Detroit Diesel Allison
Division of General Motors Corporation

COOLING

OF

DETROIT DIESEL ENGINES

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Engineering Technical Data Dept.

ENGINEERING BULLETIN No. 28

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COOLING BULLETIN INDEX

INTRODUCTION

The operation of Detroit Diesel Engines, like all other internal combustion engines, produce heat during the combustion of fuel. Without proper cooling this heat can be detrimental to the life and efficiency of the engine. The energy developed from combustion of the fuel breaks down into four categories: work, exhaust, radiation and heat rejected to the coolant. Figure 1, shows a typical engine heat balance with the total energy developed being based on the heating value of the fuel input at full load.

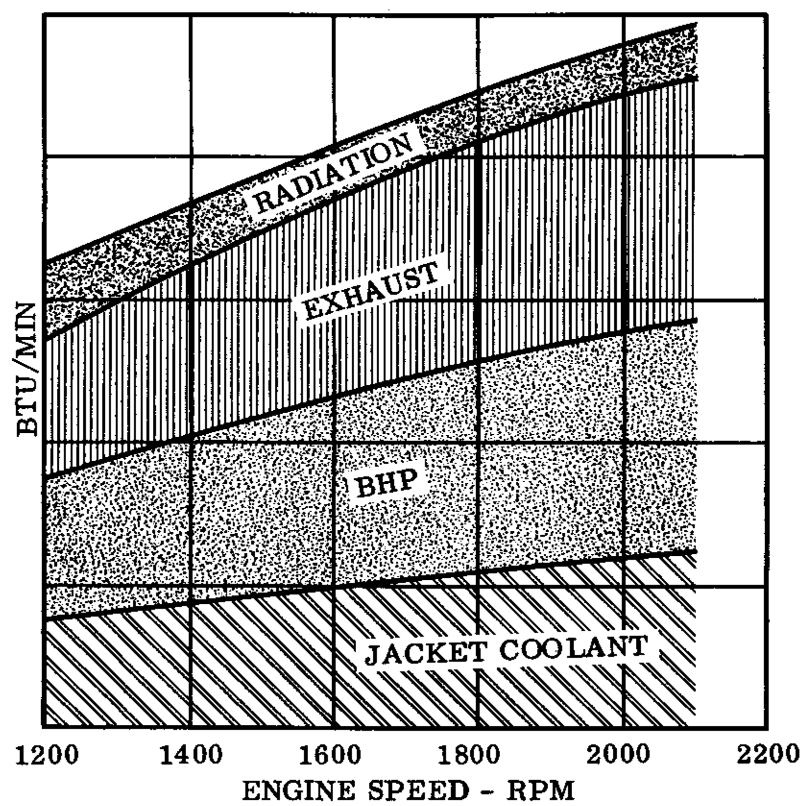


Figure 1.

Approximately one third of the fuel energy is converted to useable power, one third is exhausted to the atmosphere and the remaining third must be dissipated by the cooling system with a small portion removed by radiation.

This bulletin deals primarily with the heat rejected to the jacket coolant as shown on Figure 1. The heat transfer is accomplished by circulating coolant through the jacketed engine components and oil cooler. This heat is then dissipated through a radiator or heat exchanger to the environment. The following recommendations are made for designing and checking a cooling system, with adequate dissipation for Detroit Diesel Engines.

I. DESIGN OF COOLING SYSTEMS

A. SOURCE OF HEAT AND HEAT TRANSFER

The primary source of heat is the engine, however the cooling system heat load will be increased with the addition of accessories connected to the engine coolant circuit or with the addition of accessory radiators, for cooling other components, which utilize engine fan air flow.

1. ENGINE

Heat is transferred directly to the coolant as it passes through the engine coolant jacket and oil cooler. The specific heat rejection for Detroit Diesel Engines with dry exhaust manifolds is shown in Chart I.

CHART I.

SERIES	HEAT REJECTION (BTU/HP/MIN)
53 & 53N	33
71	35
71E & 71N	33
71T & V71T	33
V71 & V71N	30

(a) ENGINE OPERATING TEMPERATURES

Coolant out temperatures will range from 160 to 185°F depending on the thermostats used. The design temperature differential between the coolant inlet temperature and outlet temperature is 10°F at maximum engine speed and load. For torque converter applications on industrial or earthmoving engines the differential should not exceed 14°F because of the added heat load, from the converter oil cooler. This does not apply however to automotive (on Highway) vehicles with MT or similar transmissions which operate in lock up the majority of the time.

The cooling system should be designed to operate between thermostat control and 200°F maximum coolant outlet temperature when running at full throttle and maximum speed or at 70% efficiency with a torque converter in 120°F ambients and 500 Ft. elevation without the aid of a pressure cap. The maximum outlet temperature of 200°F is governed by sump oil temperatures which run from 30 to 45°F above engine coolant outlet temperature. At elevated temperatures, oil oxidation is accelerated resulting in an increased rate of contamination and possible engine damage. A cooling system operating above 200°F engine coolant outlet temperature is not considered satisfactory even though coolant boiling may not occur.

(1) HIGH AMBIENTS

At high ambients or desert conditions, the cooling system must not exceed a maximum coolant out temperature of 200°F. If it appears impractical to stay within this limit, engineering approval of the proposed system should be obtained.

Temperature differentials applicable to all engines which will provide adequate cooling for typical operating conditions and a safety factor against cooling system deterioration are tabulated in the following chart.

CHART II.

Operating Temperatures			
Application	Water Out Temperature	Ambient Temperature	Air to Water Differential
General Application	200°	115°	85°
Trucks & Busses	200°	110°	90°
Off Highway Equipment	200°	120°	80°
Power Units	200°	115°	85°
Generator Sets	200°	120°	80°
High Ambient or Desert Operations	200°	140°*	60°

*See discussion on High Ambients above.

(b) COOLING INDEX

The cooling index is the numerical value of the capabilities of a given cooling system. The cooling index is stated in one of two ways:

(1) AIR TO WATER DIFFERENTIAL

The A/W differential is the difference between engine coolant out or top tank temperature and the ambient air temperature. For example with a stabilized top tank temperature of 185°, and with air entering the radiator at 100°, the differential is 185°-100° = 85°.

(2) AIR TO BOIL

The ATB figure represents the ambient air temperature at which top tank boiling will occur. The boiling point should always be considered as 212°. For example the same engine as above at 185° in 100° air is 212°-185° = 27°+100° ambient = 127° ATB. It is automotive practice to use the boiling point of the coolant at the system pressure, however for heavy duty diesel engines a cooling system that is dependent on a pressure cap is not recommended and usually will exceed the 200°F maximum coolant out limit.

2. TORQUE CONVERTERS

When the torque converter is cooled with jacket coolant the total heat load to the cooling system is that of the engine plus that rejected by the converter. The temperature recommendations as previously described should apply when the converter operates at efficiencies as low as 70%.

The additional heat load from the converter is calculated as follows:

Heat Rejected (BTU/MIN) = 42.5 x (Net Engine HP-Shaft HP)

This calculation is normally made at the 70% converter efficiency point which is considered to be the worst sustained operating condition normally encountered. Full stall heat load may be two and three times the heat load at 70% converter efficiency. However additional oil cooling capacity is needed if stalling for periods longer than one minute is expected.

3. TORQMATIC BRAKES

The design limitation can be but briefly covered in this bulletin as this is a highly specialized field and cooling requirements will vary with gross vehicle weights, down grade speeds, percent of grade and other particulars of vehicle design and operation.

The heat exchanger may be added to the engine system between the discharge from the cylinder head and the radiator top tank, as shown in figure 2, page 9. This depicts a high mounted, horizontal, heat exchanger which must be below the quiescent water level in the radiator top tank to avoid air entrapment. Engine thermostats must be located in a suitable casting on the discharge of the heat exchanger and the thermostats at the cylinder head outlet removed. The coolant bypass line should be connected to the heat exchanger thermostat housing to permit by pass flow through the exchanger when the thermostats are closed. This prevents overheating of oil and coolant in the heat exchanger. During closed stat operation proper venting should be provided to insure air bleeding during filling.

The heat exchanger may also be connected between the engine oil cooler and the discharge side of the water pump for V-71 series engines. Shown on Figure 3, page 10, is an engine mounted tube and shell engine and torqmatic brake oil cooler. The torqmatic brake cooler may be remote mounted as shown on Figure 4, page 11. In both coolant pump flow passes through the coolers with both closed and open thermostats.

4. OIL TO AIR COOLING

When oil to air coolers are mounted in front of the radiator a blower fan should be used. When a suction fan is used the cooler should be mounted between the radiator and fan. If this is not done, the radiator must have a greater cooling capacity to accommodate the higher entering air temperature resulting from the air passing through the cooler before it enters the radiator. Both blower and suction fans should have higher delivery capacity when these coolers are used as the added restriction in addition to normal radiator restriction reduces air flow.

5. OTHER ITEMS COOLED BY ENGINE COOLANT

The above mentioned items contribute the major amount of heat rejected to the coolant. The following are the minor contributors, if they are used on an engine.

(a) COOLANT COOLED EXHAUST MANIFOLD

When a coolant cooled exhaust manifold is used, additonal heat is absorbed by the engine coolant. The added heat load is 4 BTU/HP/MIN which must be added to the engine rejection.

(b) MARINE GEAR COOLER

The heat added by the marine gear is approximately 2% of the total heat load of the engine. This is considered to be a negligible amount and is not usually considered.

(c) AIR COMPRESSORS (Engine Mounted)

The heat added is less than that for the marine gear and need not be considered in the overall heat load.

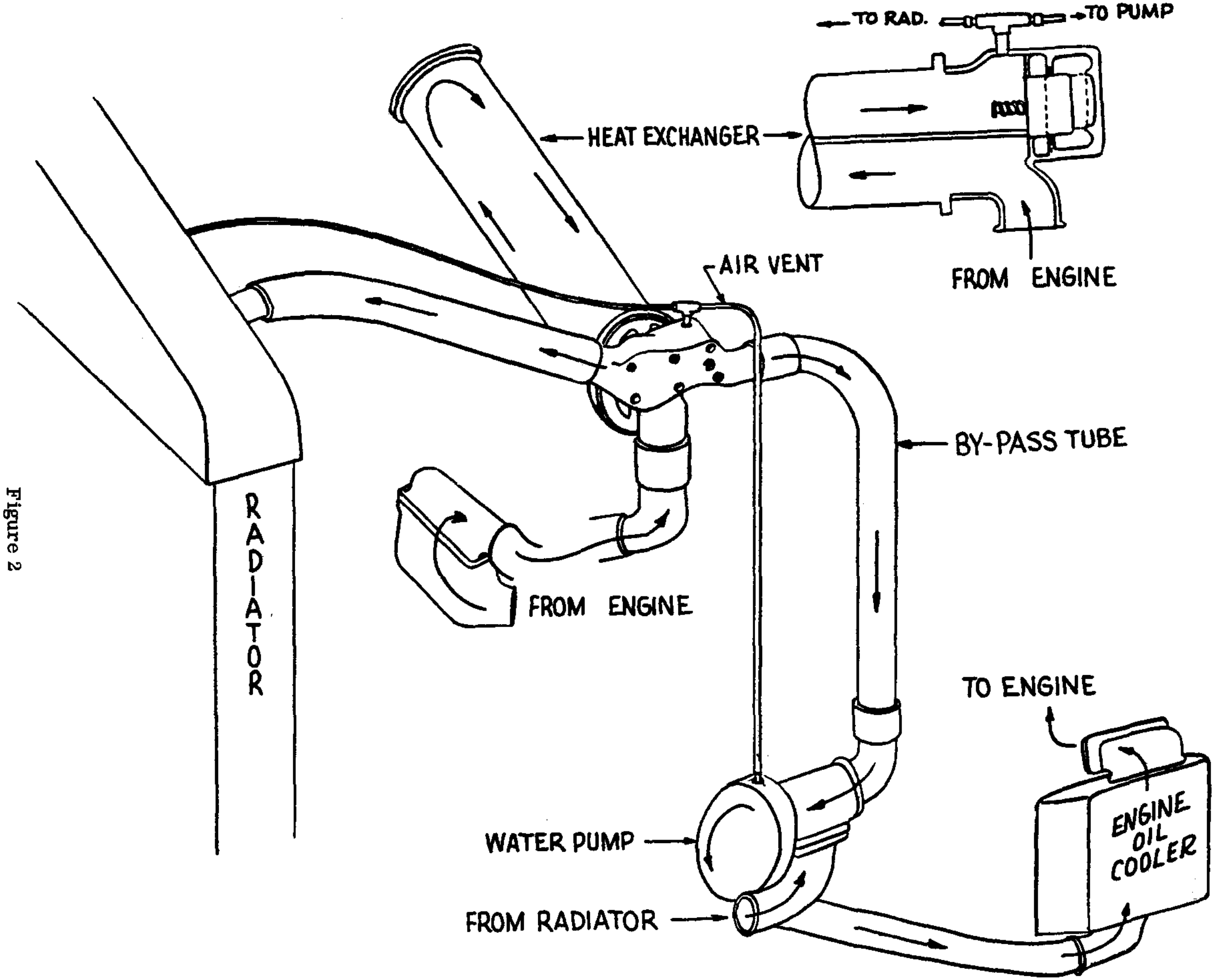
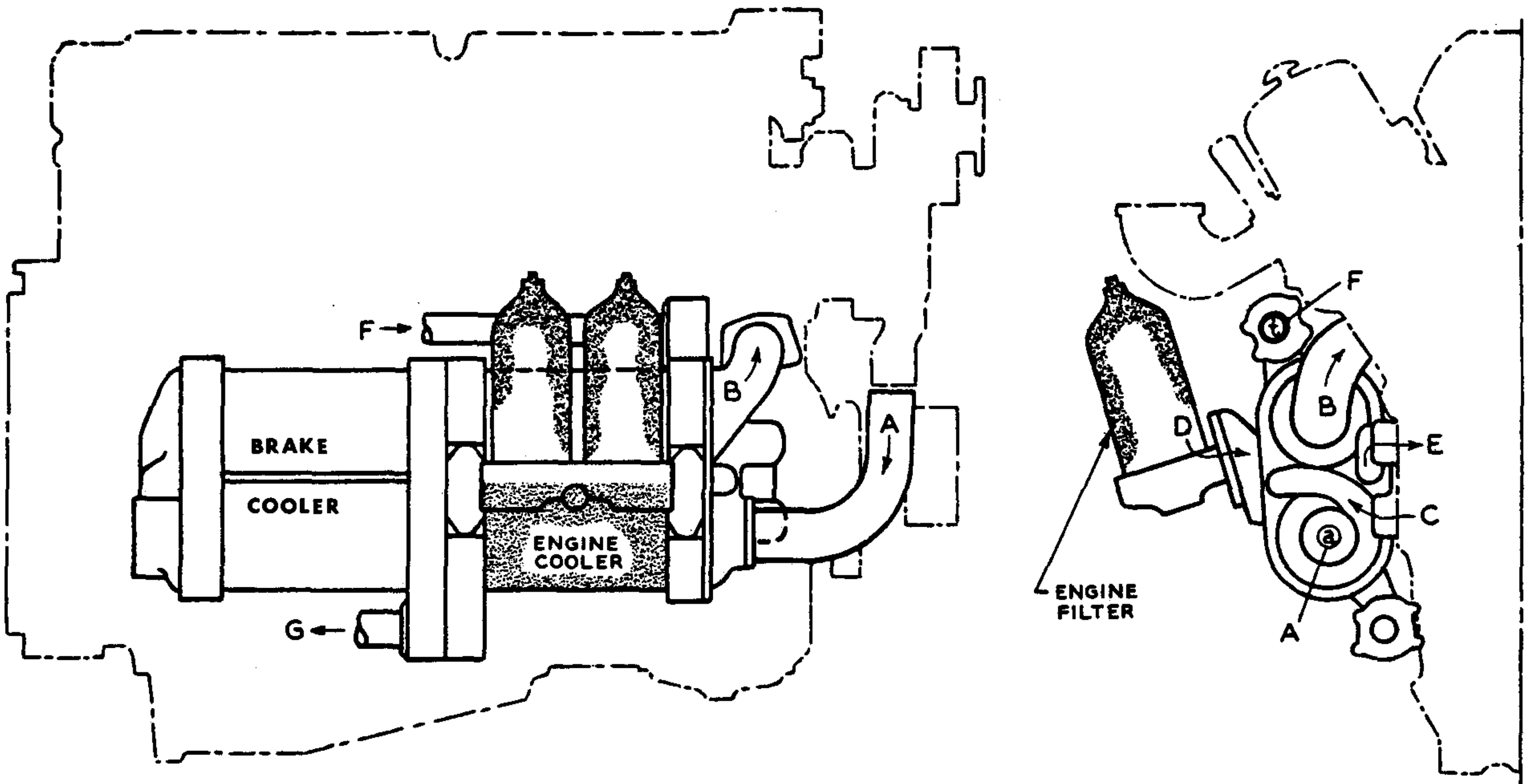


