test more difficult. Tests should be rerun at rated speed if unsatisfactory results are obtained with maximum governed speed.

9.4 Drawdown Test

The purpose of this test is to measure the amount of coolant that can be lost from a full system before air is drawn into the engine.

This test is performed at rated engine rpm* and no load with cooling system full, blocked *open thermostats*, fill cap *off*, and coolant temperature above 150°F. NOTE: Constant horsepower (TT) engines are to be checked at the full T engine rated speed.

Drain water from the engine in increments of approximately 2% of the total system capacity.

After each draining, check the engine water out sight glass for signs of air. (An alternate method is to replace the fill line with a clear plastic tube of the same ID and look for air entering the engine.) Continue to drain water in the same 2% increments until air *is* seen in the sight glass. Refer to A-4 to determine if the drawdown capacity is adequate.

If the system is not satisfactory, it must be modified to meet the specification.

*See note for paragraph 9.3.

9.5 Deaeration Test

This test is performed at rated engine rpm* and no load with cooling system full, blocked *open thermostats*, fill cap *on*, and coolant temperature above 150 °F (65.6 °C). NOTE: Constant horsepower (TT) engines are to be checked at the full T engine rated speed.

The air flow supply is installed at the lowest point on the cylinder block or on the discharge side of the water pump. See paragraph 9.1.10. Have suitable containers to catch and measure the water loss from the radiator overflow tube.

Adjust the air pressure regulator on the air supply side of the flow meter to approximtely 5 psi (34.5 kPa) higher pressure than the block pressure gauge reading. This will assure that air flows to the engine during the test.

Inject air *very slowly* up to 0.1 cfm (.003 M^3/min) and allow coolant flow to stabilize and coolant overflow to stop. *Slowly* increase the rate in 0.1 cfm increments until the desired rate as stated in paragraph 1.2 is being steadily injected, the system coolant flow has stabilized, and the overflow has stopped.

Air flow meters can be read directly only when the block pressure is the same as the meter calibration pressure. The observed air flow rates must be corrected if the block and calibration pressures are different. Corrected air flow values are provided for a Fischer-Porter model 10A3135N air meter on A-5 in the appendix. Air temperature correction is not required.

If the coolant loss is less than the drawdown capacity and the water pump flow does not drop below 50% of its original (no injected air) value, the system is satisfactory.

If the system is not satisfactory, it must be modified to meet the specification.

If the deaeration line size used is less than the minimum specified, a *closed thermostat deaeration test* must be conducted. This test is performed the same as the above except closed thermostats are to be installed in the engine and the test is begun with ambient temperature water.

The results of this test should be the same as for open thermostats except that the coolant flow should not stop (water pump should not become air bound).

*See note for Paragraph 9.3.

9.6 Cooling Index Test

All cooling index tests should be performed with the engine operating at maximum rated speed and wide open throttle (full fuel). EXCEPTION: Constant horsepower (TT), 71 and 92 series engines are to be tested at 1800 engine RPM and wide open throttle.

Tests are to be performed with a maximum of 15 MPH ram air applied to the front of the vehicle.

There are four methods of imposing rated load on a vehicle engine in order to determine the cooling index:

1. Driveline dynamometer (9.6.1).
2. Towing dynamometer (9.6.2).
3. Chassis dynamometer (9.6.3).
4. Steep hill-heavy load (9.6.4).

Test methods 1 and 2 establish the cooling index by applying load until a stabilized water temperature condition is reached.

Test methods 3 and 4 establish the cooling index based on water temperature change per unit of time. These tests consist of several short full load runs of equal duration, beginning each run at a different engine water out temperature.

The cooling index test is to be performed with the radiator fill cap on.

9.6.1 Driveline Dynamometer

1. Preliminary Preparations:

(1) Calculate and plot a shift pattern for the vehicle.

(2) Using a dynamometer absorption curve, a net engine horsepower curve, and the shift pattern, select the gear combination that will assure operation within the absorption range and speed capability of the dynamometer. Install blocked *open thermostats* and fill cap.

2. Test:

(1) The dynamometer is installed and warmed up.

(2) Verify test no load engine speed. See paragraph 9.1.2.

(3) Apply ram air to the front of the vehicle at a velocity no greater than 15 mph.

(4) Select the predetermined test gear combination, run the engine up to no load speed at full throttle. Increase the dynamometer load to bring the engine speed down to the specified test speed. See A-1 in the appendix.

(5) Record all temperatures every 60 seconds until they have stabilized for at least three minutes.

(6) Calculate the cooling index as described in paragraph 2.4.3, 2.4.4 and 2.4.5, using the stabilized water out temperature and the ambient air temperature.

(7) See paragraph 9.6.5 for evaluation of results.

9.6.2 Towing Dynamometer

1. Preliminary preparations:

(1) Calculate and plot a shift pattern for the vehicle.

(2) From the shift pattern, select the gear combination that will provide a vehicle speed no greater than 15 mph at the specified engine test speed. See paragraph 9.6.

(3) Install blocked *open thermostats* and fill cap.

2. Test:

NOTE: This test should not be conducted if ambient winds exceed 10 MPH (16 Km/h).

(1) Verify test no load engine speed. See paragraph 9.1.2.

(2) Select the predetermined test gear combination. Bring the engine up to slightly above rated speed and full throttle. Increase the dynamometer load to bring the engine down to the specified test speed.

(3) Record all temperatures periodically until they have stabilized for at least three minutes.

(4) Calculate the cooling index as described in paragraphs 2.4.3, 2.4.4, and 2.4.5, using the stabilized water out temperature and the ambient air temperature.

(5) See paragraph 9.6.5 for evaluation of results.

9.6.3 Chassis Dynamometer

1. Preliminary preparation:

(1) Same as for go power dynamometer except that a wheel horsepower curve is used in place of the net engine horsepower curve.

(2) Install blocked *open thermostats* and fill cap.

2. Test:

(1) Position the vehicle on the chasis dynamometer and warm up.

(2) Verify test no load engine speed. See paragraph 9.1.2.

(3) Apply ram air to the front of the vehicle at no greater than 15 MPH.

(4) Select the predetermined test gear combination. Bring the engine up to no load at full throttle. Increase the dynamometer load to bring the engine down to the specified test speed at full throttle.

(5) Record the ambient air and water out temperatures. Run one minute and again record the ambient air and water out temperatures. Continue recording temperatures at one minute intervals until no temperature change is observed for one minute.

NOTE: The vehicle drive tires will get very hot during a sustained run. It is recommended that the tires and rolls be allowed to cool, by running at low speed and no load for 10 minutes before proceeding to the next step.

(6) Cover the front of the radiator and then run at full load as above until the water out temperture reaches 210 °F (99 °C).

(7) Uncover the front of the radiator and record the ambient air and water out temperatures. Continue recording these temperatures at one minute intervals until no temperature change is observed for one minute.

(8) An example of these data, taken at one minute intervals, is shown in the following table.

COOLING TEST DATA-CHASSIS DYNAMOMETER

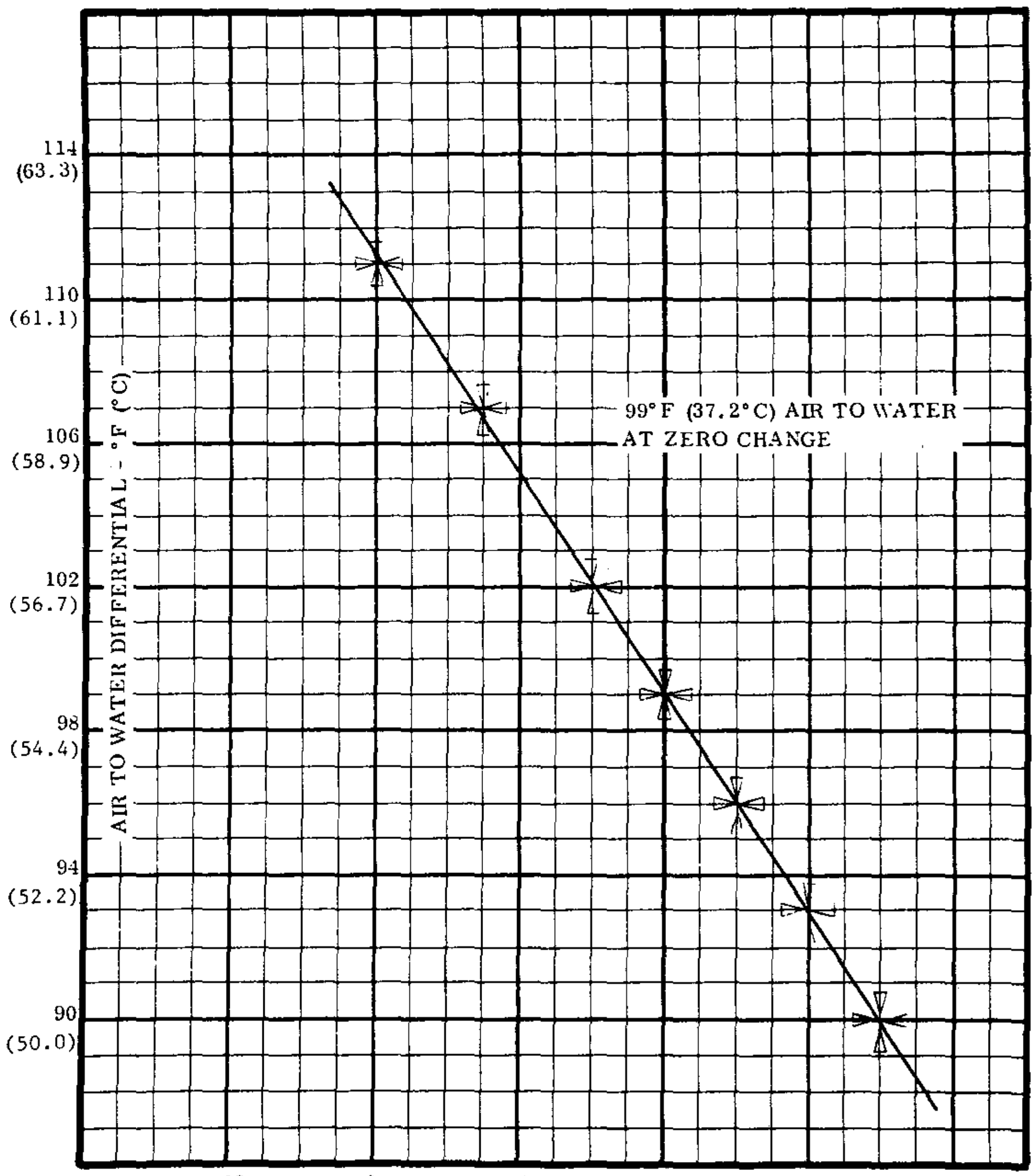
Time Min.	°F (°C) Ambient	°F (°C) Water Out	°F (°C) ATW	°F (°C) Change
01	Start of test-shutters open.			
06	85 (29.4)	169 (76.1)		
07	85 (29.4)	175 (79.4)	90 (50.0)	+6 (+3.3)
08	86 (30.0)	179 (81.7)	93 (51.7)	+4 (+2.2)
09	85 (29.4)	181 (82.8)	96 (53.3)	+2 (+1.1)
10	82 (27.8)	181 (82.8)	99 (55.0)	0 (0)
11	Removed power to let tires and rolls cool.			
24	Start of test shutters closed.			
29	Opened shutters.			
30	86 (30.0)	206 (96.7)		
31	87 (30.6)	198 (92.2)	111 (61.7)	−8 (−4.4)
32	86 (30.0)	193 (89.4)	107 (59.4)	−5 (−2.8)
33	89 (31.7)	191 (88.3)	102 (56.7)	−2 (−1.1)
34	91 (32.8)	191 (88.3)	100 (55.6)	0 (0)
	End of test. Shut down.			