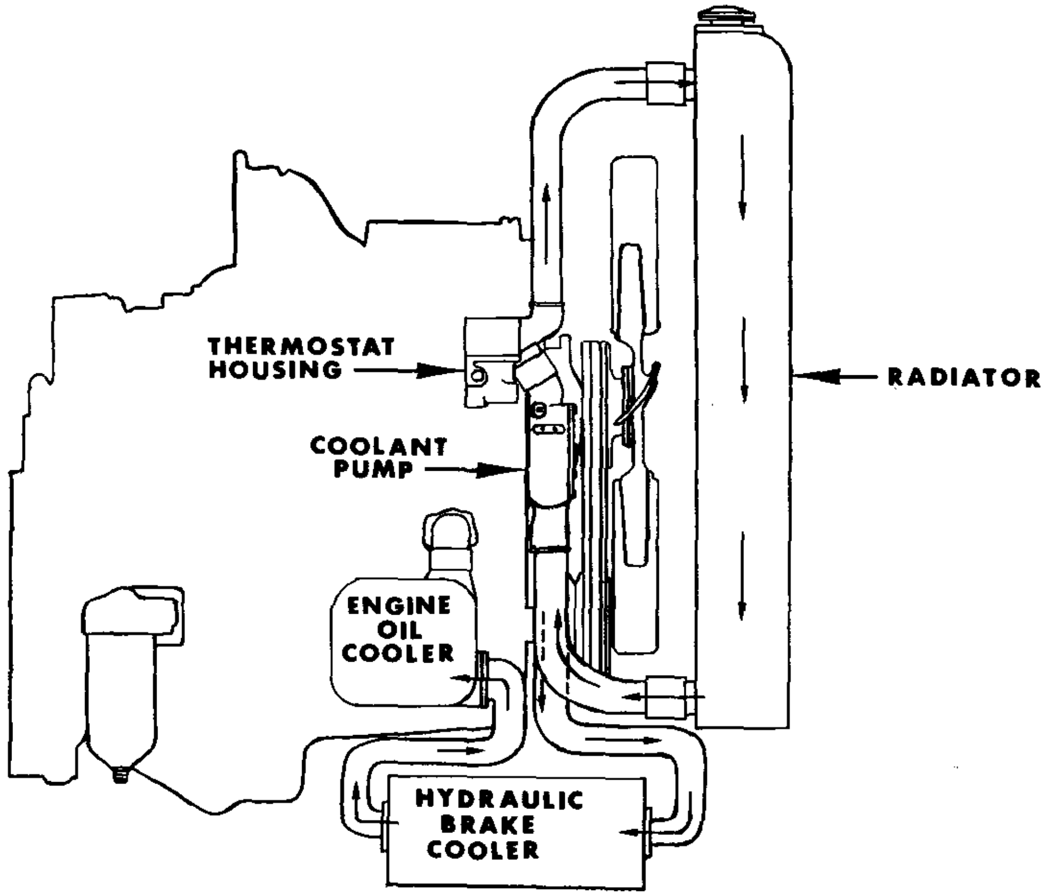
Figure 2
Page 9

TORQMATIC BRAKE COOLER



L E G E N D

- · — · - Engine Outline
——— Cooling System
ⓐ — Arrow denoting flow away from viewer.
ⓣ — Arrow denoting flow toward viewer.
A — Water from Pump to Oil Cooler
B — Water from Oil Cooler to Engine
C — Engine Oil from Engine to Full Flow Filter
D — Engine Oil from Filter to Cooler
E — Engine Oil from Cooler to Engine
F — Torque Brake-Oil from Torque Brake to Cooler
G — Torque Brake-Oil from Cooler to Torque Brake



REMOTE MOUNTED BRAKE COOLER

Figure 4

B. COOLANT

1. ENGINE COOLANT AND RECOMMENDED CONNECTION SIZES

To aid in selecting radiators and heat exchangers the coolant flows for Detroit Diesel Engines are shown on Figures 5, 6, 7 and 8 pages 13, 14, 15 and 16 respectively. The curves show flows available with various types of thermostats that are used. These flows are to be considered as nominal values since variations will occur because of coolant temperature and restrictions added to the cooling system.

The following table shows engine coolant connection sizes. Nothing less than these sizes are to be used for tubing and connections to the radiator.

CHART III

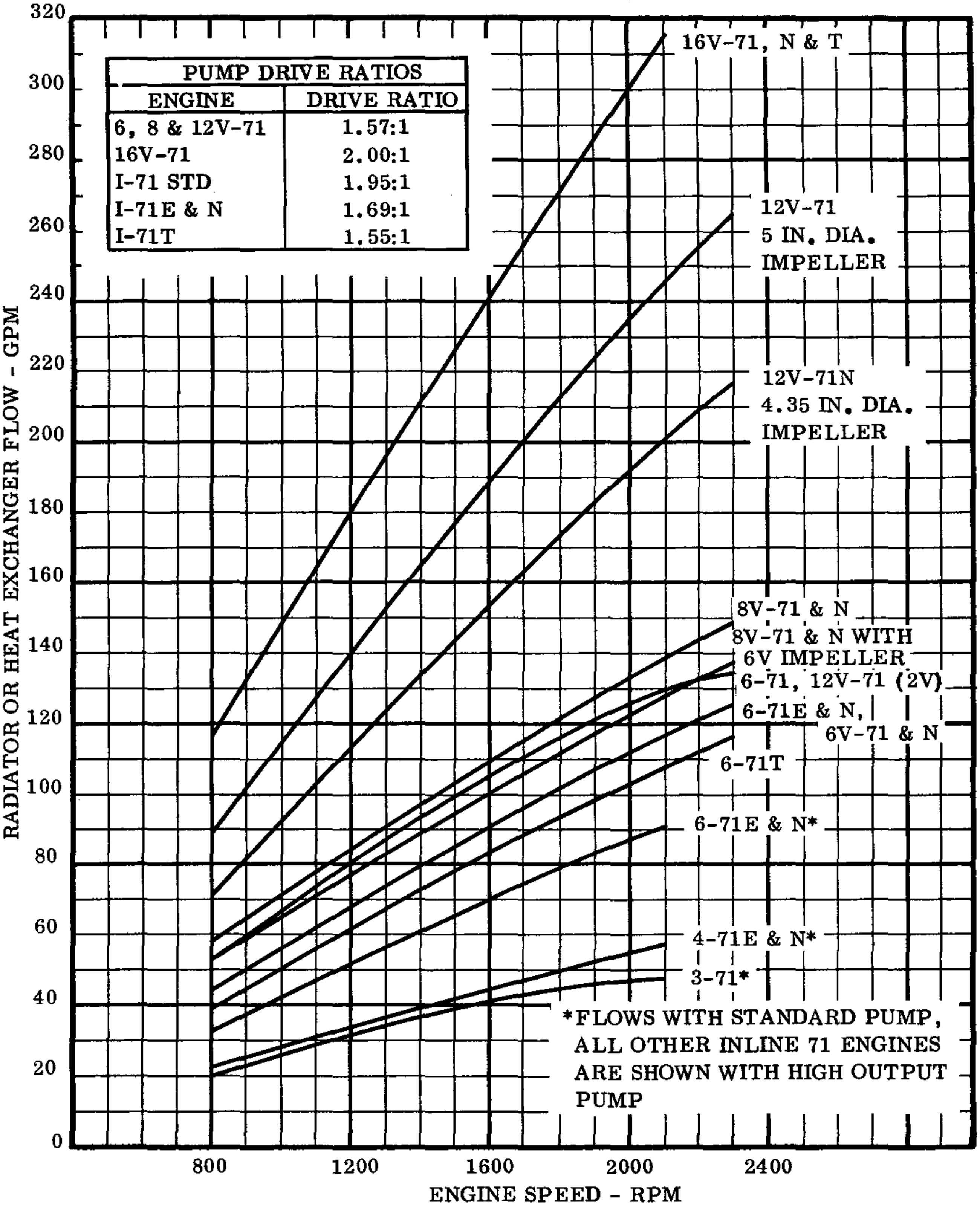
Model	Engine Coolant Inlet Connection	Engine Coolant Outlet Connection	
		Single	Double
Series 53			
2, 3, 4 & 6V-53	1 7/8 Inches	1 5/8 Inches	1 1/8 Inches
8V-53N	2 3/8 Inches	-	2 1/4 Inches
Series 71			
2-71	1 3/4 Inches	1 1/2 Inches	-
3, 4 & 6-71	2 Inches	1 7/8 Inches	-
Series V71			
6 & 8V-71	2 3/4 Inches	2 1/2 Inches	1 7/8 Inches
12V-71	3 1/2 Inches	3 Inches	2 1/8 Inches
16V-71	4 Inches	-	3 Inches

Double outlet connections are strongly recommended for "V" type engines. Where a single connection is made on "V" engines the tube or hose to the radiator should be equivalent in area to both tubes from the right and left bank.

DETROIT DIESEL ENGINE DIVISION

GENERAL MOTORS CORPORATION

ENGINE COOLANT FLOW
THRU RADIATOR OR HEAT EXCHANGER
180°F TEMPERATURE
BLOCKING TYPE THERMOSTAT

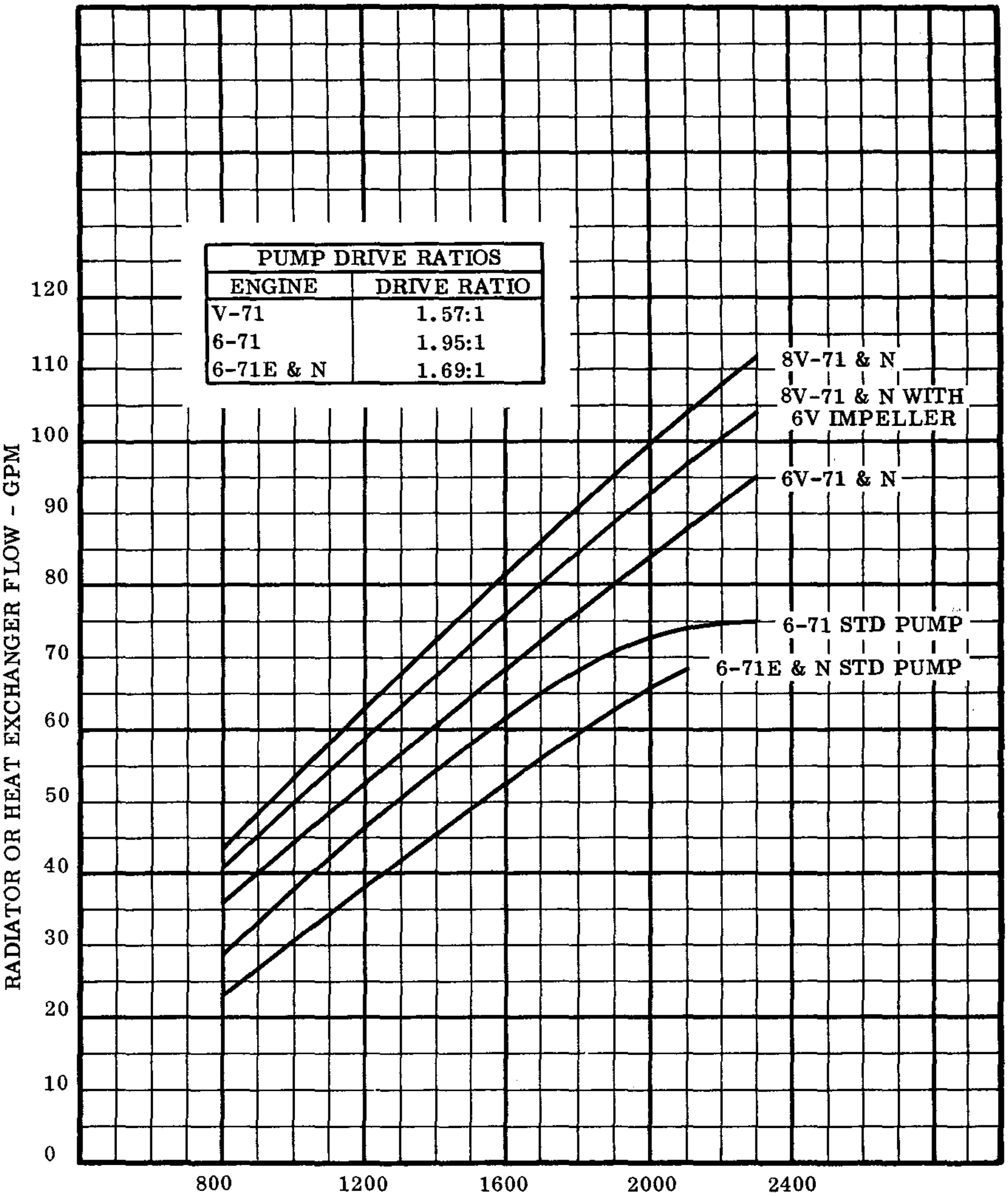


PUMP DRIVE RATIOS	
ENGINE	DRIVE RATIO
6, 8 & 12V-71	1.57:1
16V-71	2.00:1
I-71 STD	1.95:1
I-71E & N	1.69:1
I-71T	1.55:1

Figure 5

ENGINEERING-TECHNICAL DATA DEPT.

ENGINE COOLANT FLOW
THRU RADIATOR OR HEAT EXCHANGER
180°F TEMPERATURE
PARTIAL BLOCKING TYPE THERMOSTAT



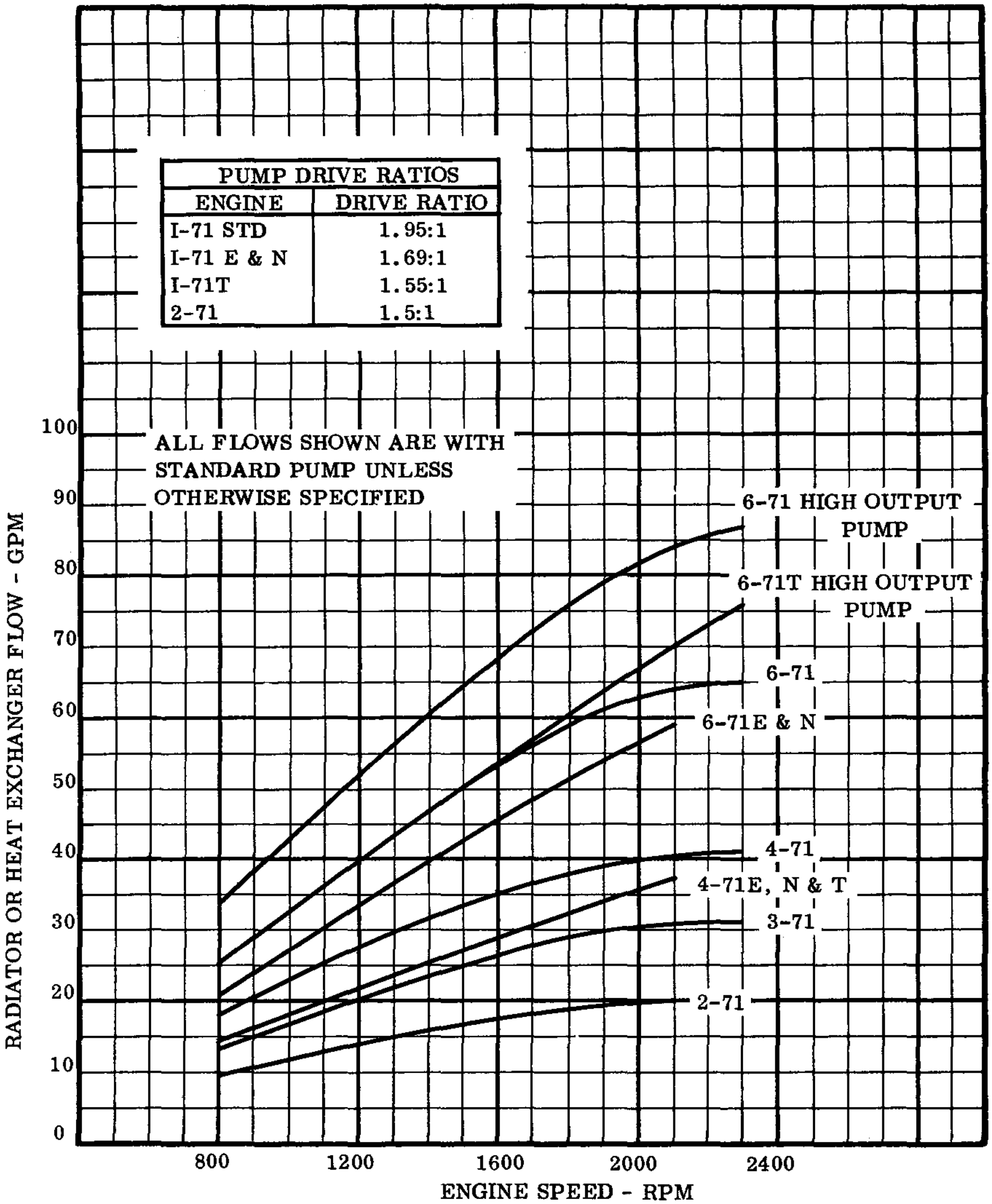
PUMP DRIVE RATIOS	
ENGINE	DRIVE RATIO
V-71	1.57:1
6-71	1.95:1
6-71E & N	1.69:1

Figure 6

DETROIT DIESEL ENGINE DIVISION

GENERAL MOTORS CORPORATION

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



PUMP DRIVE RATIOS	
ENGINE	DRIVE RATIO
I-71 STD	1.95:1
I-71 E & N	1.69:1
I-71T	1.55:1
2-71	1.5:1

Figure 7

ENGINEERING-TECHNICAL DATA DEPT.

DETROIT DIESEL ENGINE DIVISION

GENERAL MOTORS CORPORATION

SERIES 53

ENGINE COOLANT FLOW

THRU RADIATOR OR HEAT EXCHANGER

180°F TEMPERATURE

BLOCKING TYPE THERMOSTAT

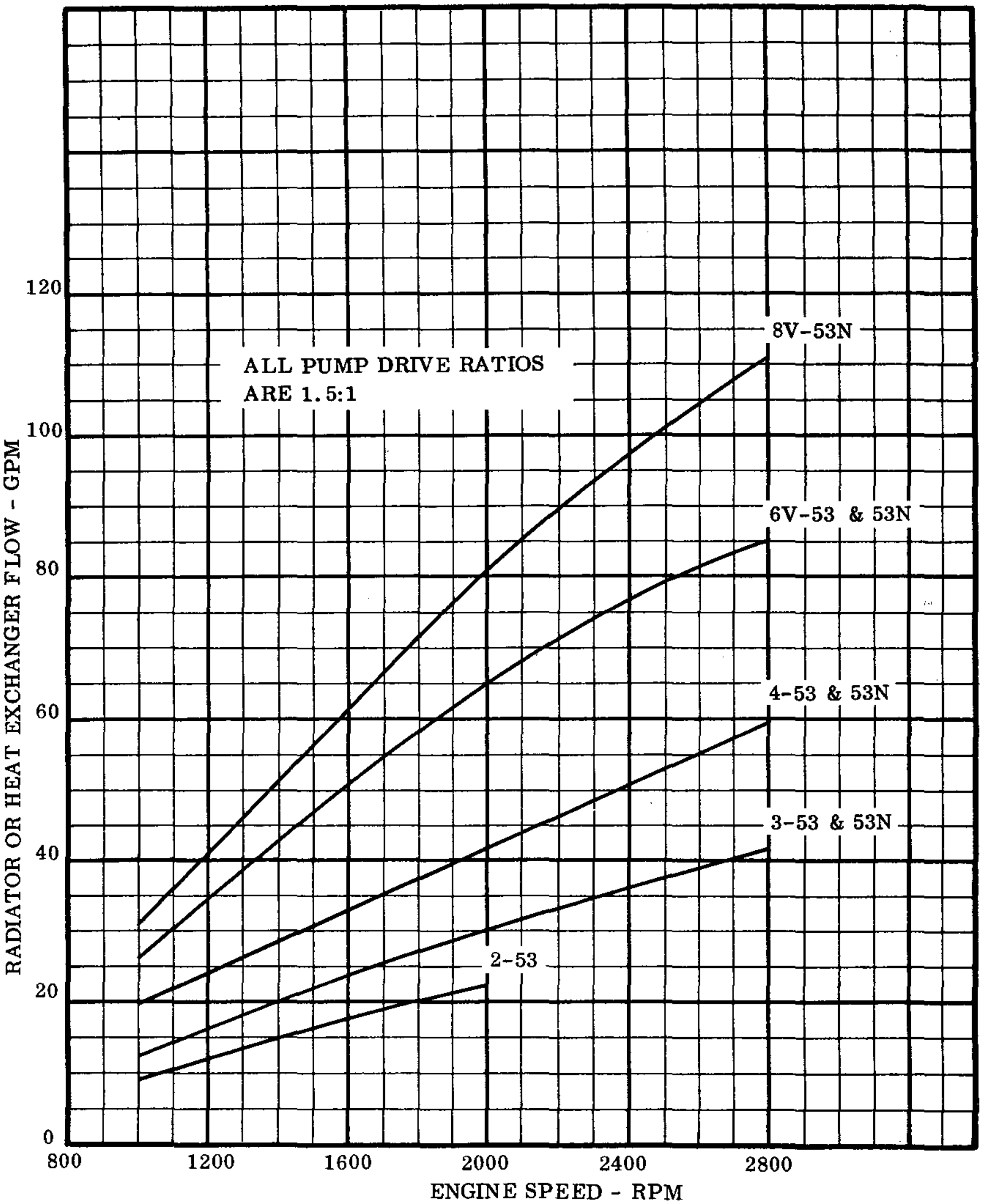


Figure 8

ENGINEERING-TECHNICAL DATA DEPT.

2. ITEMS AFFECTING PUMP FLOW

The centrifugal coolant pump is sensative to inlet restrictions and discharge flow can be seriously reduced by poor design of coolant plumbing. High inlet restriction will result in pump damaging cavitation and possible engine damage because of the resulting reduction in coolant flow.

(a) HOSE SIZES

Connections between the radiator and engine should be made with largest practical size hose. Special attention must be given to the coolant pump inlet connections to keep from inducing high pump restriction with a small hose size or unnecessary bends. See Chart III on page 12 for recommended sizes.

The hose and clamps used should also be of such quality as to withstand the maximum suction, pressures and temperatures encountered in the cooling system, especially where a pressure cap is used.

(b) REMOTE LOCATIONS

Where remote mounting of radiators or heat exchangers is necessary, consult the factory or distributor as there are many variables which can only be determined from a specific application. In any case particular attention must be given to piping to insure compliance with recommended pump suction.

When radiators are remote mounted automatic controls should be installed to insure fan operation when the engines are running. When two or more engines are connected to a common cooling system care should be taken to insure adequate flow to all engines which are in operation. Appropriate valving should also be installed to prevent the hot coolant, from an operating engine, being circulated through a companion unit instead of the heat exchanger.

(c) DEAERATION

A critical function of the cooling system which is often overlooked is its ability to remove entrained gases from the coolant or the deaeration ability of the system. Small amounts of combustion gases may find their way into the cooling system during operation. Air may be induced via the pump seal or air within the expansion cavity of the radiator top tank or heat exchanger tank may be re-introduced into the system. Aerated coolant can produce localized areas of overheat and result in engine damage which is not immediately evident. Pump flow will also be reduced with aerated coolant and may become "air bound" and cease to circulate coolant. Serious engine damage could occur before the problem was evidenced by warning or protective systems which generally require coolant flow for reliable operation.

Proper design of radiator top tank or surge tank and heat exchangers are most important to provide adequate coolant deaeration.

(d) COOLANT PUMP SUCTION

To insure proper pump flow and operation the suction at the engine coolant inlet should not exceed 1.5 PSI or 3 in. Hg vacuum within the normal operating range of 160-185°F coolant out temperature.

The recommended tap location to check pump suction is at the engine coolant inlet, or in the line just before the coolant inlet. This check must be done without a pressure cap on the system to obtain valid readings. See Figures 9, 10, 11, & 12 on pages 20 and 21 for typical engine coolant inlet locations.

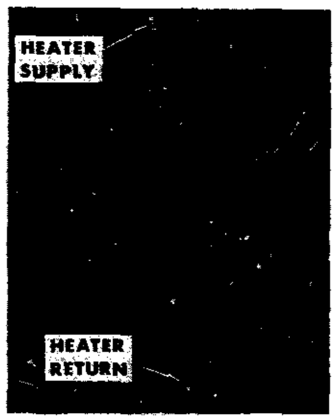
(e) TEMPERATURE CAVITATION

Operating above 185°F coolant out temperature with a non-pressurized system will decrease pump flow because of the pump tendency to cavitate at higher coolant temperatures. This factor becomes more critical when the engine is operating at altitude and will require a pressure cap to raise the boiling point of the coolant.

3. HEATER CONNECTIONS

Heater hose connections vary to suit design requirements of the application. The primary objective is to obtain sufficient pressure differential across the heater to provide adequate coolant flow. The following recommended locations are offered as a guide to efficient heater performance.

On "71" In-Line Engines, the heater supply hose is usually connected to the coolant manifold or cylinder block and the return hose to the oil cooler housing.

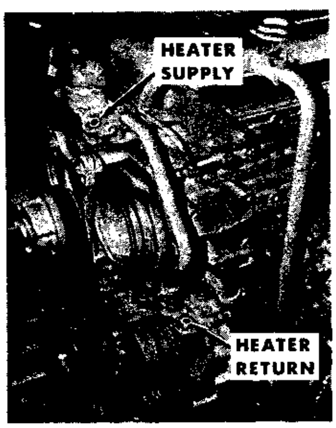


"71" In-Line Engine

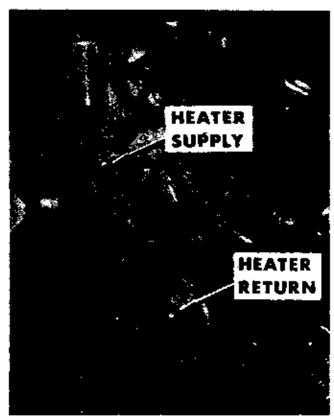
On "V71" Engines, the heater supply hose is usually connected to the oil cooler inlet, or the side or end of the cylinder block. The return hose is connected to the suction side of the coolant pump.

"V-71" Engine

On "53" and "V-53" Engines the heater supply hose is usually connected to the thermostat housing before the thermostat and the return hose to the suction side of the coolant pump.

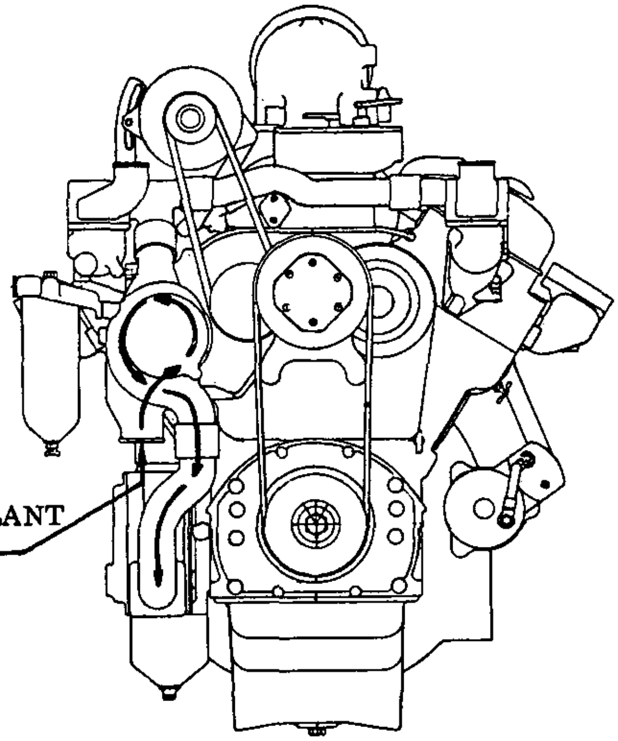


"53" In-Line Engine

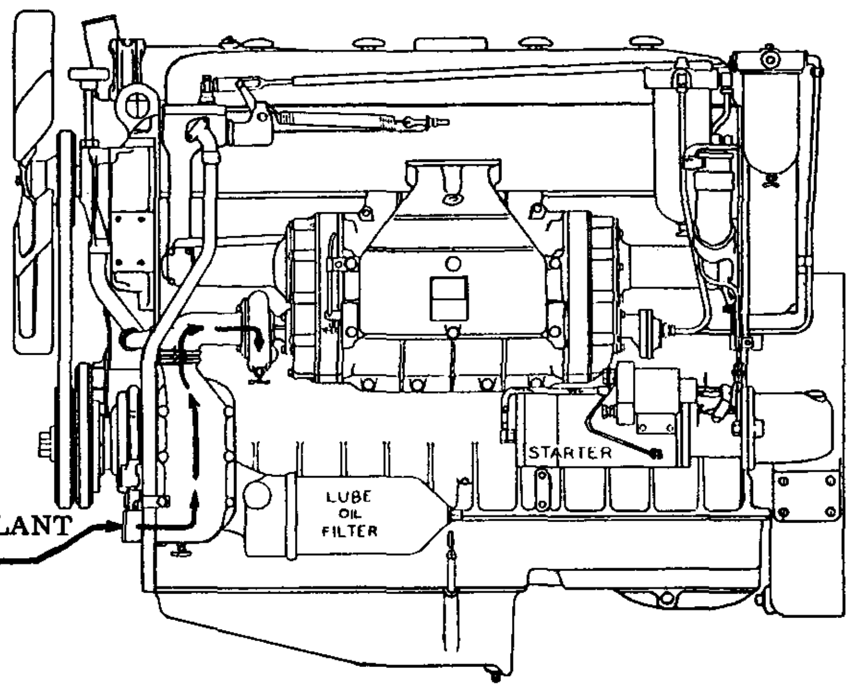


"V-53" Engine

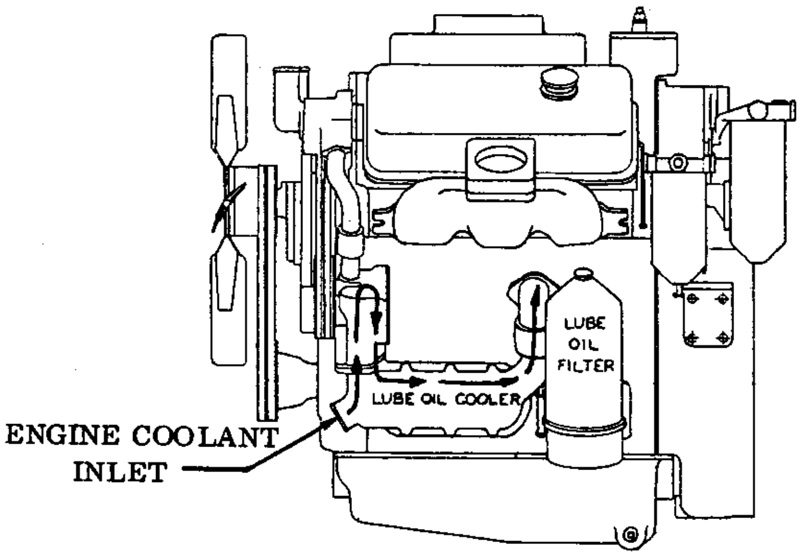
Cab heaters should preferably be located below the top tank or surge tank coolant level whenever possible to allow removal of trapped air and complete filling of the cooling system. When the heater is mounted above the coolant level a vent should be provided to remove trapped air. The engine should then be run to purge the heater of air and coolant added if necessary.



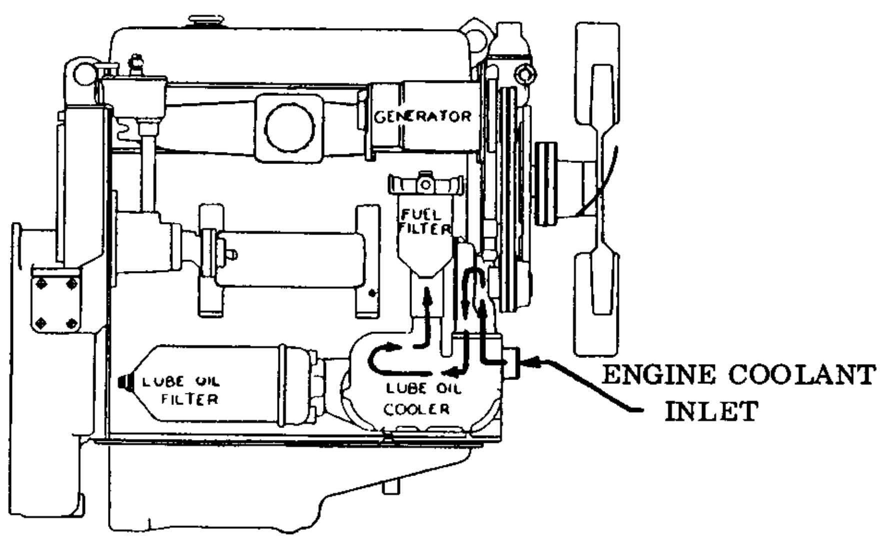
SERIES 71 VEE ENGINE
FIGURE 9



SERIES 71 INLINE
FIGURE 10



SERIES 53 VEE ENGINE
FIGURE 11



SERIES 53 INLINE
FIGURE 12

4. COOLING SYSTEM INHIBITORS

The service life of cooling system components such as gaskets, hoses, thermostats, fan belts, etc. can be controlled in a normal preventive maintenance program. However, there is an important component of the cooling system that is often neglected - the coolant itself.

There are few areas of the country in which untreated native or tap water is suitable for use in the cooling system of a diesel engine. The accompanying maps, Figure 13, page 23 serve to illustrate this point. The natural properties may create problems inside the cooling system which are not evident until damage has occurred. The basic problem is that of scale deposits within the cooling jacket which hinders adequate heat transfer and restricts coolant flow.

(a) FUNCTION

A properly maintained coolant conditioner will establish a scale and corrosion free cooling system as well as filtering suspended foreign material from the coolant. Replaceable elements are available for use with and without permanent antifreeze to provide year round protection.