(9) Plot the change of water out temperature as shown on the following graph. The point at which the line, established by the plot, crosses "zero change" is the air-to-water differential of the system. Correct the index for effect of altitude.

(10) See paragraph 9.6.5 for evaluation of results.

Detroit Diesel Allison
Division of General Motors Corporation

CHASSIS DYNAMOMETER COOLING TEST



ENGINEERING-TECHNICAL DATA DEPT.

DE 3769

9.6.4 Steep Hill Heavy Load

1. Preliminary preparations.

(1) Calculate and plot a shift pattern for the vehicle to be tested.

(2) From the shift pattern, select the gear combination that will provide a vehicle speed no greater than the allowable ram air velocity at the specified engine test speed. See paragraph 9.6.

(3) Using a wheel horsepower curve for the engine driveline combination to be tested, record the available wheel horsepower at the specified engine test speed.

(4) Using the test site % grade, calculate the approximate gross vehicle load requirement:

$$\frac{AWHP \times 3750}{\%Grade \times MPH} = \text{Gross Load in Pounds}$$

$$\frac{AWKW \times 27375}{\%Grade \times km/h} = \text{Gross Load in Kilo}$$

AWHP = Available wheel horsepower from Detroit Diesel curves.

% Grade = Percent grade in whole numbers.

Example: For 3.2% use 3.2

The final test load may vary from the above approximation. This can only be determined by operating the vehicle on the selected test hill.

(5) The test course is to be laid out using three flags or markers. One flag is placed on the grade near the bottom of the hill and another one is placed up the grade a distance equal to at least 45 seconds of vehicle running. (Approximately 1000 feet is required).

A third flag is used as a starting point. It is placed on the approach to the hill at a distance of about 5 seconds of vehicle running before the flag that is near the bottom of the hill. This five seconds of running is to allow any heat build-up to be expelled from the engine compartment.

(6) To verify the suitability of the test load, run the truck from the starting point through the test course. Between the two flags on the hill, the engine should be at full throttle and within 100 rpm, but not above, the specified test speed. For constant horsepower (TT) engines, the speed should be within ± 100 rpm of the specified test speed.

If the engine rpm is too high, add load; if it is too low, remove load.

(7) If the engine rpm varies while on the hill more than 100 rpm, a course having a more constant grade should be selected.

(8) Install blocked *open thermostats* and fill cap.

2. Test:

NOTE: This test should not be conducted if ambient winds exceed 10 MPH (16 Km/h).

(1) Verify test no load engine speed. See paragraph 9.1.2.

(2) Make a run from the vehicle starting point through the test course. Record water out temperature at the starting point. Record water out and ambient temperatures at each of the two flags on the test hill.

(3) Before starting the second run, cover the front of the radiator to obtain a starting point water out temperature 10°F (6°C) higher than at start of the first run. Uncover the radiator and begin the test run. Record the water out and ambient temperature at each of the two flags on the test hill.

(4) At the beginning of each sucessive run, the water out temperature is to be increased another 10 °F (6 °C), i.e., start 160 °F, 170 °F, etc. (73 °, 79 °C, etc.) to a maximum of 210 °F (99 °C) if this temperature is attainable.

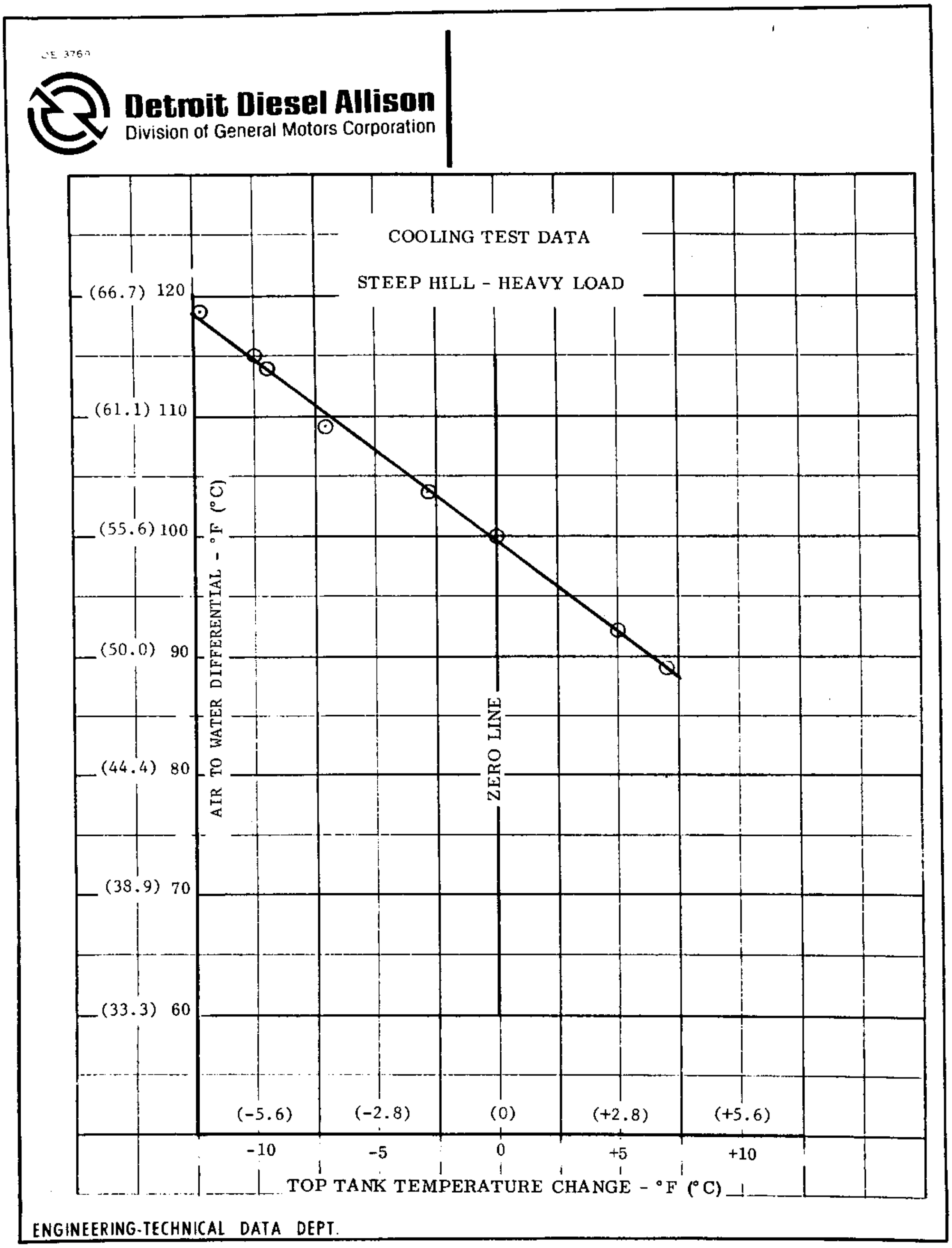
(5) An example of the data recorded and resultant calculations is shown below.

(6) See 9.6.5 for evaluation of results.

COOLING TEST DATA -- STEEP HILL - HEAVY LOAD

	- RUN NO. -								
	1	2	3	4	5	6	7	8	
Engine RPM	2120	2120	2110	2100	2090	2090	2075	2080	
Ambient Air	76 (24.4)	76 (24.4)	77 (25.0)	78 (25.6)	78 (25.6)	79 (26.1)	79 (26.1)	80 (26.7)	Ambient Air Temperature Start of Test
Coolant Out	158 (70.0)	165 (73.0)	178 (81.1)	184 (84.4)	194 (90.0)	201 (93.9)	205 (96.1)	209 (98.3)	Coolant out Temperature Start of run
Coolant Out	165 (73.9)	168 (75.6)	178 (81.1)	181 (82.8)	187 (86.1)	192 (88.9)	195 (90.6)	197 (91.7)	Coolant out Temperature End of run
Ambient Air	76 (24.4)	76 (24.4)	78 (25.6)	78 (25.6)	78 (25.6)	79 (26.1)	80 (26.7)	80 (26.7)	Ambient Air Temperature End of Test
Air to Water	89 (49.4)	92 (51.1)	100 (55.6)	103 (57.2)	109 (60.6)	113 (62.8)	115 (63.9)	117 (65.0)	Coolant Out (End of Run) minus Ambient (End of test)
Coolant Change	+7 (+3.9)	+5 (+2.8)	0 (0)	-3 (-1.7)	-7 (-3.9)	-9 (-5.0)	-10 (-5.6)	-12 (-6.7)	Coolant Out (Start) minus Coolant Out (End)

( ) Degree Celsius


DE 3769
Detroit Diesel Allison
Division of General Motors Corporation
COOLING TEST DATA
STEEP HILL - HEAVY LOAD
(66.7) 120
(61.1) 110
(55.6) 100
(50.0) 90
(44.4) 80
(38.9) 70
(33.3) 60
AIR TO WATER DIFFERENTIAL - °F (°C)
ZERO LINE
(-5.6)
(-2.8)
(0)
(+2.8)
(+5.6)
-10
-5
0
+5
+10
TOP TANK TEMPERATURE CHANGE - °F (°C)
ENGINEERING-TECHNICAL DATA DEPT.

9.6.5 Results of Cooling Index Test

The cooling index requirement for DDA automotive engine applications is shown in paragraph 2.4.2.

The results of tests conducted at altitude should be corrected as described in paragraph 2.4.5.

If the cooling index test reveals that the cooling system does not meet this requirement, the system must be modified and the test repeated to assure compliance.

If the cooling index is not adequate, an analysis of the temperatures recorded during the test will be useful in determining the cause of inadequate system performance.

The temperatures at each of the five thermocouples mounted on the front of the radiator should be within a few degrees of each other. When one or more of the thermocouples indicate excessively high temperatures, recirculation of radiator air is probable. Recirculation can be reduced by installation of suitable baffles.

The temperature rise of the air flowing through the core is the difference between the temperature of the air stream on the engine side of the fan and ambient air. A temperature rise that exceeds that recommended by the radiator manufacturer indicates insufficient air flow. This is the result of inadequate fan size or performance, and/or high air flow restriction.

If the above analysis of temperatures does not indicate a problem, the probable solution is to increase the cooling capacity of the radiator core.

If the cooling index is more than adequate, the cooling capacity of the system is excessive and the fan can usually be changed and/or its speed reduced. This will result in a favorable reduction in parasitic fan horsepower and noise. The test should then be repeated to assure that the system meets the requirement.