The hardness of a water supply is proportional to its content of calcium and magnesium salts which are broadly classified as carbonates; the calcium salts normally occurring twice as frequently as the magnesium salts. While sulfates, chlorides and nitrates may be present the standard of hardness measurement in the United States is the calcium carbonate equivalent. Normally the measure of water hardness is expressed in grains per gallon; one grain being equal to 17.1 parts per millon (PPM) expressed as calcium carbonate. Waters containing up to 3.5 grains per gallon is considered soft and will normally create no problems of deposits. Water containing in excess of 3.5 grains per gallon should be treated to remove the scale forming minerals. This is accomplished in water conditioner by a "base" or ion-exchange process in which the calcium or magnesium ions are replaced with sodium ions which will not form harmful deposits within the engine cooling system. This process (ion-exchange) is generally considered the most effective method to soften water.

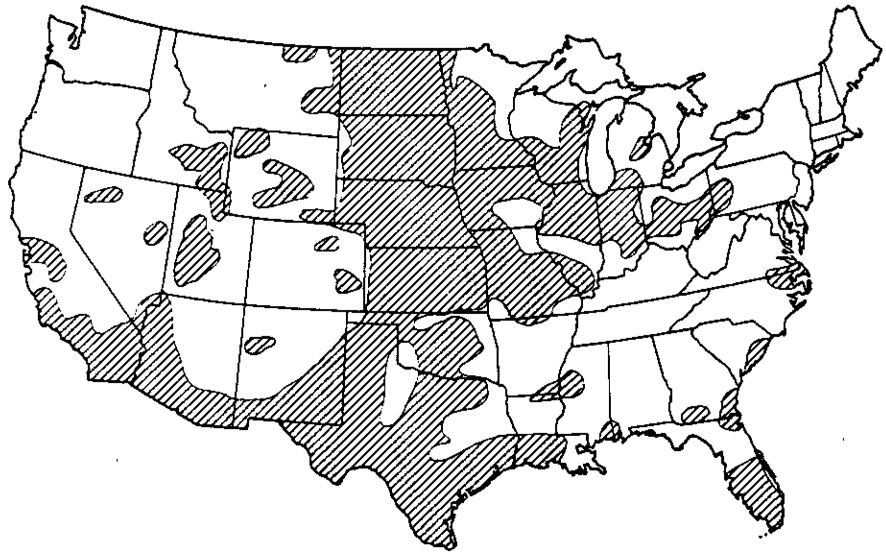
(b) INSTALLATION

Installation of the coolant conditioner should be in a position which will provide the highest possible pressure differential. Generally, the connections would be from the side of the block to the suction side of the pump. A 3/16" diameter restriction fitting should be included on the outlet of the filter when other accessories are connected into the cooling system. This is to prevent excessive by-passing of the coolant. Other possible connections are shown in the illustrations on page 24.

CORROSIVE WATER AREAS

CORROSIVE WATER AREAS

Impurity	Singly or in Combination
Cl^-	> 50 ppm
$SO_4^=$	>100 ppm
HCO_3^-	>200 ppm
(Dissolved Solids	>500)

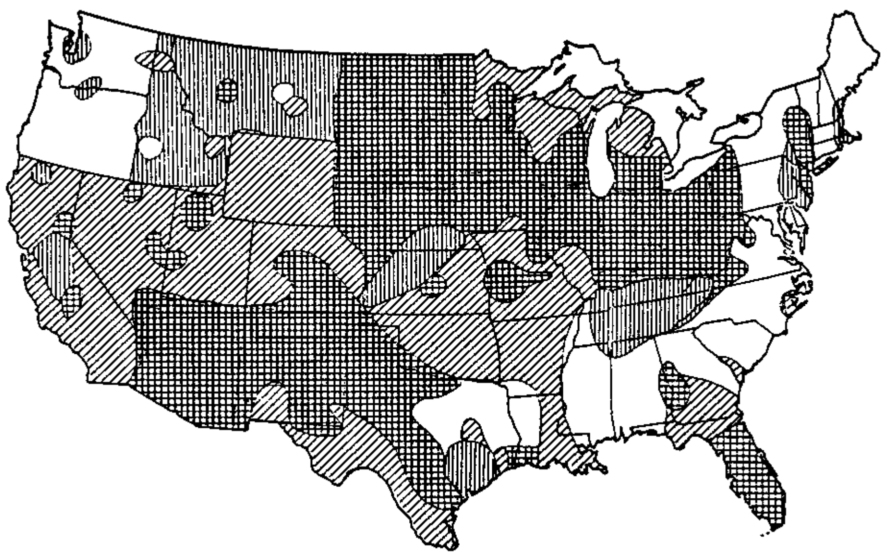


WATER HARDNESS

This map shows general water hardness areas in recognized variations. However due to the nature of the water bed sub-soil structure water hardness may vary from one source to another within a general area.

- 0-3.5 GRAINS PER GALLON*
- 3.5-7 GRAINS PER GALLON
- 7-10.5 GRAINS PER GALLON
- 10.5 AND ABOVE

Grains of hardness are expressed as calcium carbonate.
Darker areas indicate harder water.
White or lighter areas indicate softer water.
*1 Grain Is Equal To 17.1 Parts Per Million.



DISSOLVED SOLIDS

When the coolant evaporates, dissolved solids are deposited upon the walls of the cooling system, thus contributing to loss of heat transfer and possible plugging of coolant passages.

- LESS THAN 50 PPM
- 51 TO 150 PPM
- 151 TO 300 PPM
- 301 TO 500 PPM
- MORE THAN 500 PPM

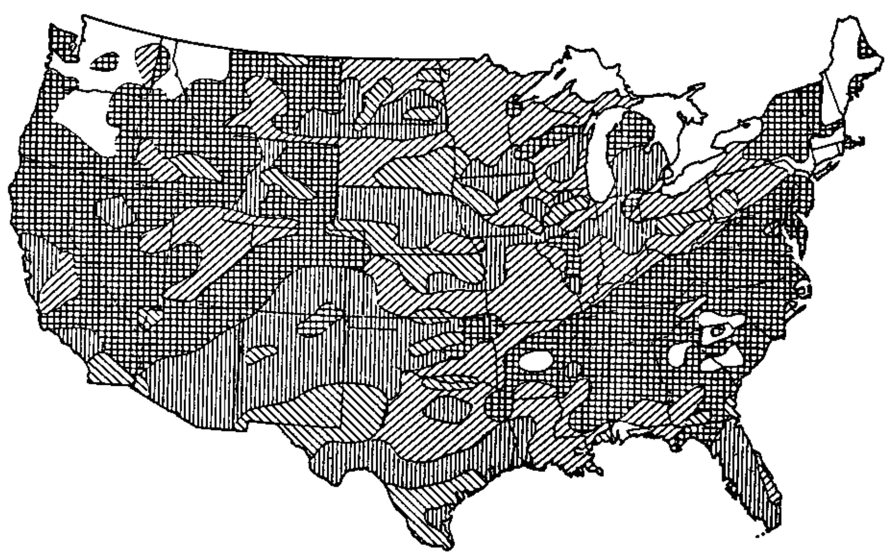
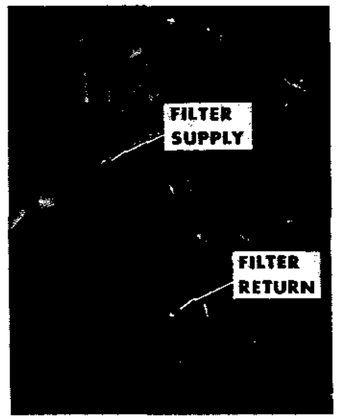


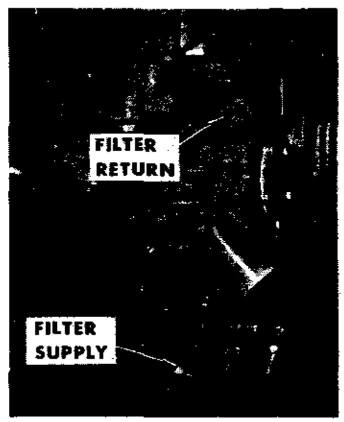
Figure 13

"53" In-Line Engine



"V-53" Engine

"71" In-Line Engine



"V-71" Engine

(c) SOLUBLE OIL

Soluble oil type corrosion inhibitors GM PT. No. 5197843 is available and can be used with aluminum block engines and wet liner cast iron engines, which includes the "53" series. Recommended concentration is 8 ounces of inhibitor for 20 quarts of water. This will provide a volume which will not exceed the allowable 1%.

For Series 71 and 110 cast iron heads and blocks, the chemical chromate type inhibitors provide better scale and corrosion protection, with very little effect on heat rejection characteristics.

Neither the soluble oil nor the chemical type inhibitor should be used in a cooling system containing a permanent type anti-freeze solution.

(d) ANTI-FREEZE

A permanent (Ethylene Glycol) type anti-freeze is recommended for Detroit Diesel Engines. Generally a corrosion inhibitor will be present in this type of solution and no additives are required. If a coolant filter is used, it should be compatible with the anti-freeze in the system.

See Figure 14, page 26 for freezing point curve, for various anti-freeze concentrations. Coolant expansion rate is shown in Figure 15, page 27.

C. RADIATOR AND FAN SYSTEM

The most commonly used engine cooling system is the radiator and fan combination and as a result this is often the most misused. Proper application of the fan and radiator consists of a combination of three items, viz. an adequate radiator core, a fan to move a given amount of air and sufficient coolant flow to maintain a uniform coolant velocity through all the tubes in the core.

1. RADIATOR RECOMMENDATIONS

The factors affecting radiator selection are available space, heat load and cost. Within these the following are necessary items which should be included to provide adequate cooling and proper operation.

(a) TOP TANK DESIGN

Factors governing the design and location of the radiator top tank are:

(1) Coolant level in tank should be such that core area is not exposed when vehicle or power unit is tilted to the maximum expected angle. See Figure 16, page 28.

(2) Coolant level in tank should be at least one inch above end of the cylinder head or water manifold when the vehicle or power unit is tilted radiator end down at the maximum expected angle. See Figure 17, page 28.

(3) Space in tank should allow for coolant expansion in the cooling system without loss of coolant through overflow. Water expands, through a 40°-200°F temperature range at the rate of approximately one-third pint per gallon of water in the system. See Figure 15, page 27.

(4) Allow an additional 30% of expansion volume described in 3 to provide additional air space for entrained gas collection.

(5) Coolant entering the top tank should be below the minimum level to avoid picking up air in the expansion area. A baffle should be incorporated in the top tank to prevent air and coolant mixing.

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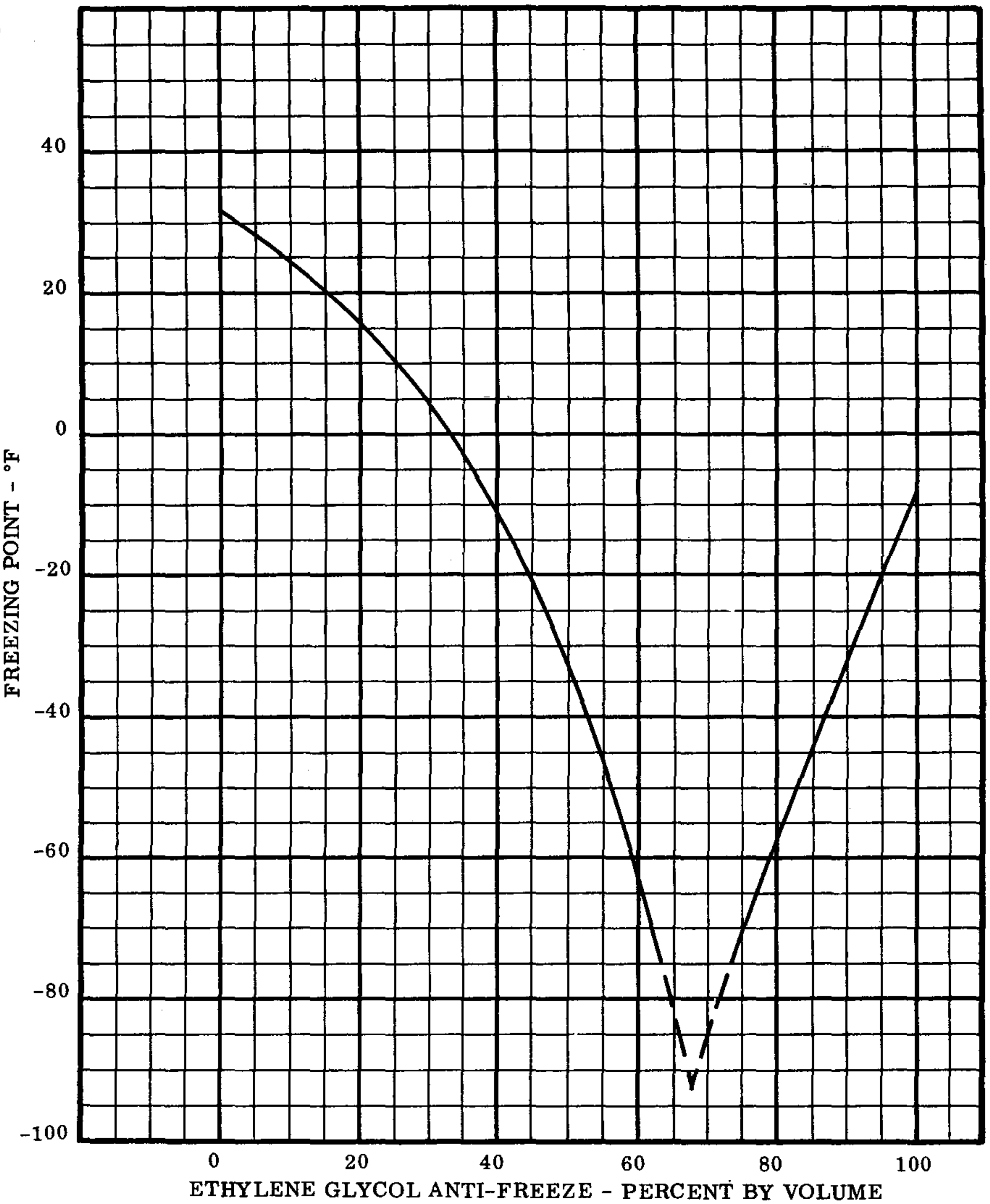
FREEZING POINTS OF AQUEOUS
ETHYLENE GLYCOL ANTI-FREEZE SOLUTIONS



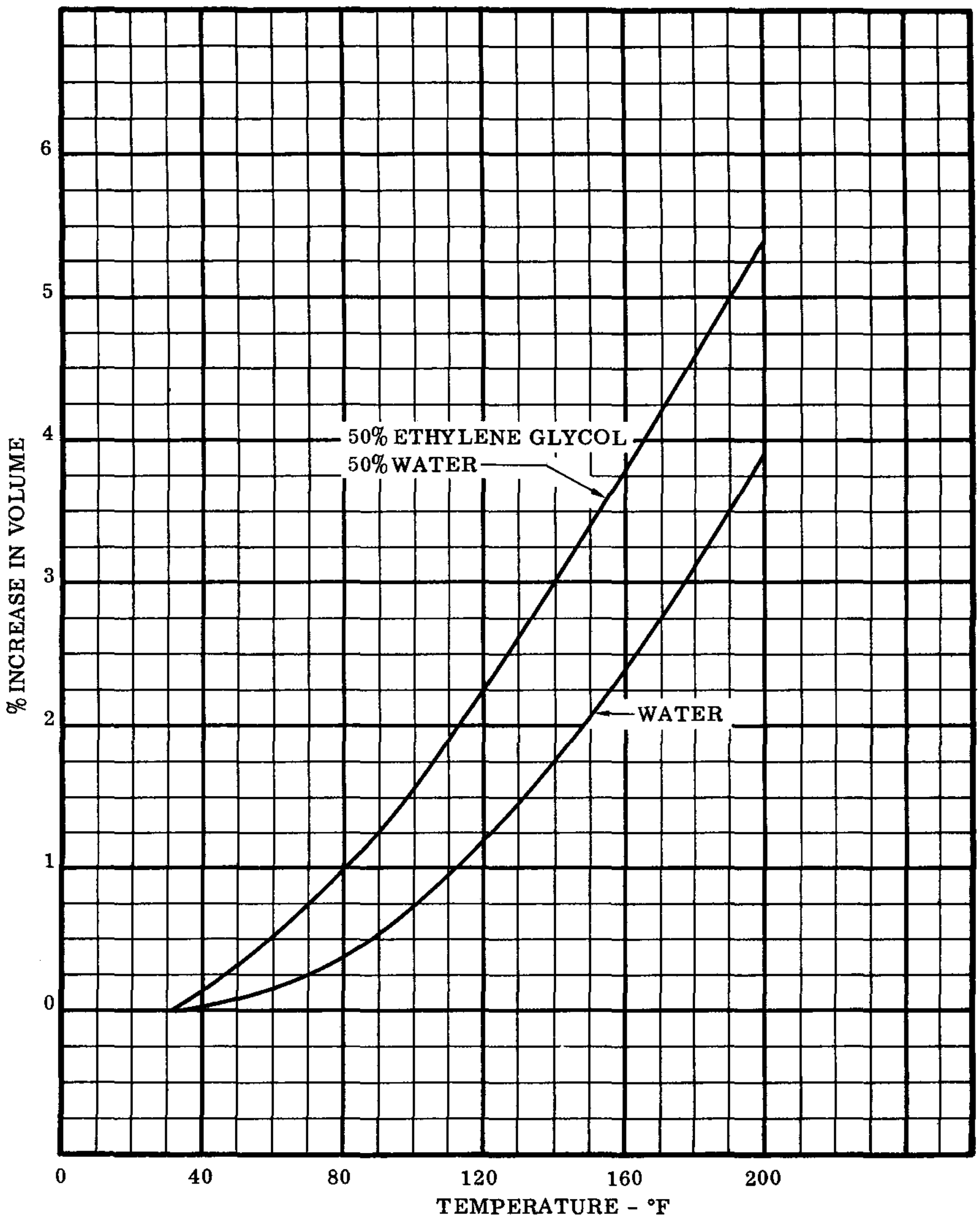
ENGINEERING-TECHNICAL DATA DEPT.