Appendix

SPECIFIED TEST SPEEDS		
Engine Model	Deaeration & Drawdown Tests	Cooling Index & Differential Tests
53	2600	2600
Inline & V71N	2100	2100
Inline & V71TT	2100	1800
V92	2100	2100
V92TT	2100	1800
8.2	3000	3000

Model	Engine Coolant Inlet Connection	Engine Coolant Outlet Connection	
		Single	Double
4 & 6V-53	1 7/8 Inches	1 5/8 Inches	1 1/8 Inches
Inline 71	2 Inches	1 7/8 Inches	-
V 71	2 3/4 Inches	2 1/2 Inches	1 7/8 Inches
V 92	3 Inches	-	1 7/8 Inches
8.2	60 MM		52 MM

AUTOMOTIVE ENGINE COOLANT CAPACITIES
U.S. QUARTS (LITERS)

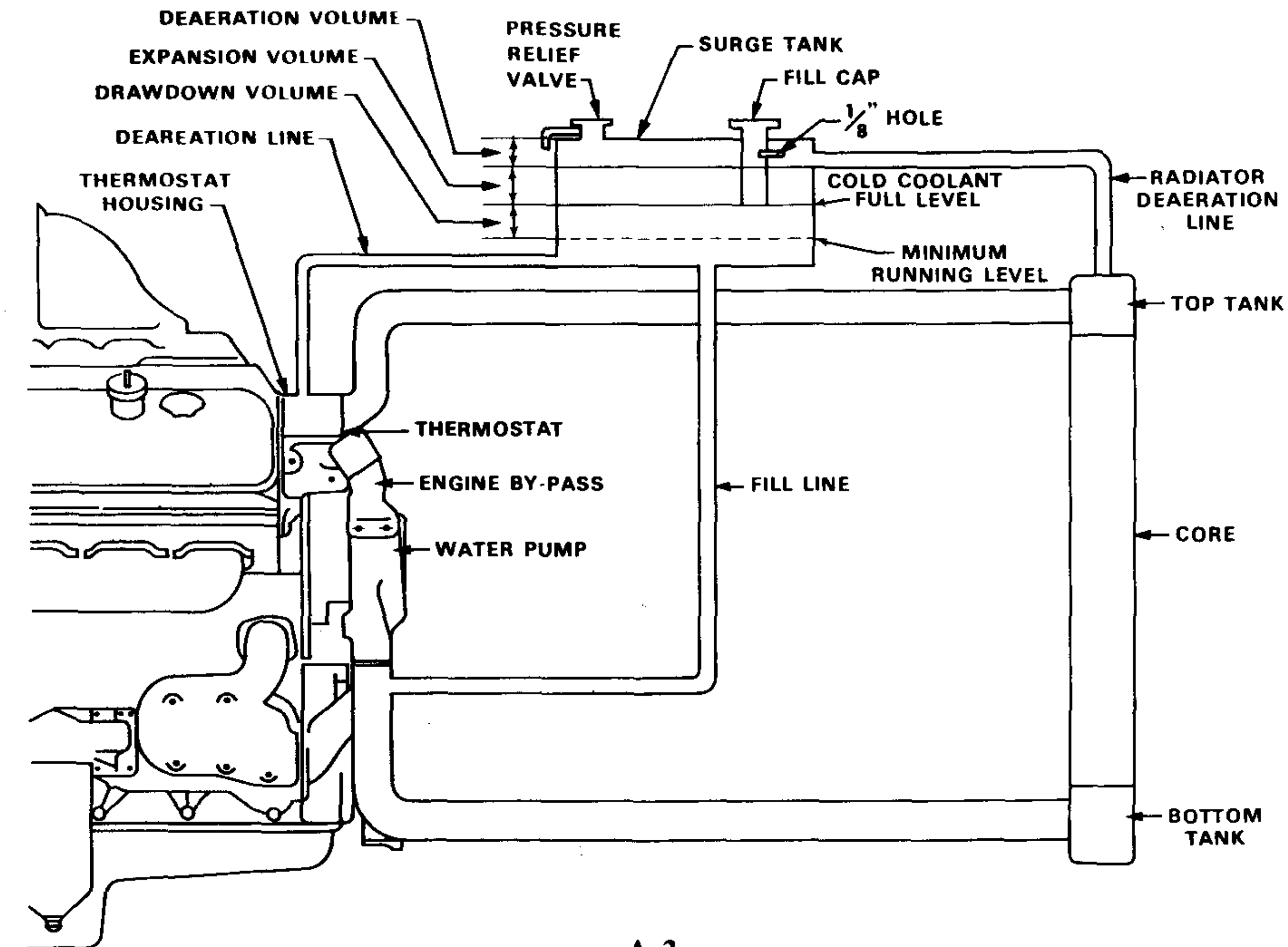
Engine	Basic Engine*	
4-53	9	(8.5)
6V-53	14	(13.3)
6-71N & T	22	(20.8)
6V-71N, T	28	(26.5)
8V-71N, T & TT	31	(29.3)
8V-71TA & TTA	32	(30.8)
6V-92N, T & TT	24	(22.7)
6V-92TA & TTA	25	(23.5)
8V-92N, T & TT	28	(26.5)
8V-92TA & TTA	29	(27.9)
8.2	12.5	(11.8)

*Engine block capacity and including after cooler capacity on TA & TTA engines.

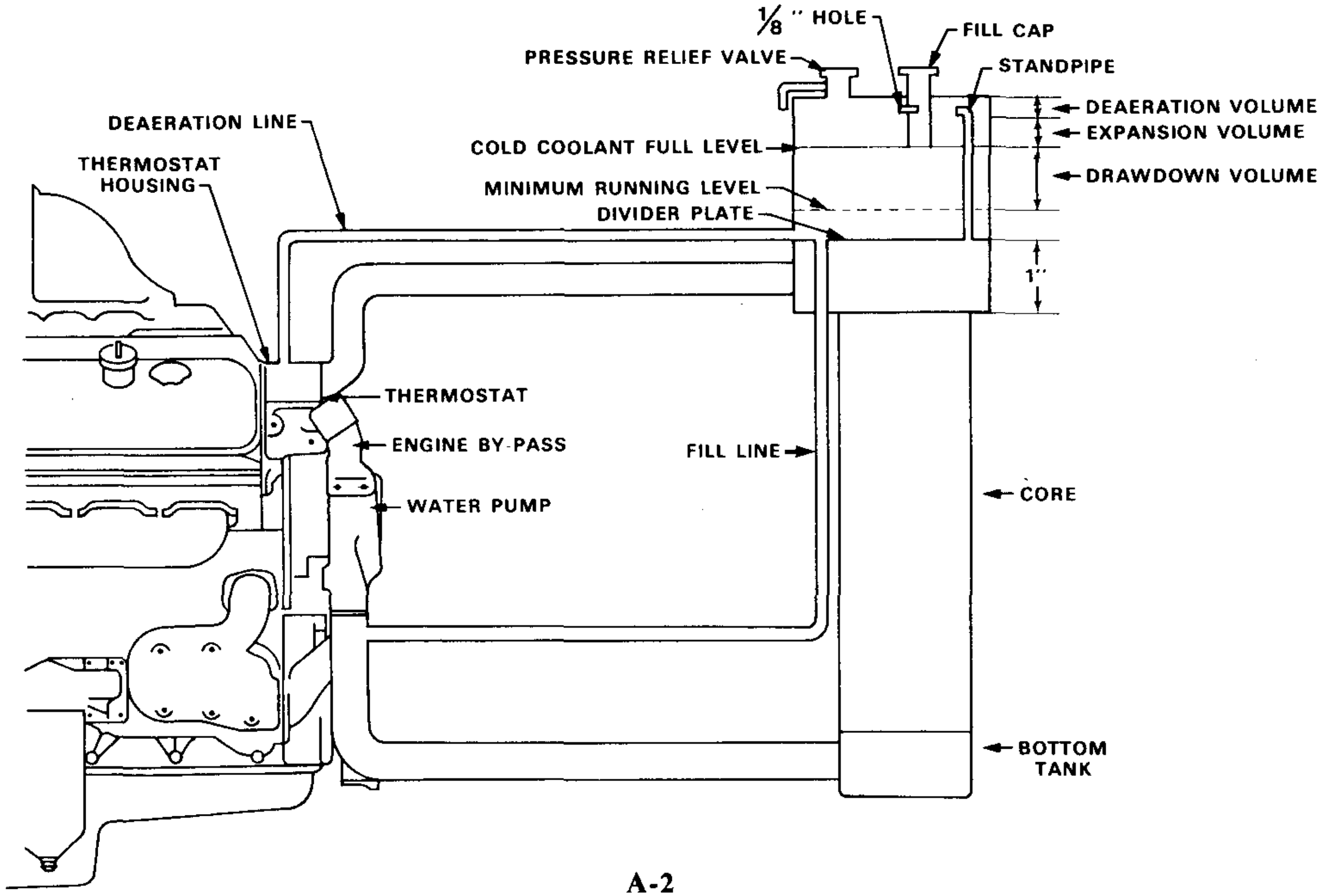
TYPICAL AUTOMOTIVE COOLING SYSTEM HOSE SIZES				
Engine	Deaeration Lines			Fill Line
	Engine to Tank	Tank to Radiator		
		Top Tank	Surge Tank	
Inline 71	1/4" I.D.	Intergal Standpipe 3/8" I.D.	3/8" I.D.	1" I.D.
V-71 & V-92	3/8" I.D.	"	3/8" I.D.	1" I.D.
53 & 8.2	1/4" I.D.	"	3/8" I.D.	1" I.D.

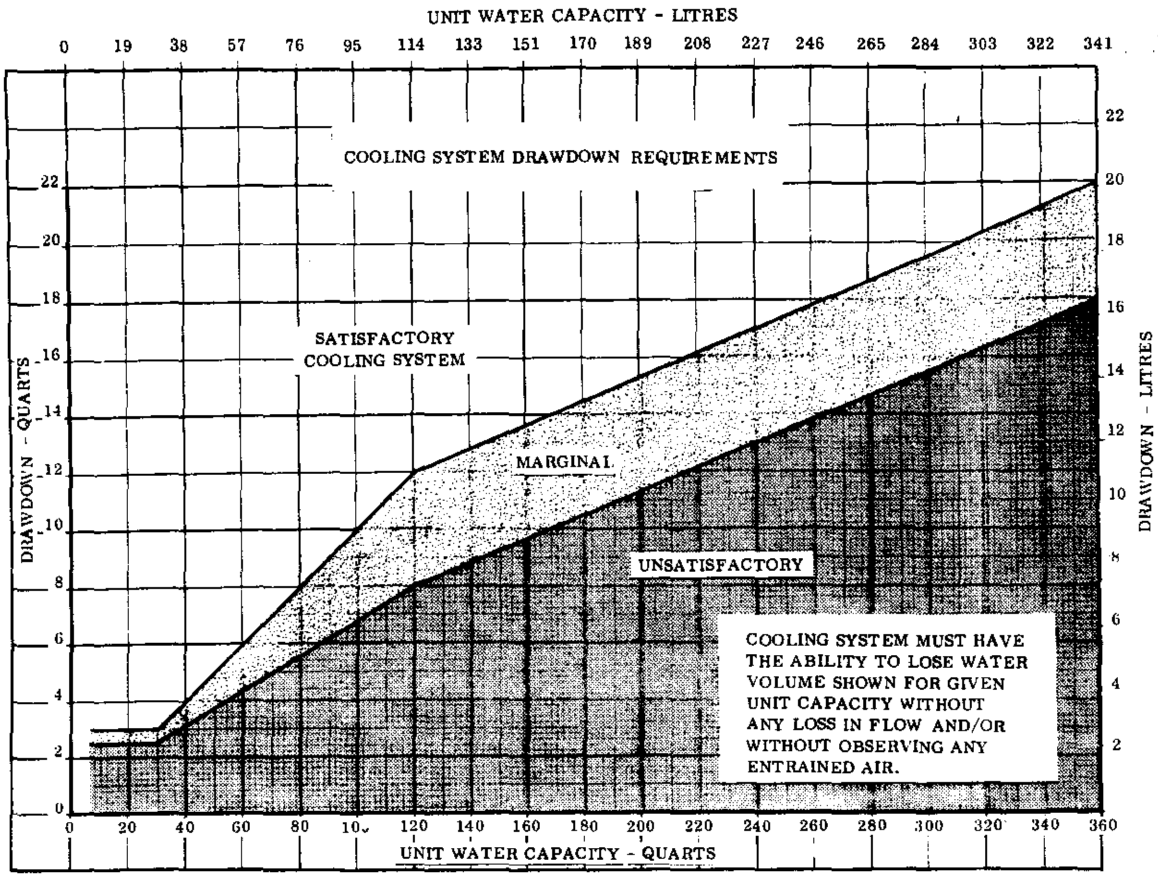
32

TYPICAL COOLING SYSTEM WITH A REMOTE SURGE TANK

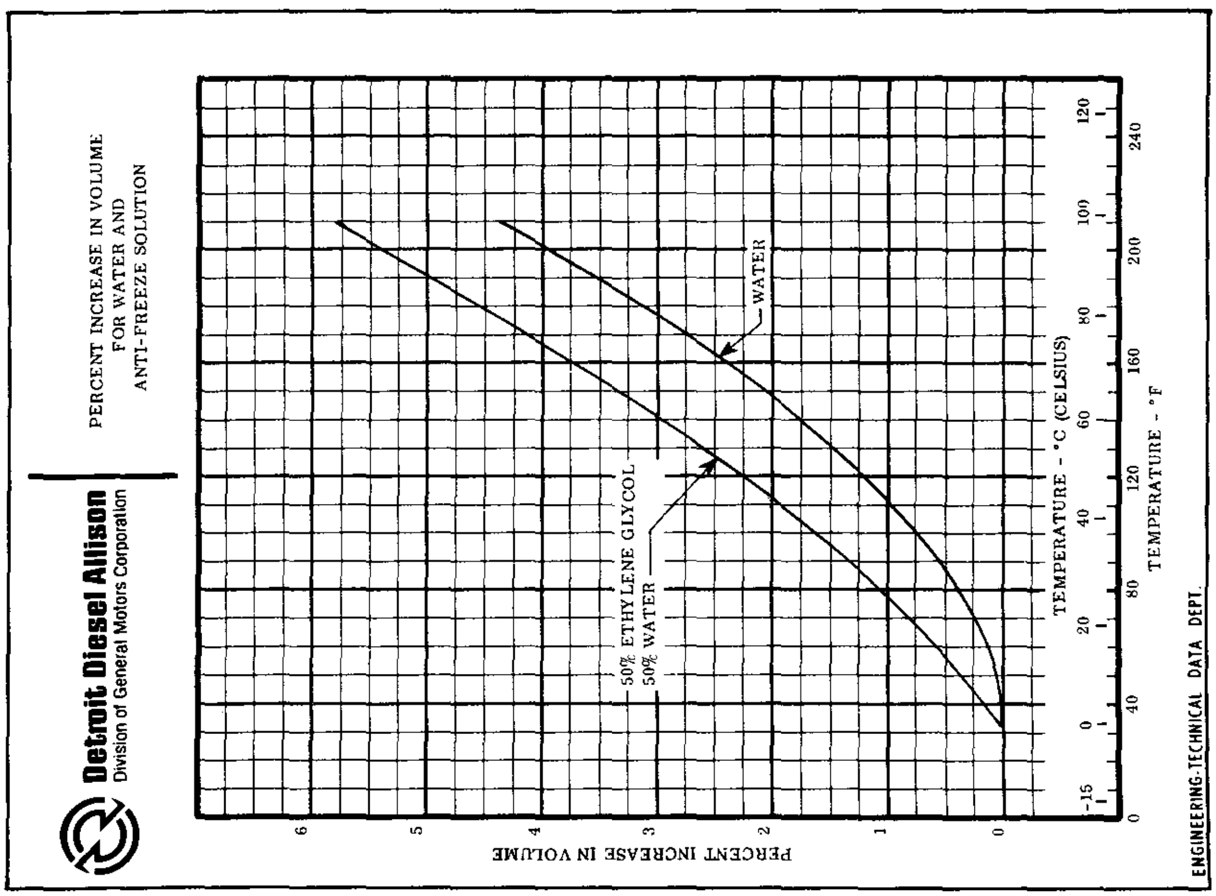


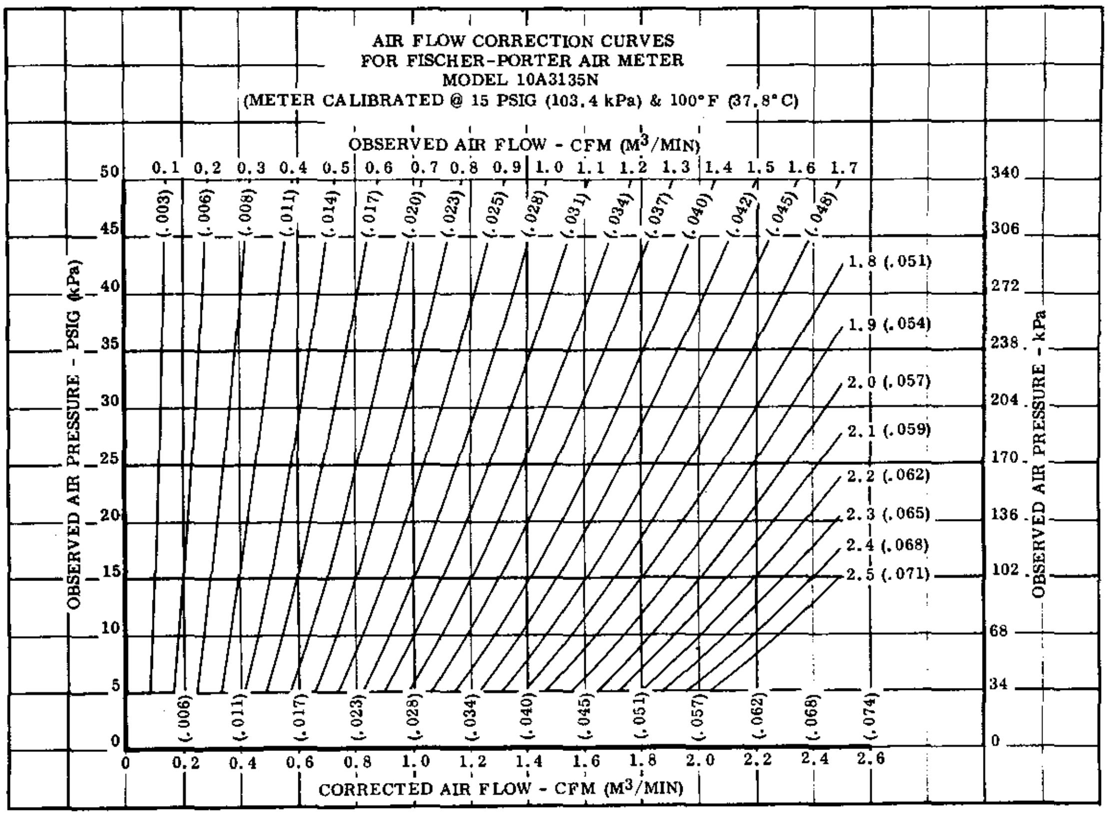
TYPICAL COOLING SYSTEM WITH AN INTEGRAL TOP/SURGE TANK RADIATOR



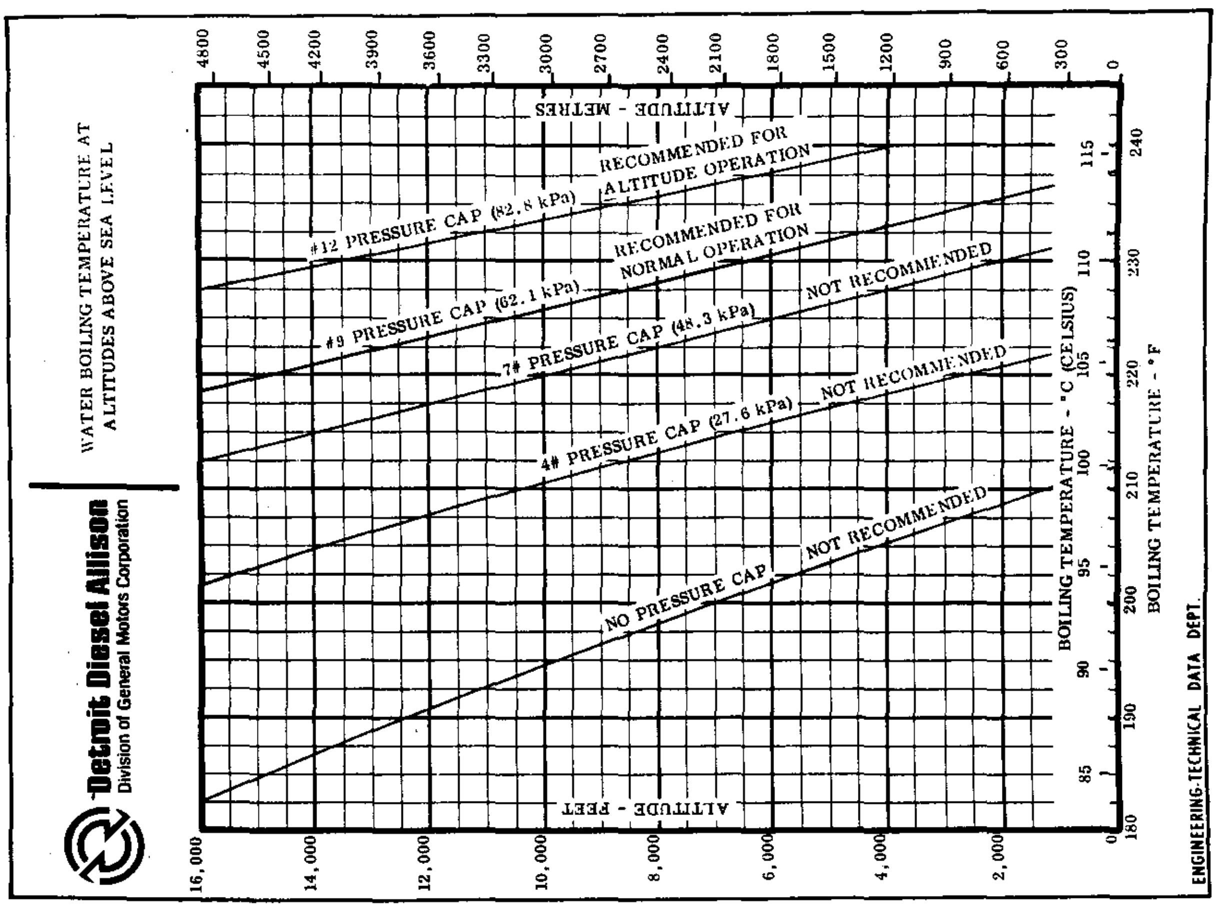

UNIT WATER CAPACITY - LITRES
COOLING SYSTEM DRAWDOWN REQUIREMENTS
SATISFACTORY COOLING SYSTEM
MARGINAL
UNSATISFACTORY
COOLING SYSTEM MUST HAVE THE ABILITY TO LOSE WATER VOLUME SHOWN FOR GIVEN UNIT CAPACITY WITHOUT ANY LOSS IN FLOW AND/OR WITHOUT OBSERVING ANY ENTRAINED AIR.
DRAWDOWN - QUARTS
DRAWDOWN - LITRES
UNIT WATER CAPACITY - QUARTS

A-4


Detroit Diesel Allison
Division of General Motors Corporation
PERCENT INCREASE IN VOLUME FOR WATER AND ANTI-FREEZE SOLUTION
WATER
50% ETHYLENE GLYCOL
50% WATER
TEMPERATURE - °C (CELSIUS)
TEMPERATURE - °F
PERCENT INCREASE IN VOLUME
ENGINEERING-TECHNICAL DATA DEPT.