Figure 14

DETROIT DIESEL ENGINE DIVISION

GENERAL MOTORS CORPORATION

PERCENT INCREASE IN VOLUME
FOR WATER AND ANTI-FREEZE SOLUTION



ENGINEERING-TECHNICAL DATA DEPT.

Figure 15

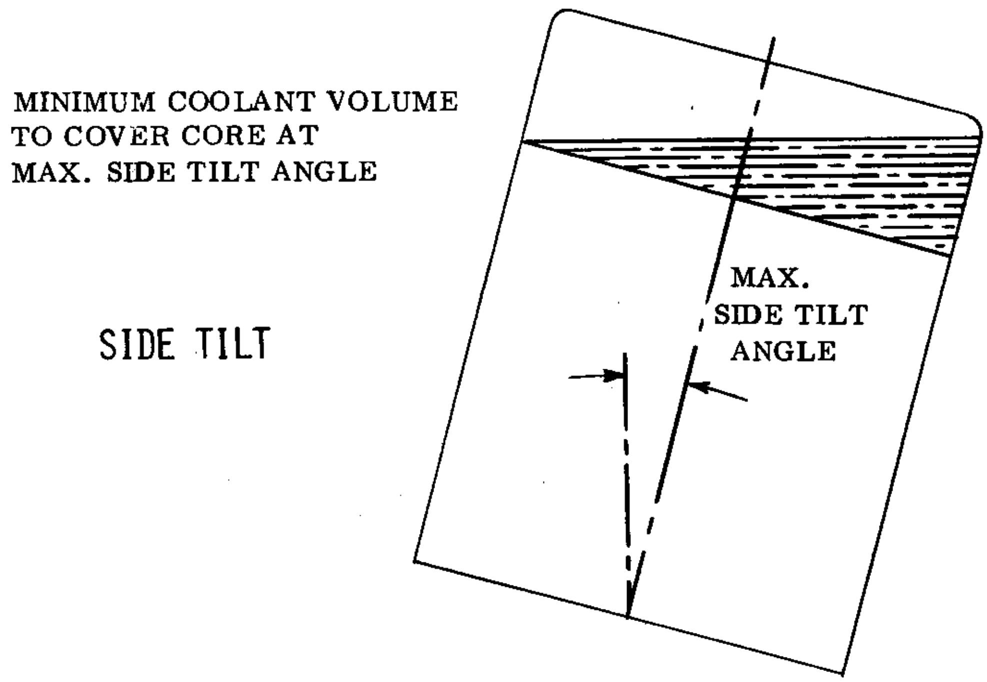
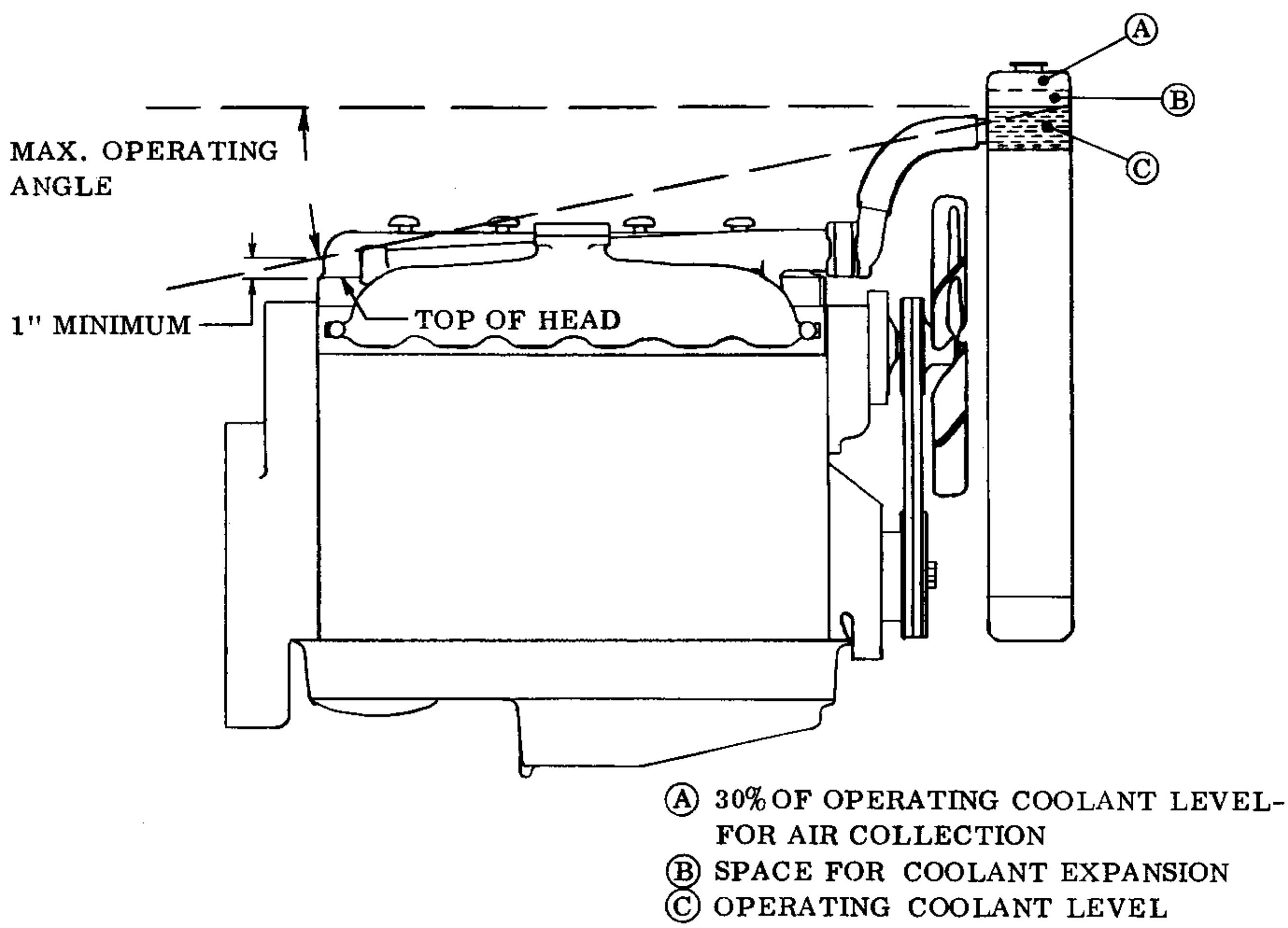


Figure 16



FORWARD TILT - RADIATOR DOWN

Figure 17

(6) The design of a baffle can elimate entrained gases by surging the outlet coolant from the engine over the header plate. Too high a coolant velocity, however, increases restriction to flow and often reduces the air separation from the coolant. All pilot models should be tested to determine the effectiveness of a baffle design by bleeding air into the system as described in Part II, page 54.

(b) <u>BOTTOM TANK DESIGN</u>

The bottom tank coolant outlet tube should by located diagonally opposite or as far from the coolant inlet as possible to provide better coolant circulation through the core and avoid direct flow from coolant inlet down through the radiator tubes and back to the suction side of the pump. See Figure 18 below.

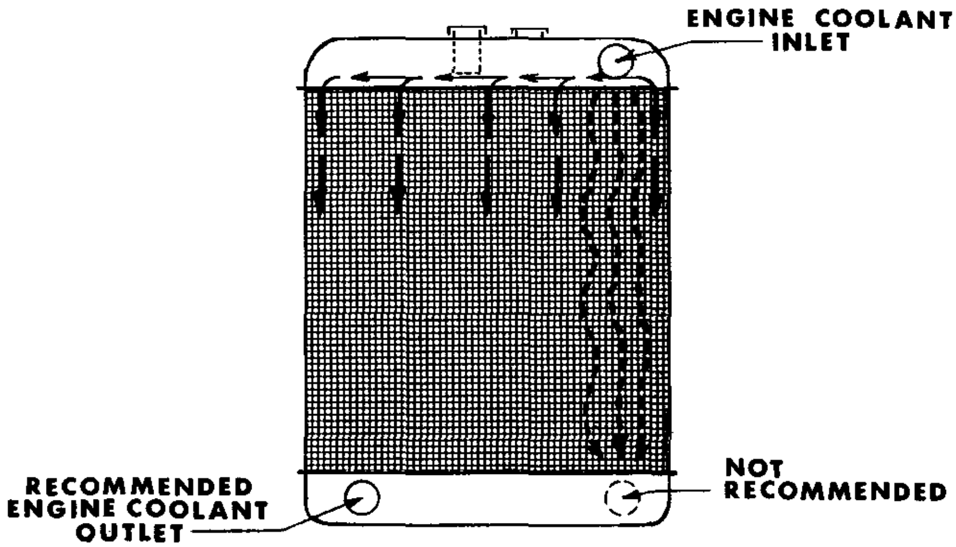


Figure 18

Unless suitable baffles are provided to distribute the coolant evenly through the core, vertically aligned, inlet and outlet connections should be avoided.

Where engine coolant inlet restriction is high a possible reduction may be gained by making the bottom tank coolant outlet a well rounded exit area on the inside as shown below in Figure 19.

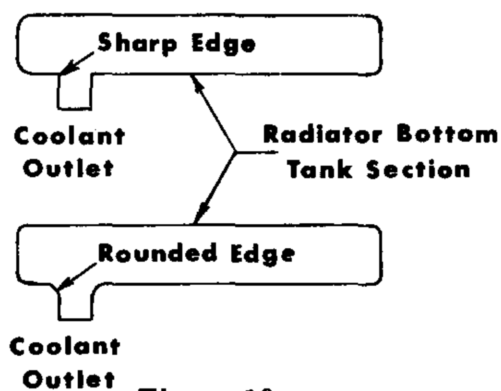


Figure 19

(c) SURGE TANKS

If space or head room does not allow the use of an adequate top tank, a surge tank of proper construction may be used, see Figure 20, page 31. When a separate surge tank is used, the top tank of the radiator or heat exchanger must be vented to the air space in the tank so that air will not collect in the radiator to displace coolant.

(d) DEAERATION

Top tank design should be such as to permit a deaeration rate of 0.1 CFM/CYL and still be capable of operating within the limits shown on page 54 of Part II. This applies to all engines.

(e) DRAWDOWN

The radiator top tank should be designed so that it has an excess coolant volume over and above the minimum operating level to meet the requirements in Part II. See Figure 38, page 59 for drawdown rating.

(f) CORE CAPACITY

The cooling ability of the radiator is measured by the temperature differential, of the coolant entering the core and the ambient air temperature. Permissible temperature differentials which will provide a safety factor against cooling system deterioration are shown in Chart II, page 6.

The radiator and system restriction must be low enough to require no more than 6% of the gross horsepower for the cooling fan.

From the stand point of coolant flow, core capacity is often described as "free flow". This term is used to indicate gravity flow rate through the core and should equal or exceed the designed flow rate of the engine coolant pump. This will assure minimum pump inlet restriction relative to core design. This is an important factor even though the point of maximum restriction may be in the coolant outlet in the bottom tank rather than in the core. Since core area is usually 2 to 3 times that of the outlet area. Large diameter lines from radiator outlet to the pump inlet are imperative. See Chart III, page 12 for minimum inlet and outlet diameter of coolant connections.

(g) CORE THICKNESS

Increasing core thickness (addition of tubes) imposes added restriction to air flow resulting in increased fan parasitic load. Increasing the core density by adding tubes and/or fins not only increases fan load but also results in a core which rapidly loses its efficiency from plugging of the air side. Core

SUGGESTED SURGE TANK DESIGN

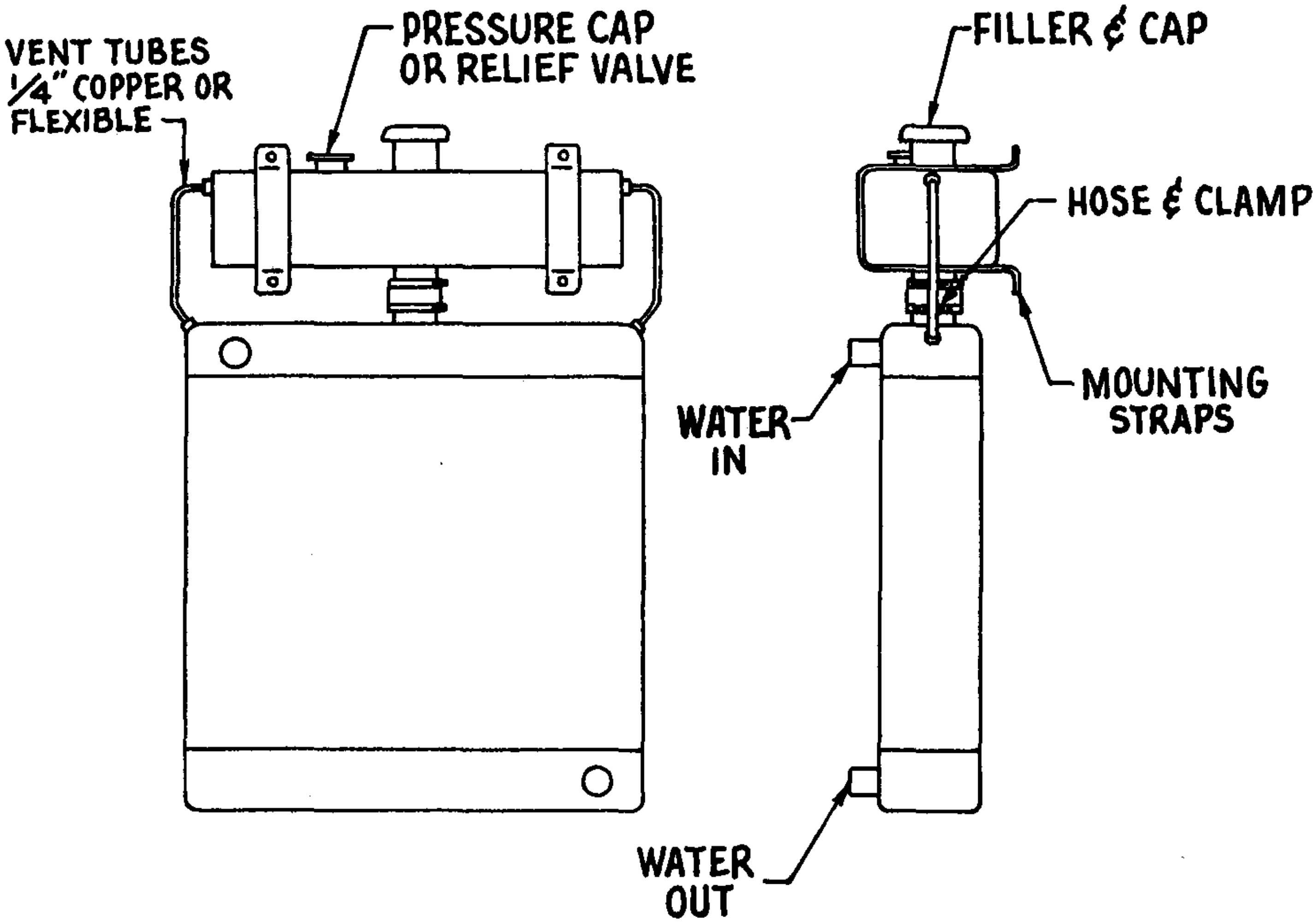


Figure 20

thickness should be held to a minimum with a fin spacing of 6 to 8 1/2 fins per inch. The air side of the system is critical and a reduction in the required air flow will much more seriously effect cooling than a similar reduction in coolant flow.

(h) FINS PER INCH

The suggested fin spacing for various applications is shown below:

Fins Per Inch	Application	Remarks
6	Agricultural	-
6 - 8	Earthmoving	6 preferred
6 - 8 1/2	General	-
10 1/2 max.	Automotive only	Engineering approval required

As more fins per inch are added, the area for heat dissipation is increased while air flow is reduced due to the added restriction. The net effect may be little or no improvement in cooling on a new installation and actually poorer cooling as the unit gets older and the core becomes plugged with air borne debris. See Figure 22, page 37 for a typical fan performance as effected by increased restriction.

(i) CORE DESIGN

Tube and fin design is superior to cellular or honeycomb type because of the lower restriction to both air and coolant flow and it is easier to clean the wide and straight coolant and air passages. It is also more rugged and thus more suitable to the operating environment of diesel engines.

(j) CORE FLOW

The downflow type radiator is customarily used and is recommended. If head room is limited, crossflow radiators may be used but it involves problems of deaeration, thermal stratification and adequate core tube coverage. Freezing damage frequently occurs as the tubes and portions of the headers may hold coolant even after the engine is thought to be thoroughly drained.