AIR FLOW CORRECTION CURVES
FOR FISCHER-PORTER AIR METER
MODEL 10A3135N
(METER CALIBRATED @ 15 PSIG (103.4 kPa) & 100°F (37.8°C)
OBSERVED AIR FLOW - CFM (M3/MIN)
0.1 0.2 0.3 0.4 0.5 0.6 0.7 0.8 0.9 1.0 1.1 1.2 1.3 1.4 1.5 1.6 1.7
(.003) (.006) (.008) (.011) (.014) (.017) (.020) (.023) (.025) (.028) (.031) (.034) (.037) (.040) (.042) (.045) (.048)
1.8 (.051)
1.9 (.054)
2.0 (.057)
2.1 (.059)
2.2 (.062)
2.3 (.065)
2.4 (.068)
2.5 (.071)
OBSERVED AIR PRESSURE - PSIG (kPa)
50 45 40 35 30 25 20 15 10 5 0
OBSERVED AIR PRESSURE - kPa
340 306 272 238 204 170 136 102 68 34 0
(.006) (.011) (.017) (.023) (.028) (.034) (.040) (.045) (.051) (.057) (.062) (.068) (.074)
0 0.2 0.4 0.6 0.8 1.0 1.2 1.4 1.6 1.8 2.0 2.2 2.4 2.6
CORRECTED AIR FLOW - CFM (M3/MIN)

A-5


WATER BOILING TEMPERATURE AT
ALTITUDES ABOVE SEA LEVEL
Detroit Diesel Allison
Division of General Motors Corporation
ALTITUDE - METERS
4800 4500 4200 3900 3600 3300 3000 2700 2400 2100 1800 1500 1200 900 600 300 0
#12 PRESSURE CAP (82.8 kPa)
RECOMMENDED FOR
ALTITUDE OPERATION
#9 PRESSURE CAP (62.1 kPa)
RECOMMENDED FOR
NORMAL OPERATION
7# PRESSURE CAP (48.3 kPa)
NOT RECOMMENDED
4# PRESSURE CAP (27.6 kPa)
NOT RECOMMENDED
NO PRESSURE CAP
NOT RECOMMENDED
ALTITUDE - FEET
16,000 14,000 12,000 10,000 8,000 6,000 4,000 2,000 0
BOILING TEMPERATURE - °C (CELSIUS)
85 90 95 100 105 110 115
BOILING TEMPERATURE - °F
180 190 200 210 220 230 240
ENGINEERING-TECHNICAL DATA DEPT.
DE 3769

COOLING TEST DATA SHEET

Detroit Diesel Allison

TEST CONDUCTED BY ______________

DATE ______________

APPLICATION ______________________________

UNIT NO. ______________ INJ ______________ RPM ______________

ENGINE LOADING METHOD ______________ TEST LOCATION ______________

FAN

SUCTION/BLOWER ________ ________ DIA.

________ BL ________ PROJ. WIDTH

DRIVE RATIO ______________

SHROUD (SKETCH PREFERRED) ______________

FAN TO CORE DISTANCE ______________

RADIATOR

MANUFACTURER ______________

PART NO. ______________

RADIATOR CORE W ____ x H ____ x T ____

FINS/INCH ________ ROWS OF TUBES ________

NO. OF TUBES ________

TOTAL SYSTEM CAPACITY ____ QTS. PRESS. CAP ____ LB

THERMOCOUPLE		TIME							STABILIZED TEMPERATURES
Radiator Air	1								
	2								
	3								
	4								
	5								
	Ave.								
L.B. Coolant out									
R.B. Coolant out									
Engine Coolant in									
Oil sump									
T_1 Ambient thermometer									
T_2 Ambient thermometer									

DEAERATION

COOLANT PUMP FLOW ________ GPM @ ZERO CFM

________ CFM HANDLED WITH ________ QTS. LOSS

COOLANT PUMP FLOW ________ GPM WITH ________ CFM

DRAWDOWN

________ GPM WITH ________ QTS. LOW

REFERENCE FIGURE 38, PAGE 59

COOLING INDEX

ATW ________ @ ________ RPM

ATW ________ @ ________ RPM

ATB ________ @ ________ RPM

ATB ________ @ ________ RPM

COOLANT PUMP INLET SUCTION AT

STABILIZED TEMPERATURE ________

REMARKS: ______________________________

____

____

REF. E.P.Q. DATED ________

BY ________

A-6

Prepare Original and Three Copies:
Original and One Copy to Sales Engineering D. D. A. D.
One Copy to Regional Office
One Copy Retained in Customer File

Date ________ ________

Assigned
EPQ Number ________

ON-HIGHWAY VEHICLE
END PRODUCT QUESTIONNAIRE
OR
PILOT MODEL INSTALLATION REPORT

Completed by (Name) ________ Employed by ________
Address ________
Vehicle Mfg. by: ________ Veh. Serial # ________ Veh. Model # ________
Eight digit
D. D. A. D. Engine Model ________ Injector Size ________ Unit No. ________
Application Code Number ________

APPLICATION DATA

GVW Rating ________ GCW Rating ________ Type Cab: ________ B. B. C. ________ in.
Max. Engine Speed: FL ________ rpm. NL ________ rpm. Idle ________ rpm.
Transmission Make ________ Model ________ Aux. Trans. Make ________
Aux. Model ________ Rear Axle Make ________ Model ________
Ratio ________ Tire Size ________ Engine: Installation Angle ________
Clearance with Frame ________ Drive Shaft Angle ________
Engine Mounts: Type-Front ________ Rear ________
Transmission Support: (if used, describe) ________
Describe in detail all accessories driven from engine:

1. Access. ________ Location ________ Type Drive ________ HP Required ________
2. Access. ________ Location ________ Type Drive ________ HP Required ________
3. Access. ________ Location ________ Type Drive ________ HP Required ________
4. Access. ________ Location ________ Type Drive ________ HP Required ________

Additional Comments: ________

____

ACCESSIBILITY FOR SERVICING

(Rating : Good, Fair, Poor - Use G, F, P)
Filters: Fuel ________ Lube Oil ________ Coolant ________ Air Cleaners ________
Rocker Covers: LB ________ RB ________ Cyl. Heads ________ Inspection Covers ________ Oil Pan Removal ________
Oil Filler ________ Dipstick ________ Filling Coolant ________ Fan Belts ________
Governor ________ Blower ________ Access. ________ Engine Removal ________
Additional Comments: ________

____

LUBE SYSTEM

Location of: Dipstick ________, Oil Filler ________, Oil Filters ________
Lube oil filters other than supplied by DDAD ________

FUEL SYSTEM

Fuel Tank: Material ________ No. Tanks ________ Capacity (each) ________ Gal.
Fuel Lines Inlet: Size ________ In. Length ________ In. Material ________
Describe Bends & Elbows ________
Fuel Lines Return: Size ________ In. Length ________ In. Material ________
Describe Bends & Elbows ________
Location of Lines in Tank ________

Distance Between Inlet & Return Lines in Tank ________ In.

Baffles in Tank ________ Do Baffles Separate Inlet & Return Lines ________

Height of Fuel Tank to Fuel Pump ________ In.

Inlet Restriction at Pump ________ In. Hg. at ________ rpm.

Governor Type ________ Starting Aid (if used) ________

Additional Comments: ________

COOLING SYSTEM

Type System (Conventional or Rapid Warm-Up) ________ System Capacity ________ qts.

Radiator: Make/Model ________ Size (Height, Width, Thickness) ________ in. X ________ in. x ________ in.

No. Rows of Tubes ________ Fins./in. ________ Heat Capacity (if known) ________ BTU/Min.

Expansion Tank (if used) Capacity ________ qts. Location ________

Cooling Line Sizes: Inlet ________ in. Outlet ________ in. Location ________

Fan: Suction/Blower Diam. ________ in. No. of Blades ________ Proj. Width ________ Drive Ratio ________

Fan type: Solid ________ Viscous ________ Clutch ________

Distance from Core ________ in. Fan Location in Shroud ________

Shroud: Box/Venturi, etc. ________ Dia. ________ in. Fan Location ________ Fan Tip Clearance ________ in.

Heater Connections: Size and Location ________

Stabilized Coolant Temperature at Max. HP ________ °F.

Cooling Test ________ ? (Attach Data Sheet) Form DE 2804.

Describe Deaeration System (Line Sizes and Location) ________

Describe any Obstructions to Air or Water Flow (Coolers, Shutters, etc.) ________

AIR SYSTEM

Air Cleaner: Type ________ Make ________ Model ________

Cleaner Location ________ (Describe Ducting, Size) ________

________ (Length) ________ (Material) ________

Restriction Gate Used (if applicable) ________ (Location, if used) ________

Air Inlet Restriction ________ in. H_2O at FL/NL ________ rpm.

Additional Comments: ________

EXHAUST SYSTEM

Muffler: Single/Dual, Make ________ Model ________ Size ________

Location of Mufflers ________ Location Relative to Air Intake ________

Manifold to Muffler Connection (Solid/Flex) ________

Rain Cap or Other Protection ________ Exhaust Pipe Size ________

(Length) ________ (Describe Bends & Elbows) ________

Exhaust Back Pressure: in. Hg ________ Left Bank : FL/NL at ________ rpm. in Hg ________ Right Bank: FL/NL at ________ rpm.

Additional Comments: ________

ELECTRICAL SYSTEM

Battery: System Voltage ________ Battery Voltage ________ No. Used ________

Type of Starter ________ SAE Cold Cranking Amp @ 0°F ________

Cable: Size ________ Length ________ Generator Capacity ________ Amp.

Generator Drive Location ________ Type Drive ________ Starter Switch Type ________

Shutdown or Alarm System ________

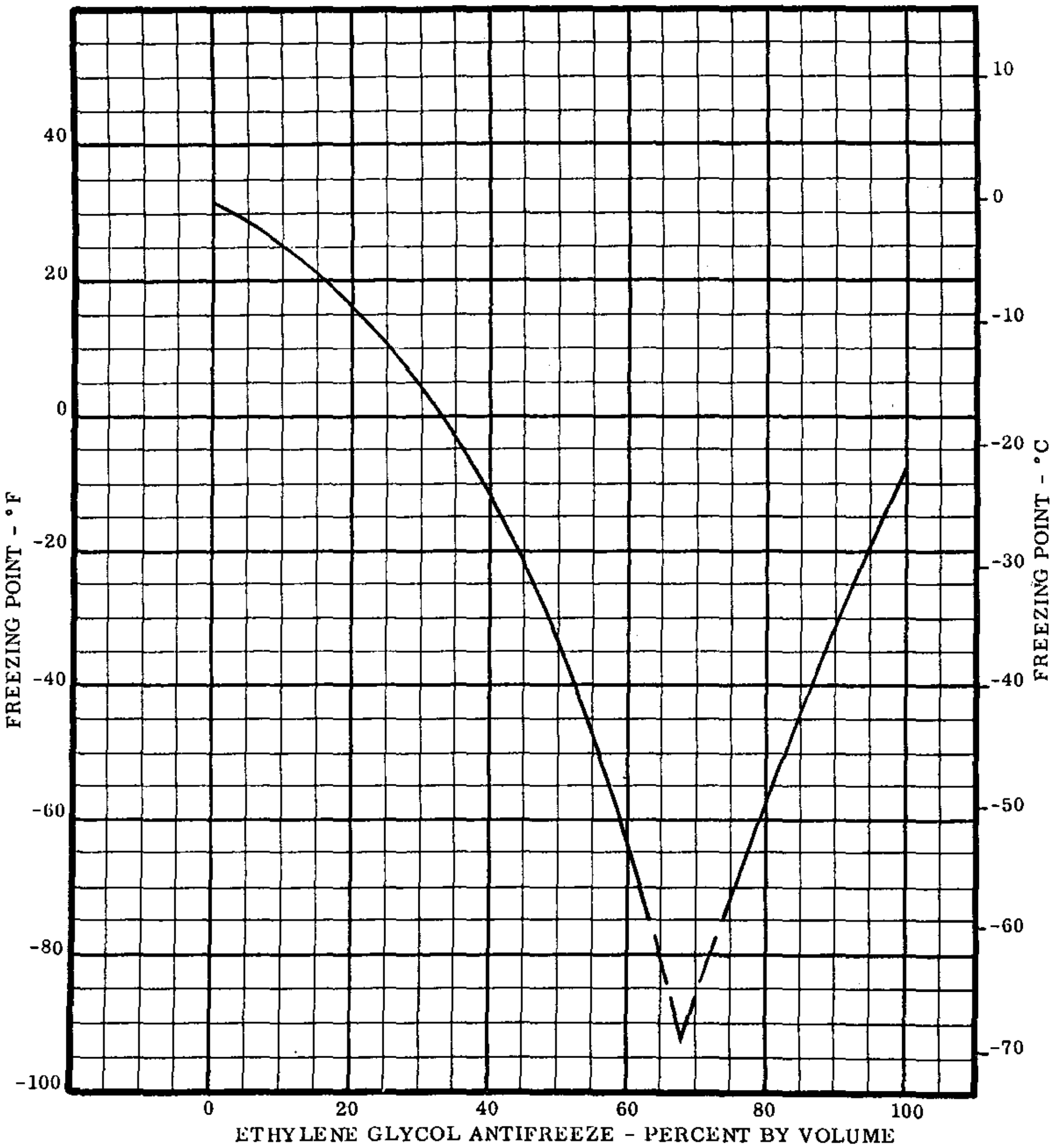
Additional Electrical Requirement and Comments: ________

SPECIAL TESTS

Cooling, Emissions, Sound Level, Vibration, Torsional, etc. (Attach Data).

Detroit Diesel Allison
Division of General Motors Corporation

FREEZING POINTS OF AQUEOUS ETHYLENE GLYCOL ANTIFREEZE SOLUTIONS



C4M-0000-00-1
7-12-79

ENGINEERING-TECHNICAL DATA DEPT.

DE 3769

A-9

1979 AUTOMOTIVE ENGINE HEAT REJECTED TO COOLANT AT 195°F (90.6°C)

Engine	Injector	Rated Power BHP @ RPM	Rated Power kW @ RPM	Emission Level NO_x + HC	Heat Rejected to Coolant BTU/Min.	Heat Rejected to Coolant W
4-53T	5A60	170 @ 2500	127 @ 2500	10	4600	80888
	5A55	155 @ 2500	116 @ 2500	10	4000	70337
4-53TC	5A60	170 @ 2500	127 @ 2500	7.5 + 1 Calif.	4950	87042
	5A55	155 @ 2500	116 @ 2500	7.5 + 1 Calif.	4200	73854
6V-53T	5A55	235 @ 2600	175 @ 2600	10	6350	111660
6V-53TC	5A55	235 @ 2600	175 @ 2600	7.5 + 1 Calif.	6350	111660
6-71N	7E65	230 @ 2100	172 @ 2100	10	7300	128365
	7E60	215 @ 2100	160 @ 2100	10	6950	122211
	7E55	200 @ 2100	149 @ 2100	10	6200	109022
	7E50	185 @ 2100	138 @ 2100	10	5700	100230
6-71NC	B60	210 @ 2100	157 @ 2100	7.5 + 1 Calif.	6100	107264
	B55	197 @ 2100	147 @ 2100	7.5 + 1 Calif.	5700	100230
	71B5	183 @ 2100	137 @ 2100	7.5 + 1 Calif.	5300	93197
6-71T	7C75	275 @ 2100	205 @ 2100	10	8850	155621
	7C70	260 @ 2100	194 @ 2100	10	8350	146829
	N65	245 @ 2100	183 @ 2100	10	7600	133640
6-71TT	7E75	230 @ 1950	172 @ 1950	10	6800 *	119573 *
* 6V-71N Coach (2 Valve Head)	7E60	190 @ 2100	142 @ 2100	10	5810	102165
	7E55	175 @ 2100	131 @ 2100	10	5500	96713
	7E50	160 @ 2100	119 @ 2100	10	4850	85284
8V-71N	7E65	304 @ 2100	227 @ 2100	10	9400	165292
	7E60	285 @ 2100	213 @ 2100	10	9050	159138
	7E55	265 @ 2100	198 @ 2100	10	8400	147708
	7E50	245 @ 2100	183 @ 2100	10	7950	139795

* Heat rejection @ 1800 rpm.

Engine	Injector	Rated Power		Emission Level	Heat Rejected to Coolant	
		BHP @ RPM	kW @ RPM	NO_x + HC	BTU/Min.	W
8V-71N Coach (4 Valve Head)	7E60	270 @ 2100	201 @ 2100	10	8500	149466
	7E55	250 @ 2100	187 @ 2100	10	7650	134520
	7E50	230 @ 2100	172 @ 2100	10	6950	122211
8V-71TA	7C75	370 @ 2100	276 @ 2100	10	12950	226837
	7C70	345 @ 2100	257 @ 2100	10	13100	230354
	N65	318 @ 2100	237 @ 2100	10	11150	196064
8V-71TTA	7C75	305 @ 1950	228 @ 1950	10	9900*	174084*
8V-71TAC	7A75	350 @ 2100	261 @ 2100	7.5 + 1 Calif.	13150	231233
	7A70	335 @ 2100	250 @ 2100	7.5 + 1 Calif.	12550	220682
	7A65	308 @ 2100	230 @ 2100	7.5 + 1 Calif.	11550	203098
	7A60	280 @ 2100	209 @ 2100	7.5 + 1 Calif.	10050	176722
	7A55	263 @ 2100	196 @ 2100	7.5 + 1 Calif.	9450	166171
	7A50	245 @ 2100	183 @ 2100	7.5 + 1 Calif.	8800	154741
8V-71TTAC	7A75	305 @ 1950	228 @ 1950	7.5 + 1 Calif.	9900*	174084*
8V-71TAC (Coach)	7A60	262 @ 2100	195 @ 2100	7.5 + 1 Calif.	9700	170567
	7A55	246 @ 2100	184 @ 2100	7.5 + 1 Calif.	9100	160017
	7A50	228 @ 2100	170 @ 2100	7.5 + 1 Calif.	8450	148587
6V-92TA	9B90	335 @ 2100	250 @ 2100	10	9900	174084
	9B85	315 @ 2100	235 @ 2100	10	9350	164413
	9B80	294 @ 2100	219 @ 2100	10	8800	154741
	9B75	282 @ 2100	210 @ 2100	10	8500	149466
	9B70	270 @ 2100	201 @ 2100	10	8250	145070
6V-92TTA	9B90	270 @ 1950	201 @ 1950	10	7600*	133640*
	9B90	250 @ 1800	187 @ 1800	10	7100*	124848*

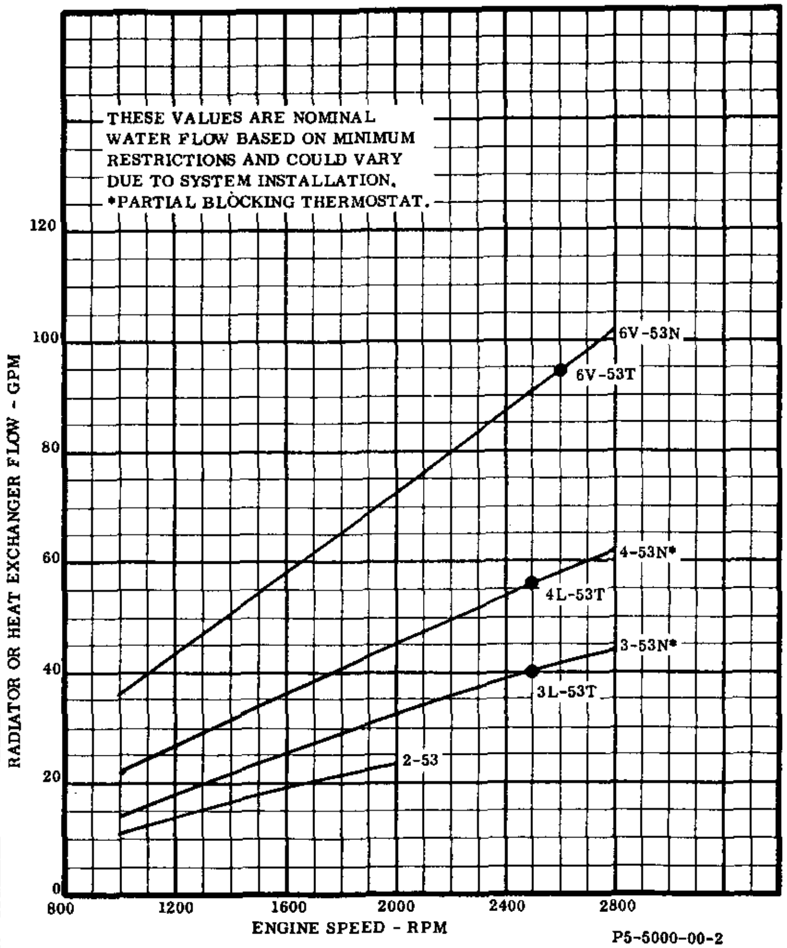
 (Continued)