(k) SHAPE

A square core equal to or slightly larger than the fan diameter is ideal in order to provide optimum fan coverage. If other considerations determine that a narrower radiator must be used then some sacrifice in cooling ability will result.

(l) PRESSURE SYSTEMS AND CAPS

The radiator selection should be designed for proper cooling with a non-pressurized system for all operating conditions. Pressure caps are then to be used for protection against boiling at above base line elevations. See Figure 21, page 34 for proper pressure cap recommendations. A 12 to 15 pound cap may be required at high altitudes to provide a margin above boiling. Their use, however, must have approval of the radiator manufacturer, so as to stay within pressure limitations of the radiator construction and must be compatible with coolant hose burst strengths.

If a pressure cap is used it should, whenever possible, be separate from the filler cap and located in the radiator top tank. Care should be taken that it is placed in an area high above the quiescent coolant level to avoid loss of coolant and to keep it from being splashed and clogged with dirt.

(m) SHUTTERS, SHUTTERSTATS AND THERMATIC FANS

Outdoor operation in cold climates requires hoods and either manual or thermostat controlled shutters to assure a minimum coolant temperature of 160°F. When shutters are required the louvers should be mounted horizontally to prevent hinges from becoming blocked with dirt or ice. Rigidly constructed shutter frame assures long life and dependable operation.

Shutterstats should be set 10° above the engine thermostat opening temperature when mounted in the coolant outlet manifold or radiator top tank. Care must be taken to insure that the thermal elements are fully submerged in coolant at all times. With shutterstats mounted in the radiator bottom tank or sensing coolant temperature leaving the radiator setting should be equal to that of the engine thermostat opening temperature. This is required to allow for the temperature drop across the radiator.

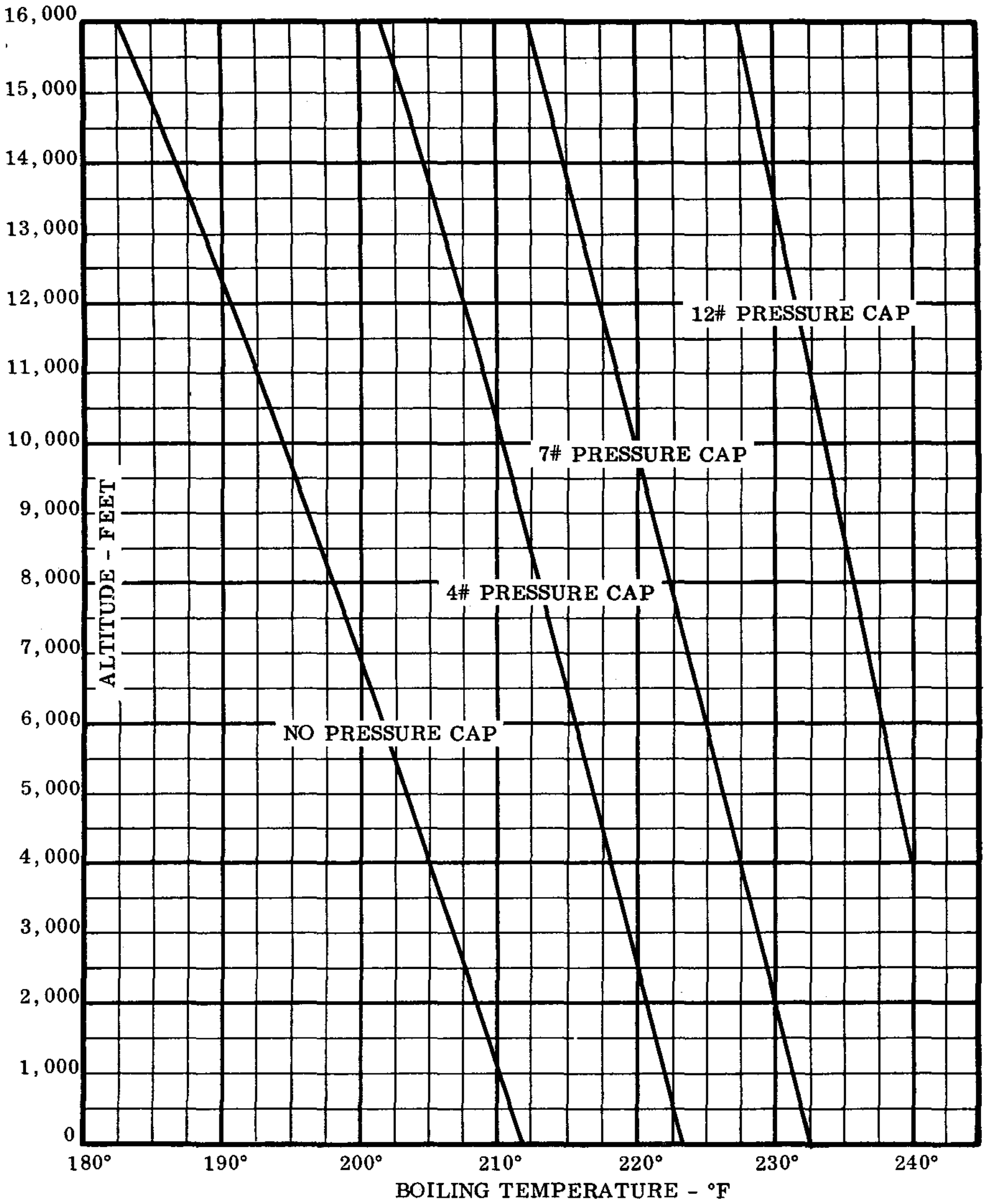
A thermostatically controlled fan may also be installed in conjunction with, or instead of, the shutters to improve temperature control and/or fuel economy. When thermal controlled fans and shutters are used together, set the shutter control 10° higher than the engine thermostat opening temperature and the fan 10° higher than the shutters. Example: Thermostat opens 170°, shutters full open 180° and fan engaged at 190° for top tank mounted stats.

(n) COOLER IN FRONT/REAR OF RADIATOR

When an oil to air cooler or auxiliary core for accessory cooling is added in front of or behind the radiator, an access panel should be provided to clean the area between the cores. Frequent inspections are necessary to keep this area free from material which will restrict air flow.

DETROIT DIESEL ENGINE DIVISION
GENERAL MOTORS CORPORATION

WATER BOILING TEMPERATURES
AT ALTITUDES ABOVE SEA LEVEL



ENGINEERING-TECHNICAL DATA DEPT.

Figure 21

(o) FILLER PIPES

To assure a positive expansion chamber, the filler neck should be extended down into the top tank to the desired coolant level which will provide an easy mark for filling. The pressure relief valve must be installed in the top of the tank, preferably at the tank center and not in the filler cap. Filler pipes should be located near the tank center to insure a more complete filling if the vehicle or power unit is tilted.

In power units and generator sets which are not exposed to rough handling and are more frequently attended, the filler tube may have the pressure relief valve integral with the cap. If such is the case, the coolant inlet should be separated by a baffle to prevent coolant being forced out of the filler.

2. FAN RECOMMENDATIONS

Proper fan selection is necessary to insure maximum system efficiency. Parastic fan losses should be kept to a minimum and should not exceed 6% of basic engine horsepower. The fan can be matched to the radiator by predetermining the radiator air flow required and ascertaining the total static air pressure which the fan must overcome.

(a) FAN SELECTION

The air flow required to produce the desired amount of cooling in terms of air to water differential is determined by the heat to be dissipated and the coolant flow available. When selecting a fan, a good design point for air flow is characterized by the static pressure point "A" in Figure 22, page 37 which should be in the range of 1 1/2 to 2 inches of water. A small change in static pressure from this point does not mean a large change in air flow. A point which would fall in the shaded portion or stall area of the fan curve should be avoided because unstable flow characteristics occur as evidenced by the wide range of air flow for small changes in static pressure.

(b) FAN PERFORMANCE

Figure 22 on page 37 shows fan performance in the form of flow (CFM), static head (inches of water, gage) and horsepower. Most performance curves are drawn on the basis of theoretical output which is seldom achieved. An additional curve, probable air flow, is often shown also. This usually is 85% of the theoretical and refers to air flow (CFM) only. The theoretical output can be approached with a well formed, close fitting shroud. The fan must be properly located in the shroud and with no more than 1/16" tip clearance. Frequently a close fitting shroud is not practical to use and a compromise to a loose fitting box shroud must be made. Performance with the latter will be 85% of that of a tight fitting shroud.

The above mentioned fan curve also shows performance at various fan speeds. When information at other speeds is required it may be calculated using the standard fan performance laws where:

CFM varies directly with RPM,
Static head varies with RPM^2
Horsepower varies with RPM^3

Other factors influencing actual fan performance are air temperature, atmospheric pressure and humidity. While CFM is not greatly affected by air density it must be remembered that it is the air "mass" flowing across the core that does the cooling and when planning a system for very high altitude operation this factor must be taken into consideration.

DETROIT DIESEL ENGINE DIVISION

GENERAL MOTORS CORPORATION

RADIATOR - FAN MATCH

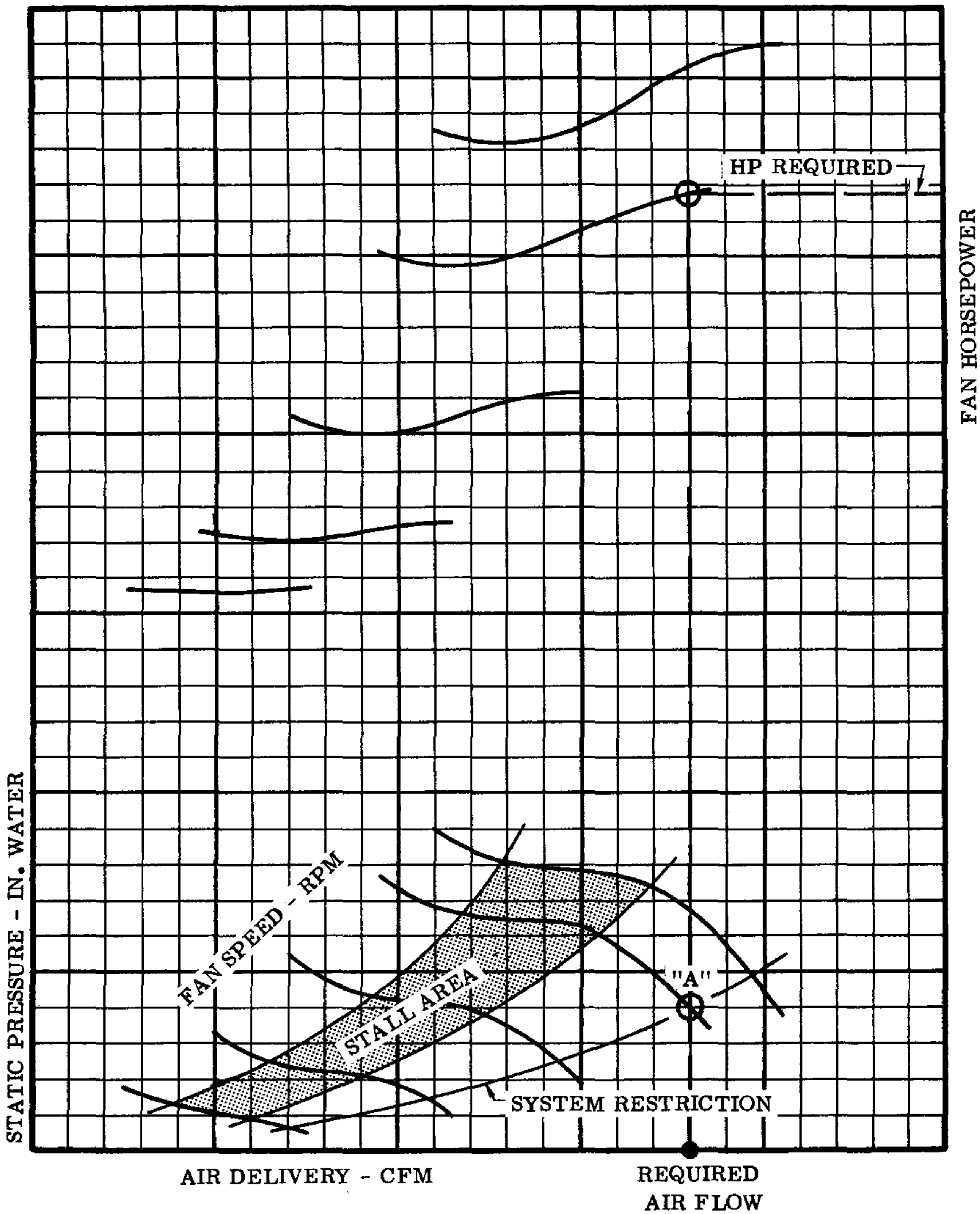


FIGURE 22

ENGINEERING-TECHNICAL DATA DEPT.

(c) FAN SIZE

A study of various fan sizes will indicate that for the most efficient installations the largest diameter fan turning at the lowest speed to deliver the desired amount of air (CFM) will prove to be the most economical to operate. As radiators by necessity become smaller in area they must be made thicker with more fins and/or tubes presenting a denser core. This presents increased restriction to air flow and fans must be either driven faster or blade angles must be increased to produce a greater static pressure capacity. Fan blade angles can be increased to 35° maximum, beyond this point only an increase in fan speed will produce additional air flow.

With horsepower varying with the cube of the speed, it becomes apparent that radiators with areas as large as possible and with fans turning as slow as possible are most suitable. Fan diameters greater than the radiator width or height are of no advantage.

(d) FAN POSITION

Fan position relative to the radiator depends on the fan diameter and the radiator frontal area. When the fan swept area is approximately the same as the radiator frontal area, locate the front of the fan 2 to 4 inches from the core. As the swept area becomes less than the radiator frontal area the fan should be moved further away from the core. This allows the air to spread over the full core area which will not occur if the fan is too close to the radiator. It is also necessary to use a shroud to prevent air from being dispersed around the core and recirculated.

(e) BLOWER VS SUCTION FANS

The application will generally dictate the type of fan to be used, ie. vehicles normally use a suction fan and stationary power units frequently use blower fans. Blower fans are generally more efficient in terms of power expended for a given mass flow since they will always operate with lower temperature air as compared to a suction fan. The air entering the suction fan is heated as it passes through the radiator where as a blower fan even though engine mounted will receive air nearly at ambient temperatures.

The use of blower fans do however, require more attention to shroud configuration to realize the fans inherent higher efficiency. The air leaving the blower fan will diverge at a predetermined rate depending on velocity and temperature. The fan must be spaced from the core to insure that the complete core is covered by the fan air flow, requiring a well designed shroud.

3. FAN SHROUDS

The use of a shroud is very important in achieving an efficient fan installation. A good shroud not only will increase air flow through more efficient usage of core area but will also eliminate recirculation of air.

The shroud attachment to the core must be air tight to prevent recirculation of the air around the radiator core. Radiator guards, grills and other obstructions to flow may also cause air to recirculate. Therefore, the pilot model should be tested to determine if recirculation exists and, if so, it should be corrected as necessary.

(a) TYPES OF SHROUDS

Basically there are three types of shrouds, the well rounded entrance venturi ring, a ring type and box type, see Figures 23, 24 and 25 respectively. The maximum air delivery is obtained with a close fitting, venturi type shroud. This should be used with a fixed fan position and an idler for belt adjustment or an adjustable shroud.