Engine	Injector	--------Rated Power--------		Emission Level	---Heat Rejected to Coolant ---	
		BHP @ RPM	kW @ RPM	NO_x + HC	BTU/Min.	W
6V-92TAC	9B90	335 @ 2100	250 @ 2100	7.5 + 1 Calif.	10050	176722
	9B85	315 @ 2100	235 @ 2100	7.5 + 1 Calif.	9500	167050
	9B80	294 @ 2100	219 @ 2100	7.5 + 1 Calif.	9000	158258
	9B75	279 @ 2100	208 @ 2100	7.5 + 1 Calif.	8600	151225
	9B70	260 @ 2100	194 @ 2100	7.5 + 1 Calif.	8200	144191
6V-92TTAC	9B90	270 @ 1950	201 @ 1950	7.5 + 1 Calif.	7850 *	138036 *
	9B90	307 @ 1950	229 @ 1950	7.5 + 1 Calif.	8800 *	154741 *
6V-92TA (Coach)	9B75	277 @ 2100	207 @ 2100	10	8100	142433
6V-92TAC (Coach)	9B75	272 @ 2100	203 @ 2100	7.5 + 1 Calif.	8100	142433
8V-92TA	9A90	435 @ 2100	325 @ 2100	10	13500	237388
	9A85	408 @ 2100	304 @ 2100	10	12750	224199
	9A80	381 @ 2100	284 @ 2100	10	12000	211011
8V-92TTA	9A90	365 @ 1950	272 @ 1950	10	10400 *	182876 *
	9A90	335 @ 1800	250 @ 1800	10	9600 *	168809 *
8V-92TAC	9A90	430 @ 2100	321 @ 2100	7.5 + 1 Calif.	13750	241784
	9A85	403 @ 2100	301 @ 2100	7.5 + 1 Calif.	12600	221562
	9A80	375 @ 2100	280 @ 2100	7.5 + 1 Calif.	11800	207494
8V-92TTAC	9A90	365 @ 1950	272 @ 1950	7.5 + 1 Calif.	10950 *	192548 *
8.2	4A53	165 @ 3000	123 @ 3000	10	4950	87042
8.2C	4A53	165 @ 3000	123 @ 3000	6.0 California	5350	94076
8.2T	4A65	205 @ 3000	153 @ 3000	6.0 California	6300	110781

Detroit Diesel Allison
Division of General Motors Corporation

COOLANT FLOW THROUGH RADIATOR

MODEL:	SERIES 53
COOLANT TEMP.:	180°F
THERMOSTAT:	BLOCKING TYPE
PUMP DRIVE RATIO:	1.55:1



P5-5000-00-2
Rev. 7-19-79

ENGINEERING-TECHNICAL DATA DEPT.

DE 3769

Detroit Diesel Allison
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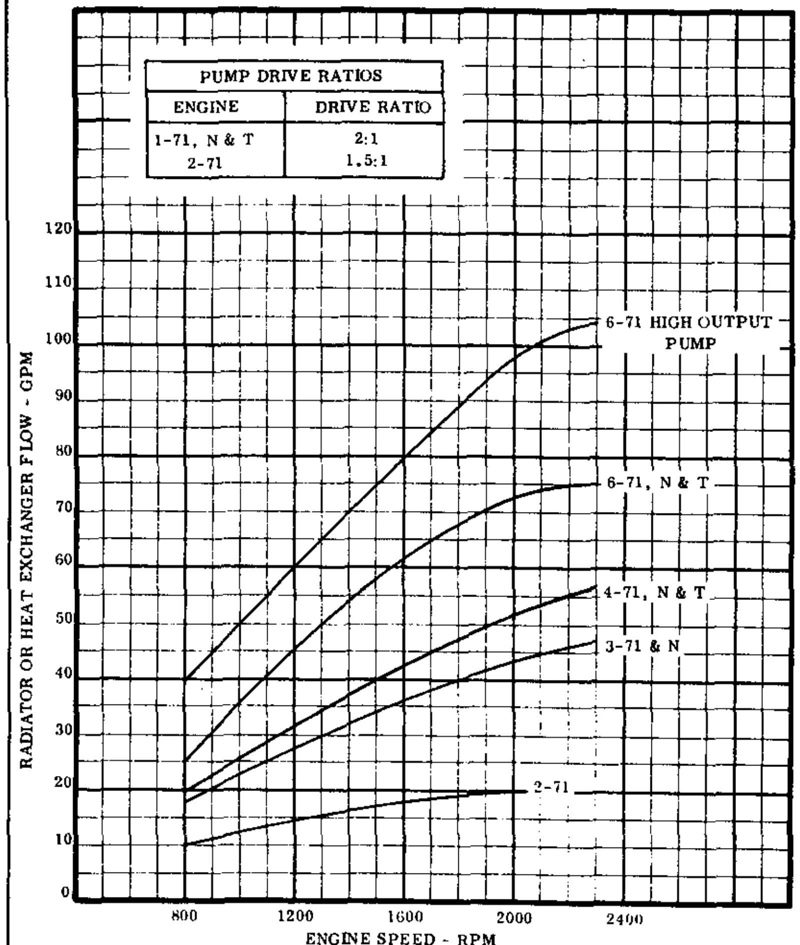
ENGINE COOLANT FLOW
THRU RADIATOR OR HEAT EXCHANGER
180°F TEMPERATURE
NON-BLOCKING TYPE THERMOSTAT



PUMP DRIVE RATIOS

ENGINE	DRIVE RATIO
1-71, N & T	2:1
2-71	1.5:1

P5-0000-00-2C
Rev. 8-18-72

ENGINEERING-TECHNICAL DATA DEPT.

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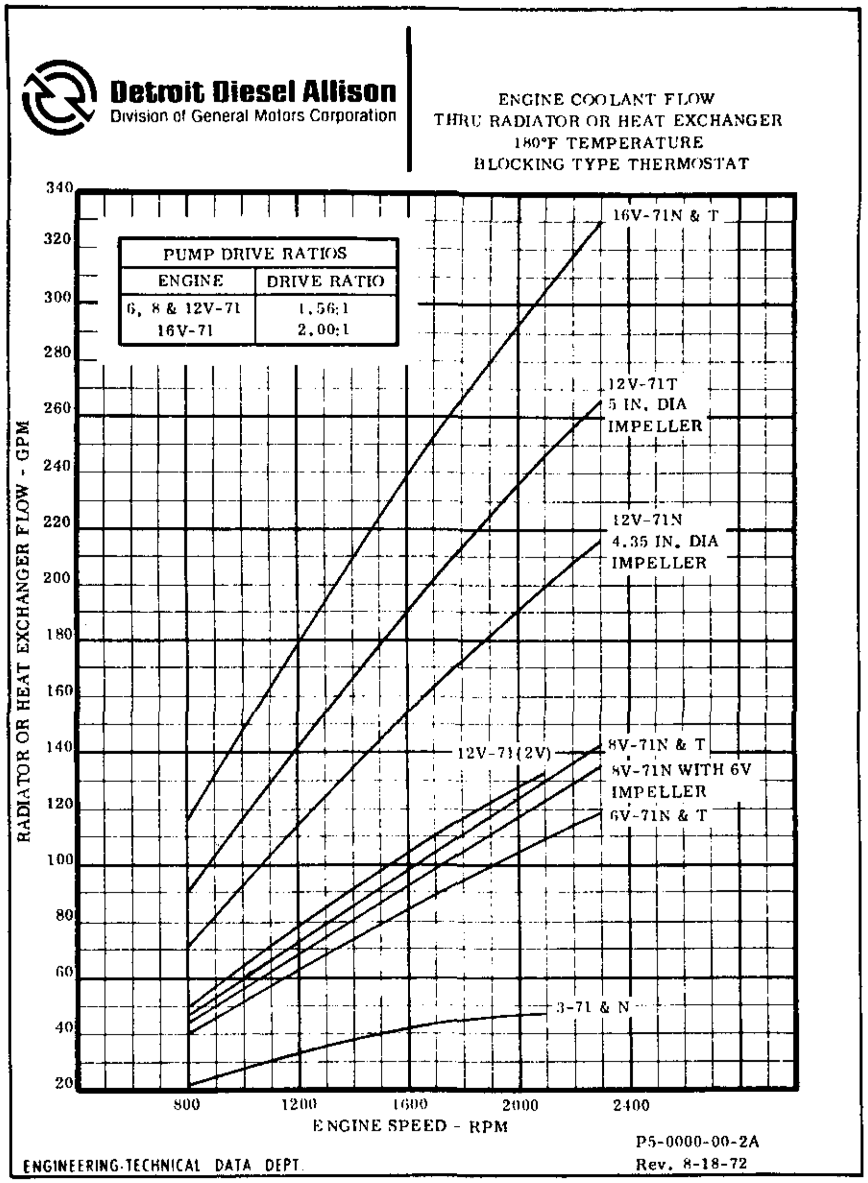
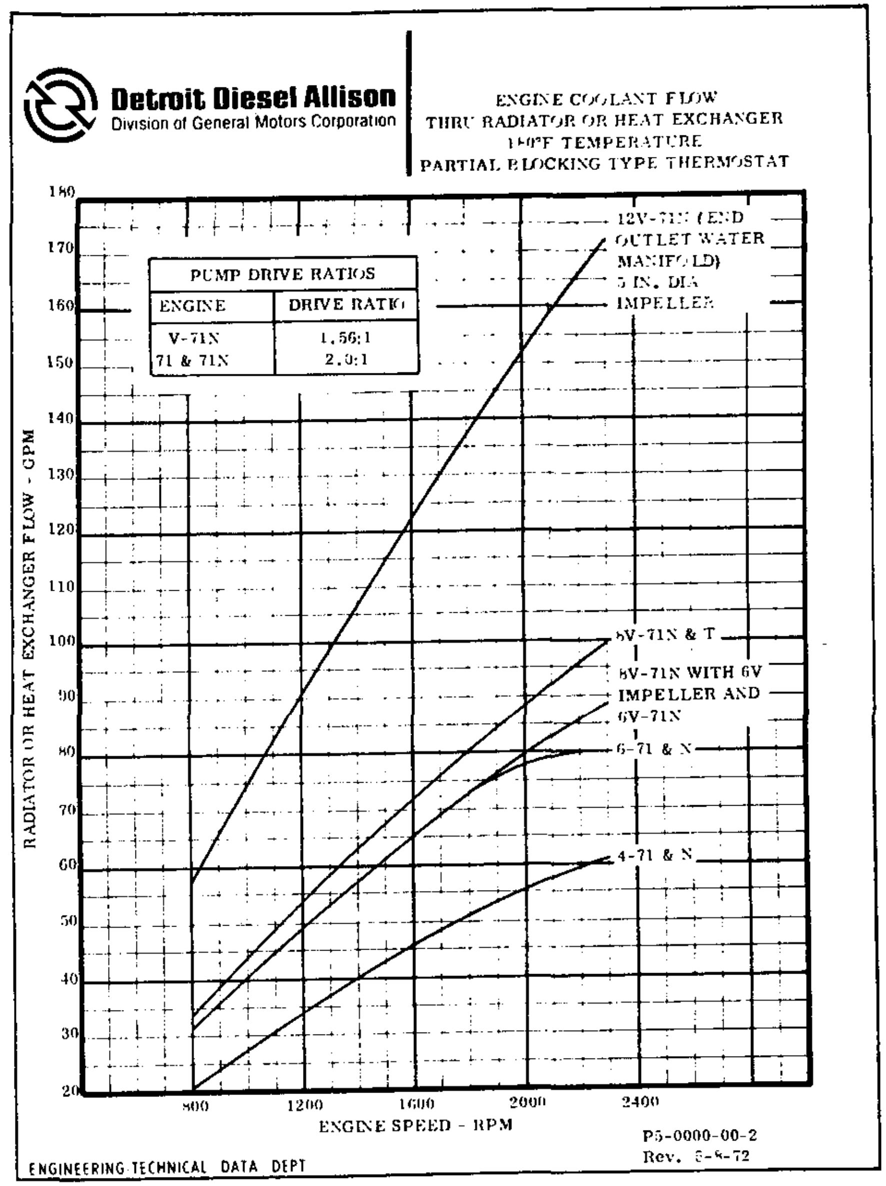