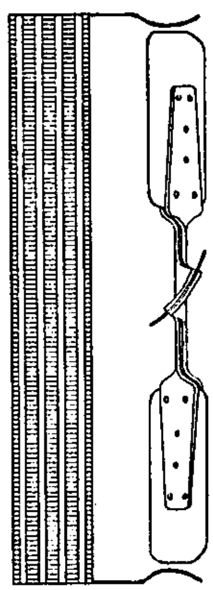


Figure 23

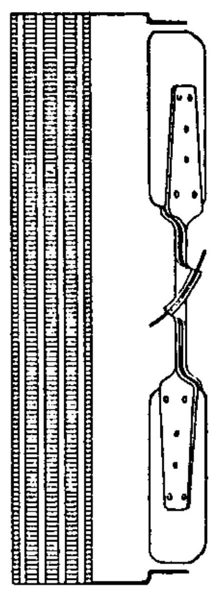


Figure 24

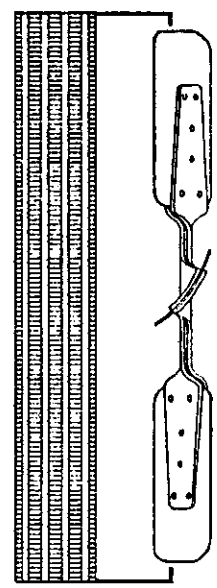
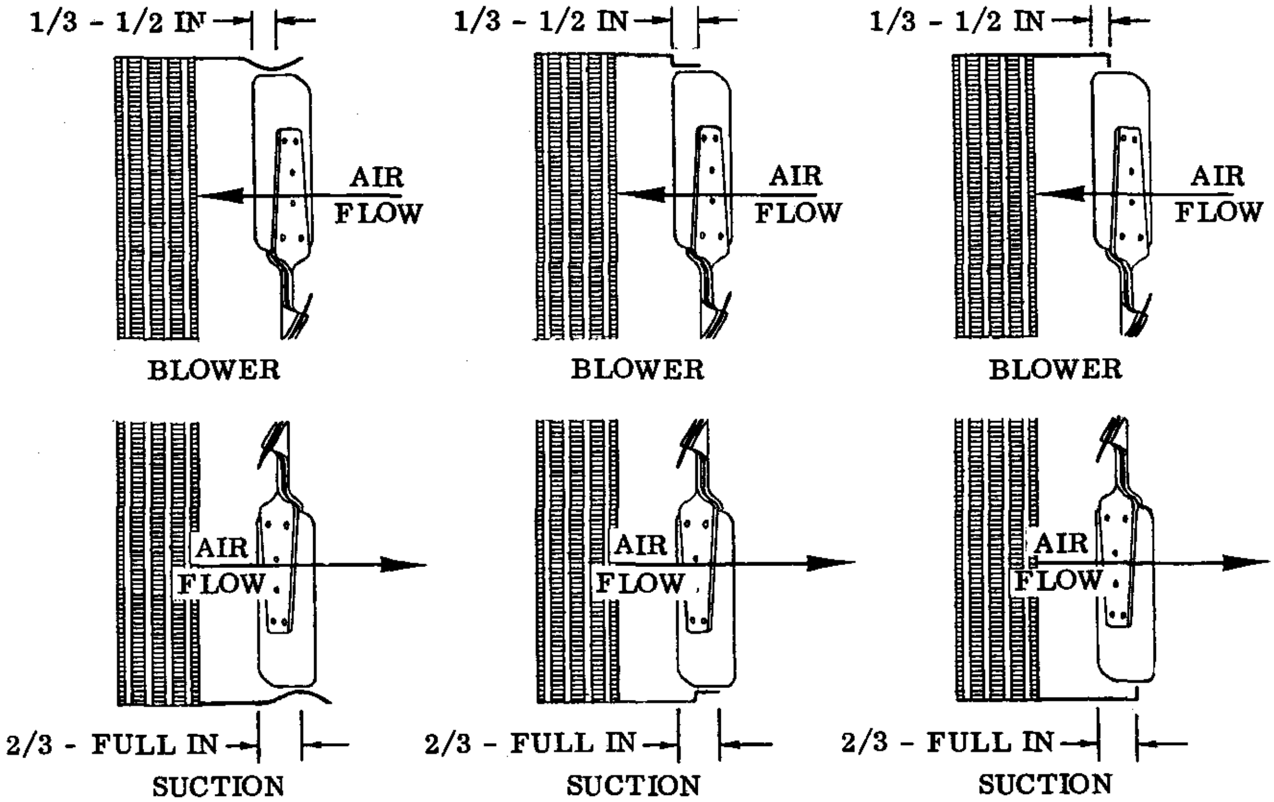


Figure 25

Ring or box type shrouds are most commonly used, but are not as efficient as the venturi type. The tip clearance reduces air flow significantly and should be kept down to a minimum of approximately 1/2 inch or less. Additional clearance must also be provided for fan belt adjustment.

(b) FAN-SHROUD POSITION

Figures 26, 27 and 28 show the generally recommended position of the suction or blower type fans with respect to the shroud. A suction fan extends 2/3 to full projected width into the shroud and a blower fan 1/3 to 1/2 the projected width into the shroud. These positions may vary depending upon air flow restriction.



Venturi Type
Figure 26

Ring Type
Figure 27

Box Type
Figure 28

4. IMPROVED DEAERATION SYSTEM

This system was developed for vehicle applications to provide better air handling ability and closer temperature control than the conventional system.

(a) CONVENTIONAL VS. IMPROVED DEAERATION SYSTEM

Figures 29 and 30 on page 42 show the closed and open thermostat coolant flow through the conventional cooling system. During closed thermostat operation, deaeration is necessary and is accomplished with bleed holes in the thermostat and thermostat housing. This can cause overcooling in low ambient temperature since there is some flow through the radiator at all times.

Figures 31 and 32 on page 43 show the closed and open thermostat coolant flow through the improved cooling system. This system permits coolant flow through the radiator only when the thermostats are open. The closed thermostat deaeration is accomplished through the deaeration line from the engine side of thermostat housing or by-pass side to the radiator top tank. The coolant is then returned to the engine through the supply line without passing through the radiator core and being cooled.

The improved system differs from the conventional system as follows:

Engine changes:

(1) Eliminates thermostat bleed hole.

(2) Eliminates holes in thermostat housings which vent coolant and air from the engine to the radiator.

(3) Suitable engine connections for piping to and from separate surge tank or integral tank must be provided.

On systems with surge tanks, three fittings are required (See Figure 34):

(1) Low or bottom connection for a deaerating line from engine thermostat housing.

(2) Bottom connection for a line to the suction side on the engine coolant pump.

(3) Top connection for a line to the radiator top tank.

On systems with integral top tank (See Figure 33):

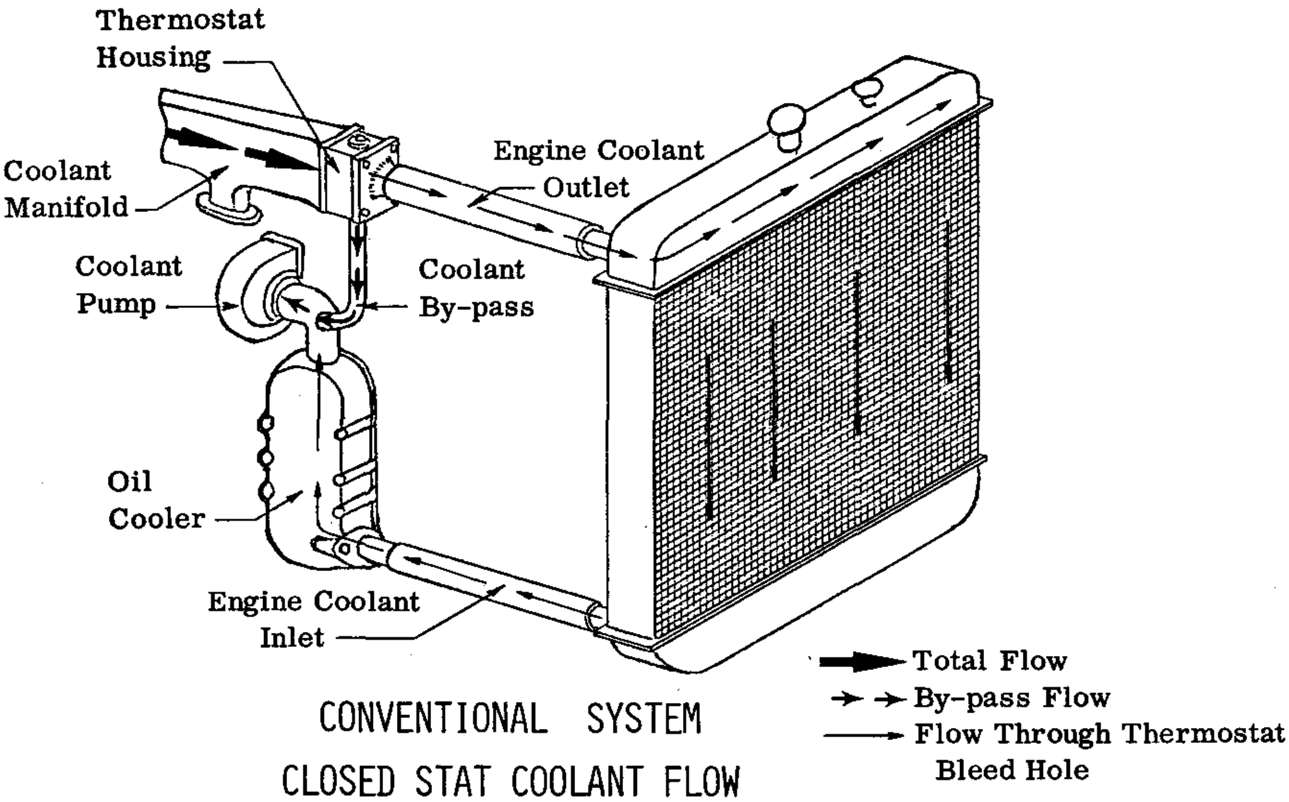
(1) Low or bottom connection for a deaerating line from thermostat housing.

(2) Bottom connection for a line to the suction side of the coolant pump.

(3) A standpipe from baffle separating top tank and radiator to above top tank coolant level.

The use of anti-freeze is mandatory with the improved deaeration system. Even if shutters are used, the lack of radiator flow with closed thermostats will allow the radiator to freeze.

The following items cover in detail the components used with the improved deaeration system:

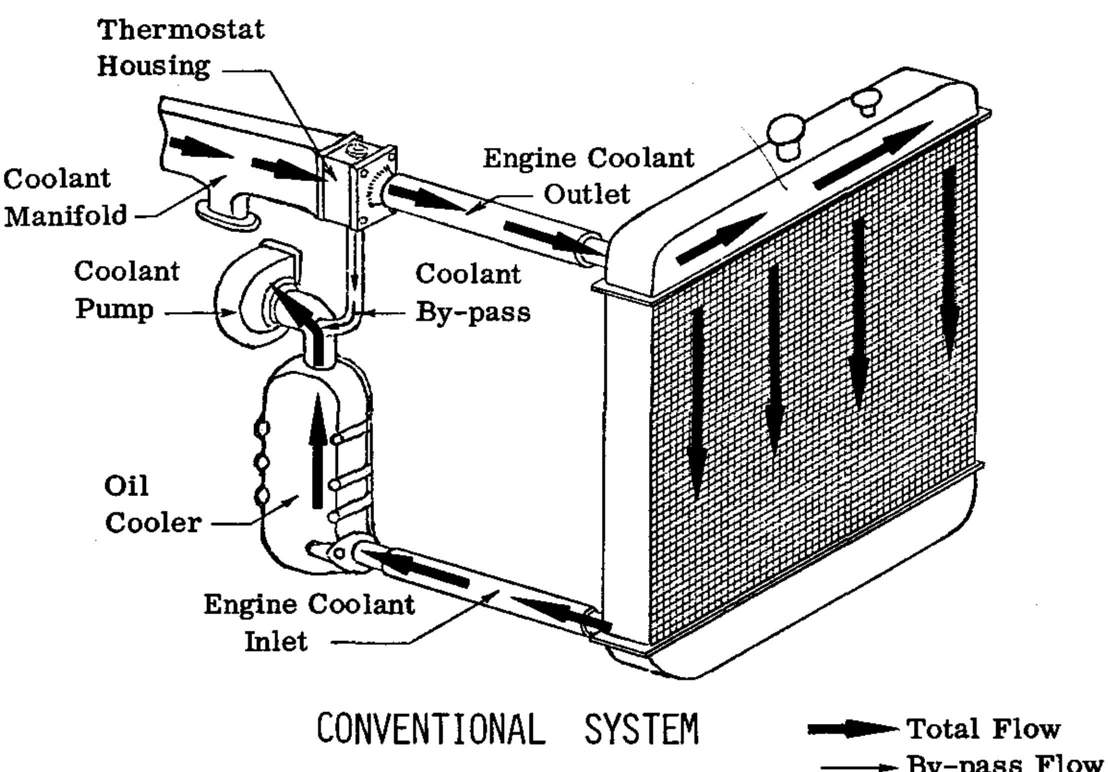


CONVENTIONAL SYSTEM

CLOSED STAT COOLANT FLOW

(NON-BLOCKING TYPE STAT)

Figure 29



CONVENTIONAL SYSTEM

OPEN STAT COOLANT FLOW

(NON-BLOCKING TYPE STAT)

Figure 30

(1) TYPICAL HOSE SIZES

Figures 31 and 32 on page 43 show schematics of an inline engine with connections for the improved system. The typical hose sizes for this type of system are as follows:

CHART IV

TYPICAL IMPROVED COOLING SYSTEM HOSE SIZES					
	Deaeration Lines				
Engine	Engine to Tank	Tank to Radiator		Fill Line	Remarks
		Top Tank	Surge Tank		
Inline 71	5/8" I.D.	Intergal Standpipe	3/8" I.D.	1" I.D.	1/4" I.D. restricted fitting in top of thermostat housing
V-71	3/8" I.D.	"	3/8" I.D.	1" I.D.	-
Inline 53	1/4" I.D.	"	3/8" I.D.	1" I.D.	-
V-53	1/4" I.D.	"	3/8" I.D.	1" I.D.	-

The hose sizes shown above are typical sizes but will not necessarily be adequate for all cooling systems. Any new cooling system must be checked by the methods specified in Part II to determine if the system is satisfactory. In addition to checking for balanced flow through the lines, it may be necessary to make the lines larger or smaller depending upon the conditions encountered.

(2) TOP AND SURGE TANK DETAILS

The improved system radiator top tank has the same general requirements specified earlier with respect to air collection volume, expansion volume, fill and pressure cap. The added requirements are as follows:

(a) STANDPIPE

A standpipe is required to vent the core during filling for open thermostat deaeration. The coolant level above the baffle including the expansion volume should not be capable of reaching the top of the standpipe. The filler neck should be lowered into the tank to produce the desired full coolant level. A 1/8" hole should be provided in the filler neck for safety and to prevent water loss when opening the filler cap when the coolant is hot.

(b) MINIMUM COOLANT LEVEL

This coolant level should be determined by test to allow coolant coverage of the deaeration line and supply line at all times. This should include operation under tilt conditions also. The minimum level is necessary to prevent aeration of coolant returning to the engine. The minimum coolant level may be reduced if vortex baffles are added.

(c) DEAERATION LINE

Typical line sizes are shown in Chart IV, on page 45. On V-71 engines deaeration lines are always to be installed below the minimum top tank or surge tank coolant level. This is to prevent the possibility of air being taken from the top tank when the thermostats are open at which time the deaeration line may sense pump suction.

On inline engines the deaeration line may be above or below the top tank coolant level since this line is always under a positive pressure with open or closed thermostats. It is necessary in this case to install a 1/4 inch diameter orifice in the deaeration line to prevent over filling of the top tank.

(d) SUPPLY LINE

The supply line should be connected to the lowest point of the tank, and away from the deaeration line to prevent picking up aerated coolant. Chart IV, on page 45 shows the typical line sizes.

When the supply line is connected to the bottom of the integral or remote mounted surge tank vortex baffles should be used. They may not be necessary if lines connect to the side of the tank and are covered by coolant under all conditions, but may be added to reduce the minimum coolant level required.

(e) TOP TANK BAFFLE

The radiator header tank is separated from the deaeration chamber by a solid baffle. The only direct connection is through a stand pipe used for deaeration. The baffle should be a minimum of one inch above the radiator core to provide coverage of all the core tubes.

(f) ENGINE COOLANT OUTLET

The engine coolant outlet line connecting into the radiator top tank must have the bottom of the line below the top tank baffle to prevent trapping air in the hose and well when filling. Depending on the length of hose in the installation enough air may be trapped to cause an aeration problem. If the line can not be placed low, then a stand pipe should be placed in the well area to eliminate the air when filling the cooling system.

3. SHUTTERS AND IMPROVED SYSTEM

Shutters may be eliminated with the improved system since there will be no radiator flow (through bleed holes) until the thermostats open. Whereas flow does occur through the radiator in the conventional system and shutters are required to keep the flow of air through the radiator from cooling the circulating coolant.

When using the improved cooling system on applications where there is a considerable amount of low speed, low load operation or idling, it is recommended that shutters be used to provide adequate protection from engine operation at low jacket coolant temperatures due to cooling from radiation.

II. TESTING THE COOLING SYSTEM

A. COOLING SYSTEM TESTS

Cooling systems should be tested for the following three things:

1. Satisfactory Cooling Index.
2. Deaeration or separation of entrained air.
3. Effect of low coolant level.

It is recommended these tests be run on any new application to make sure the cooling system will do the job it is intended to do or whenever problems of overheating occur in the field that cannot be solved without the factual data these tests will provide.

1. SATISFACTORY COOLING INDEX

This test will factually tell the cooling index of a given cooling system and determine if the system is overcooled, undercooled or performing satisfactorily.

(a) STATIONARY UNIT TEST

For stationary engines it is necessary to impose the greatest load possible to the PTO (such as high air pressure or hydraulic pressure in the case of compressors and mud pumps). Generator sets should be operated at maximum KW delivery.

(b) VEHICLE UNIT TEST

Vehicles can be tested by one of three methods, a chassis dynamometer, a towing dynamometer, or running up a steep hill with the heaviest practical load.

(1) CHASSIS DYNAMOMETER METHOD

The following procedure will permit the calculation of the cooling index with reasonable accuracy without prolonged running of the vehicle on the dynamometer. It consists of several short full load runs of equal time duration of one minute, beginning each run at different top tank temperatures. These runs are to be made at full load at or near governed speed and at 70% of the governed speed or as near peak torque as possible. This is to insure that there is adequate cooling in both areas of operation.

(a) The vehicle should first be prepared in the manner described in Part C, under INSTRUMENTATION AND PROCEDURE.

(b) If the vehicle is equipped with air operated shutters, replace the shutter stat control with a manually operated quick acting valve so that the shutters may be rapidly opened and closed. If modulating shutters are used, disconnect the shutter stat control and operate the shutters manually.