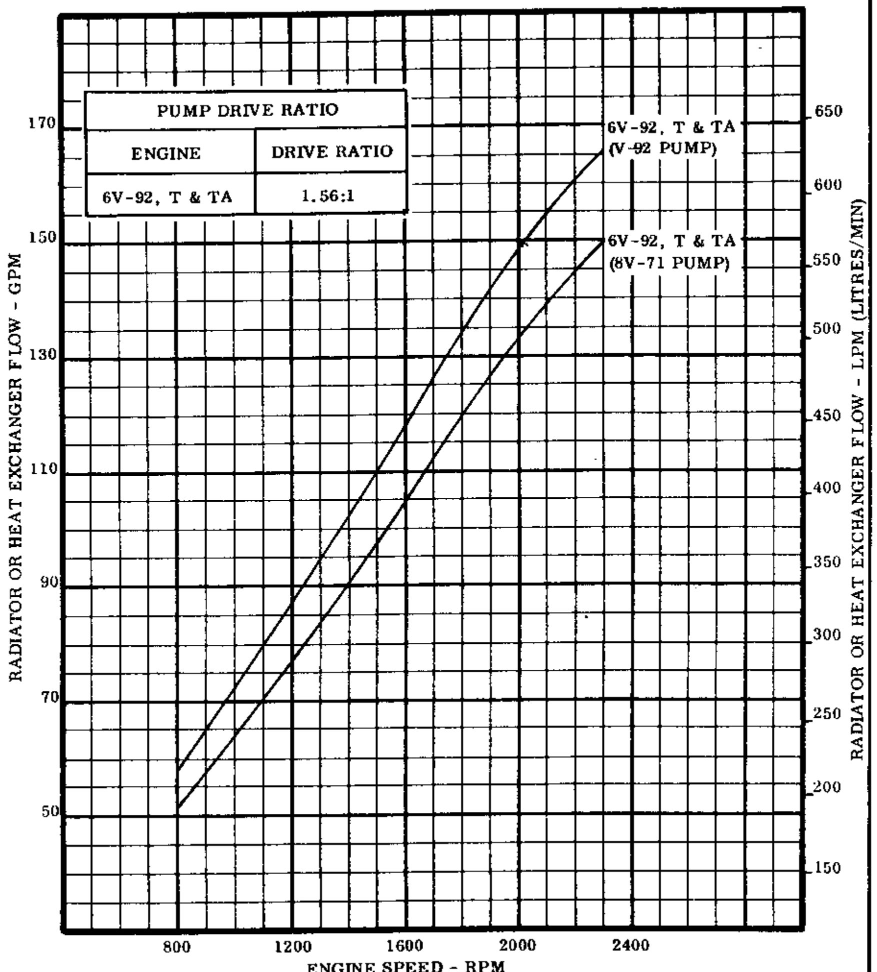


A-11 *(Continued)*

JW (9)

Detroit Diesel Allison
Division of General Motors Corporation

SERIES 92
6V-92, T & TA ENGINE COOLANT FLOW
THRU RADIATOR OR HEAT EXCHANGER
BLOCKING TYPE THERMOSTAT
180°F (82°C) ENGINE WATER OUT TEMPERATURE

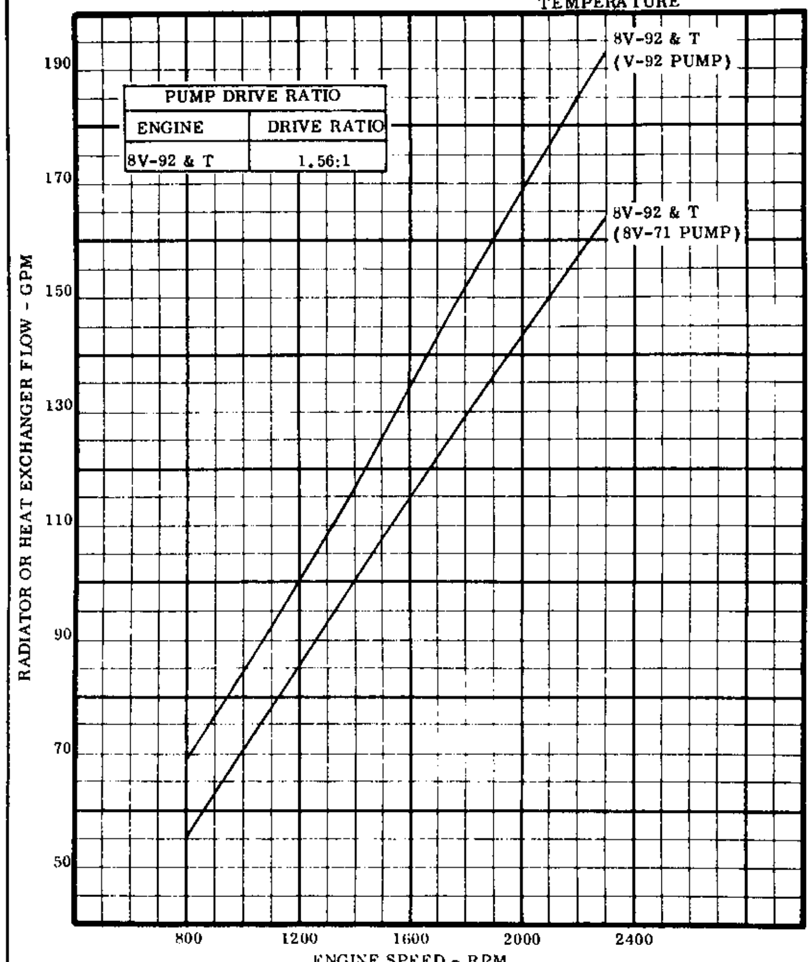


P5-0000-00-8
Rev. 4-9-75

ENGINEERING-TECHNICAL DATA DEPT.

Detroit Diesel Allison
Division of General Motors Corporation

SERIES 92
8V-92 & T ENGINE COOLANT FLOW
THRU RADIATOR OR HEAT EXCHANGER
BLOCKING TYPE THERMOSTAT
180°F (82°C) ENGINE WATER OUT TEMPERATURE



P5-0000-00-7
Rev. 11-29-79

ENGINEERING-TECHNICAL DATA DEPT.

DE 3769

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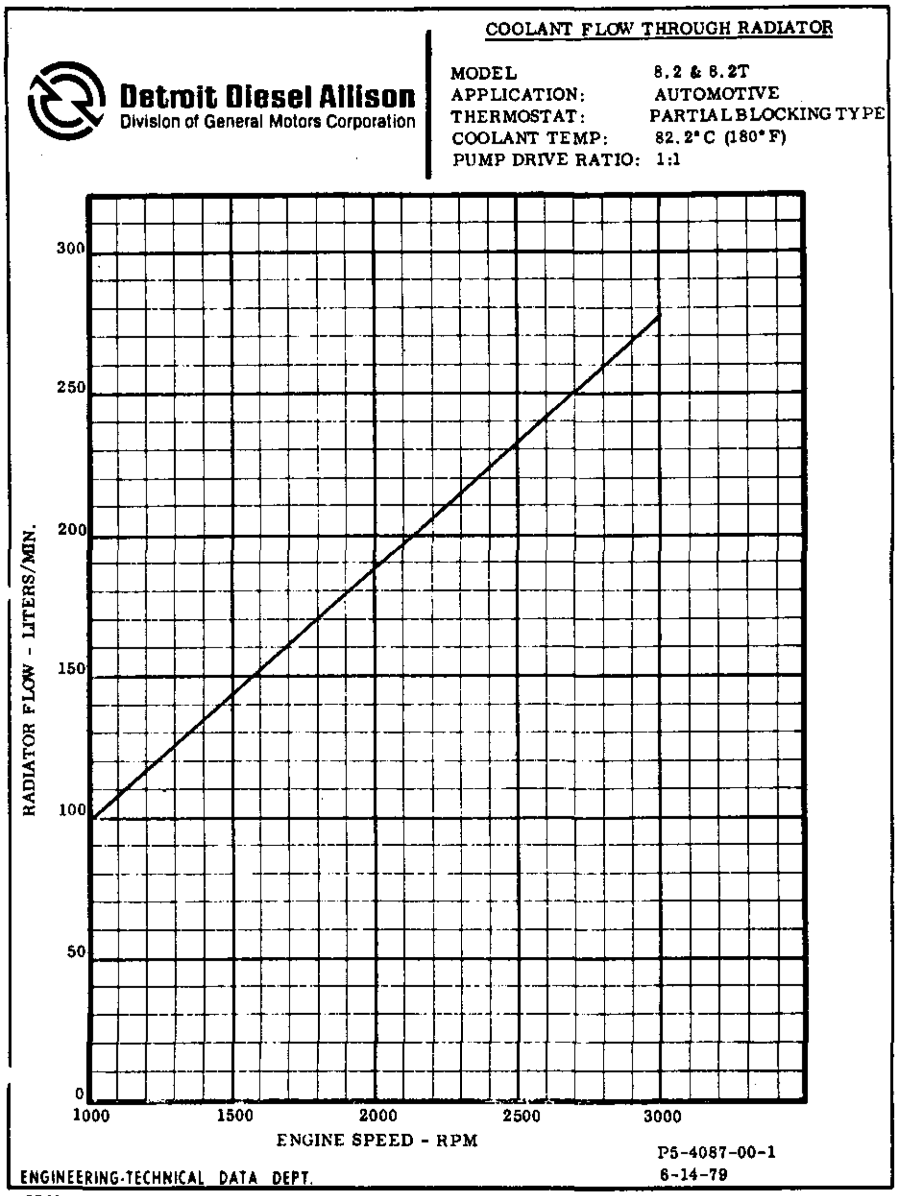

A-12 *(Continued)*