If there are no shutters, use cardboard, Masonite, etc., to quickly and effectively cover and uncover the radiator core. The basis of these tests is the temperature change per unit of time. With blocked open thermostats installed, it is necessary to control the top tank temperature prior to making each timed full load run.

The ambient temperature must not change during the one minute run. However, the ambient temperature does not have to be held to the same temperature as a previous run. Where winds or drafts are encountered provisions should be made to shield the unit.

The air temperature entering the radiator must not vary due to recirculation. If the physical arrangement of the vehicle on the dynamometer and the dynamometer in the building promotes air recirculation to the radiator some means of ducting outside air or otherwise preventing air recirculation must be found.

(c) No data is to be taken until sufficient running time either on the road or at part load on the chassis dynamometer is completed to insure warm-up of the engine and the complete drive train.

(d) With shutters closed run engine at or approximately 100 RPM below full load governed speed. Do this by applying sufficient load at the dynamometer to bring the engine down off the governor. This will assure full load operation.

(e) When top tank temperature reaches 170°F, note time, temperature and run for exactly one minute with shutter open. Write down the change in top tank temperature after the one minute run. (Example: If top tank temperature was 170° initially and rose after one minute's operation to 176° then there is a rise of 6°.) NOTE: Top tank temperature may rise or fall during the one minute run.

(f) Repeat Part (e) beginning at a 10°F to 15°F higher top tank temperature. Run for exactly one minute as before and note the top tank temperature change.

(g) Repeat Part (e) again starting with a still different temperature. Begin at a higher temperature (above 185°F) if Parts (e) and (f) showed a temperature rise. Begin at a lower temperature (below 170°F) if Parts (e) and (f) showed a temperature drop.

The purpose of the above temperature checks is to provide three changes in temperature each time starting with a different top tank temperature so that three points may be plotted as shown on the example in Figure 35 page 51. It is extremely important that all three runs be made at the same exact time interval. Plotting the temperature changes as shown on Figure 35 page 51, will define the cooling index of the vehicle.

The following example shows how the cooling index of a vehicle can be determined on a chassis dynamometer using the method described above. The required three runs were made as follows:

	Run No. 1 (Part e)	Run No. 2 (Part f)	Run No. 3 (Part g)
Top Tank Temperature:	170	180	190
Ambient Air:	90	90	90
Air/Water Differential:	80	90	100
Temperature change after each run:	+6	+2	-2

Plot the amount of change in top tank temperature for each run, on the graph as shown on Figure 35, page 51. The air/water differential is figured on the basis of the starting top tank temperature for each run.

Draw a straight line between the points. The point at which the line crosses the "Zero change line" is the air/water differential of the system. The curve determined should be rechecked by selecting two additional points different from those already recorded. If the points are not within a couple of degrees of the established curve the entire procedure should be repeated.

The above procedure must then be run for the 70% of governed speed or peak torque.

(2) TOWING DYNAMOMETER METHOD

Vehicles should be operated at a low MPH (No faster than 15 MPH) to minimize the effect of ram air and should, if tested on the level with vagrant winds, be run in two directions to minimize the effect of the wind.

On-highway vehicles should be run at full load, governed speed and also at 70% of governed speed, still at full load. Cooling at the reduced RPM will usually be 4° to 6° poorer but will indicate the degree of margin in the system since the vehicle can always be operated in the next lower gear to obtain top governed RPM and slightly less than full load which will improve the cooling.

Off-highway vehicles should be run at full load and 70% torque converter efficiency in whatever gear the 15 MPH speed limit will not be exceeded. The vehicle should first be prepared in the manner described in Part C under INSTRUMENTATION AND PROCEDURE.

After the engine is started and warmed up, it should be operated at full load and maximum governed speed. If shutters are used, they may be closed to assist rapid warm-up but must be opened and kept open once the engine is up to temperature. Readings may be taken as soon as temperatures stabilize. It is suggested that several set of readings at suitable time increments be taken to make certain temperatures are stable. A sample of DE-2804 "Cooling Test Data Sheet" is shown on page 57 which suggests a sequence for taking the data and other information.

(3) HEAVIEST PRACTICAL LOAD AND STEEP HILL METHOD

When the towing or chassis dynamometer methods are not possible, the cooling index can be determined by loading the vehicle to be tested and making runs up a steep slope. The vehicle must have shutters to accomplish the test.

The test itself is run exactly the same as the chassis dynamometer method, see Part 1, on page 48.

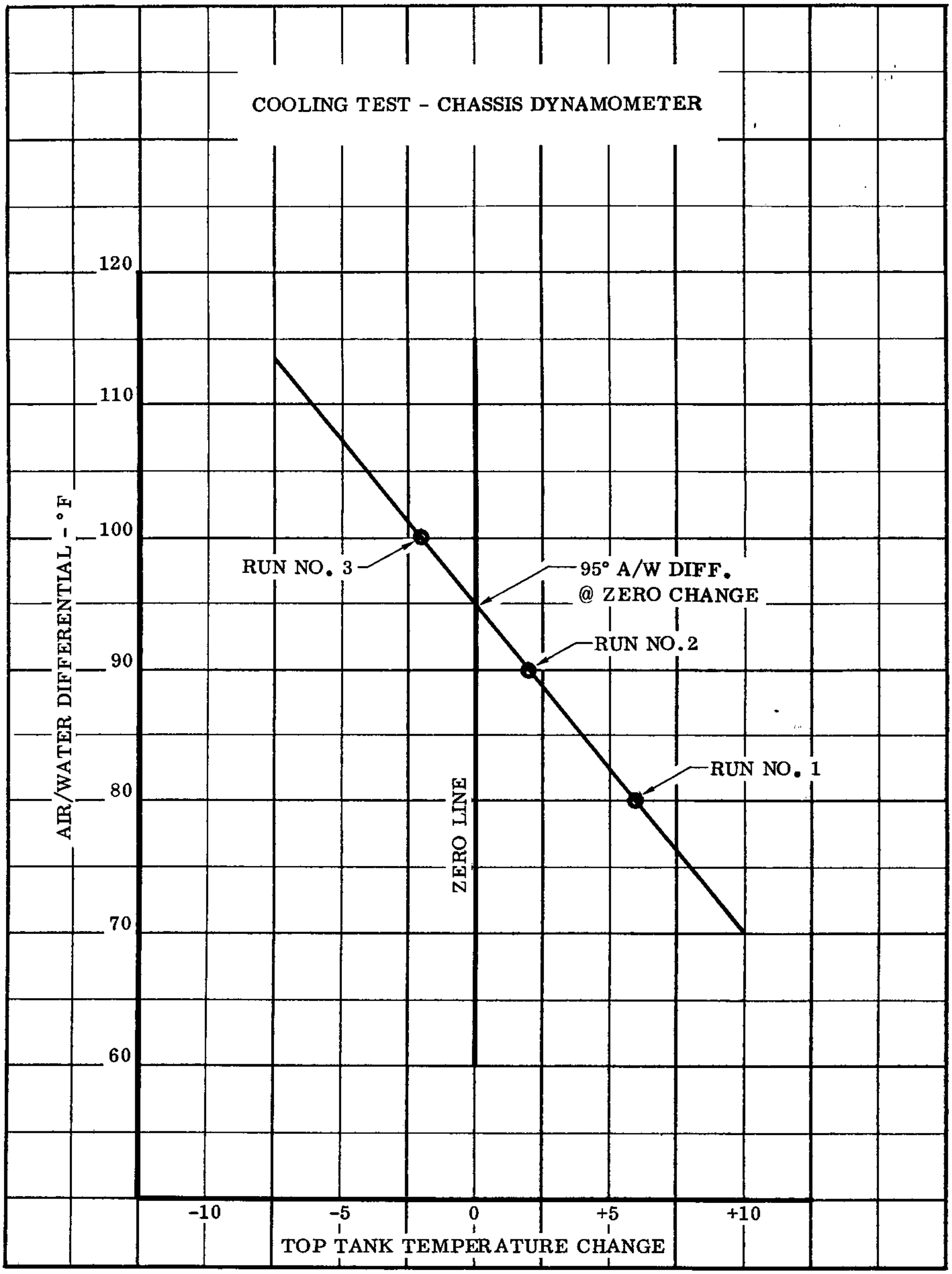

COOLING TEST - CHASSIS DYNAMOMETER
AIR/WATER DIFFERENTIAL - °F
120
110
100
90
80
70
60
RUN NO. 3
95° A/W DIFF.
@ ZERO CHANGE
RUN NO. 2
RUN NO. 1
ZERO LINE
-10
-5
0
+5
+10
TOP TANK TEMPERATURE CHANGE

Figure 35

(c) TEST INSTRUMENTATION AND PROCEDURE

(1) Drain the Coolant and refill with water, noting the total capacity in gallons. Do not use anti-freeze and/or soluble oil mixtures to run the test as these will not have the same heat rejection capabilities as water.

(2) Pressure Cap - All the above tests are to be run with a closed system to obtain the true cooling index. The following shows how the pressure cap should be used in other cooling system checks:

(a) Coolant flow measurement - Cap off
(b) Pump suction check - Cap off
(c) Drawdown test - Cap off
(d) Deaeration test - Cap on
(e) Cooling index - Cap on

(3) Install Blocked Open Thermostat(s) - On engines equipped with lip type seals in the thermostat housing, check condition of seals and replace if necessary. Thermostats should never be removed when running a cooling test as they present a normal restriction to coolant flow and in all cases (except non-blocking type thermostats) block or restrict the by-pass passage. With an unrestricted by-pass there will be less radiator flow and false results will be obtained in the cooling test.

(4) Install Coolant Flow Meter - Location of flow meter will sometimes be dictated by the installation accessibility but the venturi or turbine type meters will work equally well on the suction or discharge side of the coolant pump. Connect both legs of a "U" tube mercury manometer to two paralled taps on the flow meter. Fill manometer above the mercury zero line with coolant and evacuate all air from the hoses and manometer. This is required to obtain correct differential pressure reading.

(5) Install Thermocouples as follows:

(a) Engine Coolant Outlet:* In water outlet elbow or thermostat housing(s). On "V" engines install one thermocouple in each bank and average the readings.

(b) Engine Coolant Inlet:* At pump to oil cooler connection or in oil cooler housing drain on "V" engines. On inline 71 engines the boss on the water pump body or the lower part of cylinder block water jacket may be used.

(c) Oil Sump:* As near pick-up screen as possible. Use drain plug location in preference to dipstick hole.

(d) Radiator Air:* For suction fans, install a 5-thermocouple harness in front of the radiator to obtain average air temperature. For, blower fans, attach the harness to the fan guard. If oil-to-air coolers are mounted in front of the radiator, the harness should be between the two cores so that actual engine radiator air inlet temperature is measured.

NOTE: If shutters are used, block them open but do not remove as they present a normal restriction to air flow.

(e) Ambient Air: Locate two (2) shaded thermometers or thermocouples at an appropriate location (out of fan blast, radiation, etc.) to most accurately measure the ambient air temperature.

(f) Connect thermocouples to multiposition switch and connect switch master leads to potentiometer. Make sure potentiometer is working properly and is zeroed in. A thermos bottle full of crushed ice and water makes a convenient check on the potentiometer. Insert thermocouple so it is well submerged in the ice water.

NOTE: Iron constantan thermocouples are polarity sensitive--the RED wire is NEGATIVE and the WHITE is POSITIVE. Make sure the polarities are not reversed. Scrape all wire to assure good electrical conductivity.

(d) RESULTS OF TEST

This test if conducted as outlined above will give the cooling index of a given cooling system. The temperature rise across the engine can be determined. The air-to-water differential or air-to-boil figure (see explanation and recommendations on page 6) will tell whether the unit is adequately cooled for the application or if it is undercooled or overcooled. If it is undercooled, that is, the A/W differential is too high or the ATB is too low, then changes to the system will be required to bring the unit to recommended specifications. If excessive cooling is indicated, then the fan can usually be slowed down or changed to save fan horsepower or a less dense radiator can be used. The test should be repeated to make sure the unit does not then become undercooled.

2. DEAERATION AND COOLANT FLOW

This test is performed at top governed RPM but no load (vehicle stationary). Blocked open thermostat (s) still installed. Pressure cap closed to atmosphere. Have several suitable containers handy to catch and measure water loss from hose to overflow tube. Air purge meter connected to lowest point on cylinder block on discharge side of water pump. Set air regulator valve on purge meter above pump pressure to assure air going into engine rather than water into meter.

With coolant at operating temperature, pressure cap off and engine at top governed RPM, note flow (or pump suction) with no air injected. This should be approximately the same as the curve shown on pages 13, 14, 15 and 16 depending on the engine and type of thermostat used. If coolant flow at governed speed and operating temperature is substantially less than the appropriate curve, check items affecting pump flow on page 17.

Begin air injection at the rate of .025 CFM per cylinder and allow the system to stabilize. Note coolant flow and loss. Increase rate to .05 CFM per cylinder and note flow and loss of coolant. Repeat until desired rate of .1 CFM per cylinder is being steadily injected and system is stable. The air induction meter can be read directly only at 15 PSI and 100°F air temperature. Deviations from this condition require a correction to the observed rate of air induction which is easily made on Figure 37, page 58. This chart corrects for air pressure only; air temperature correction is considered to be negligible.

If coolant loss is 10% or less of the total capacity with .1 CFM (corrected) per cylinder of air injected and pump flow under these conditions does not drop below 50% of its original (no air) value, the system will be satisfactory.

If a given system is not capable of handling .1 CFM per cylinder without excessive coolant loss and/or loss of pump flow the system should be modified to meet the specification. This can often be accomplished by the addition of a surge tank or redesign of the baffles in the radiator top tank. See page 25. Closed thermostat deaeration should comply with the .1 CFM limit without having the coolant pump become airbound.

3. LOW COOLANT LEVEL CHECK

This test is sometimes called a "drawdown" test since its purpose is to see how much coolant can be drained from the system before pump flow is seriously affected. Remove the air purge meter connection on the cylinder block and install a suitable hose and shut-off valve. Other equipment and operation will be the same as for the Deaeration Test.

With a full cooling system pressure cap off, coolant at operating temperature and engine at governed speed (no load) drain recommended quantity shown on page 59, Figure 38. Depending on the system capacity being checked, drain coolant in increments, building up to the satisfactory limit if possible, and determine maximum quantity required to cause aeration and effect pump flow or pressure. A flow meter may be used to determine the above but experience has shown that the use of a sight glass and pressure gage will result in a more accurate answer. In either case pump flow or pressure must not drop from its original value, within the limits specified, to assure safe operation with a low coolant level.

B. RESULTS OBTAINED FROM SOME TYPICAL COOLING TESTS

The following results of actual tests conducted in the field indicate how the data obtained can be used to improve the cooling system of a given application. They illustrate that much can be learned from these tests that will benefit the customer and/or pinpoint the source of field problems.

8V-71 Hi-Output Truck Engine: These tests were conducted on the prototype vehicle to evaluate the acceptability of the cooling system. Full load was provided by a towing dynamometer. The initial tests showed too low an ATB and since the unit already had an ample fan and drive ratio, a radiator having another row of tubes and one more fin per inch was tested. This improved the cooling index to the point where a slower fan drive ratio could be used and adequate cooling still maintained. The data is shown in graphic form in Figure 39, page 60. If the tests had not been run, the trucks would have and inadequate cooling system which could have caused overheating problems in summer operation. The tests prevented this situation from occurring and also showed that fan horsepower and, therefore, fuel consumption, could be reduced.

12V-71 Hydrofrac Units: In this case a group of 12V-71 engines were operated side by side on a drilling rig. A cooling test was conducted in an effort to diagnose engine problems that had occurred. With all engines running together, the test showed that while the cooling system itself was acceptable, the 36" blower fans were recirculating air from one unit to another and the air entering the radiator was 54° hotter than the true ambient. This gave an effective air/water differential of 139° greatly in excess of the recommended figure of 85°. The solution was to use a larger fan to improve the core coverage, make a better shroud to increase fan efficiency and move the units apart to reduce recirculation. This test showed the importance of testing a unit under actual operating conditions; if one unit had been tested with the other shut-down, recirculation would not have occurred and the trouble would not have been located.

6V-53 Highway Truck: This test was also conducted as a result of a field complaint. Examination of the installation indicated the upper radiator connections were probably restricting water flow. A flow check was made and showed water flow only 50% of normal rate. A cooling test substantiated the low flow rate by indicating excessive temperature rise across the engine. The connections were revised and the test temperature rise across the engine. The connections were revised and the test repeated. The data is shown in Figure 40, page 61, and shows that the final system provides ample cooling. A further study of this application indicated a reduced fan drive ratio could be used without adversely affecting the cooling and this was also incorporated by the OEM.

SUMMARY:

The list of cooling tests that have been run on Detroit Diesel Engines is extensive. The examples mentioned above only indicate how some of these tests were able to improve a given application. Each test is different and will give different results. Each test can be considered a challenge - to the designer to see if the final results agree with his prognostications and to the person running the test, who has a unique opportunity to review the data and offer constructive suggestions for improving the total performance.

DE-2804

DETROIT DIESEL

COOLING TEST DATA SHEET

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 ______

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 ______________

FIGURE 36

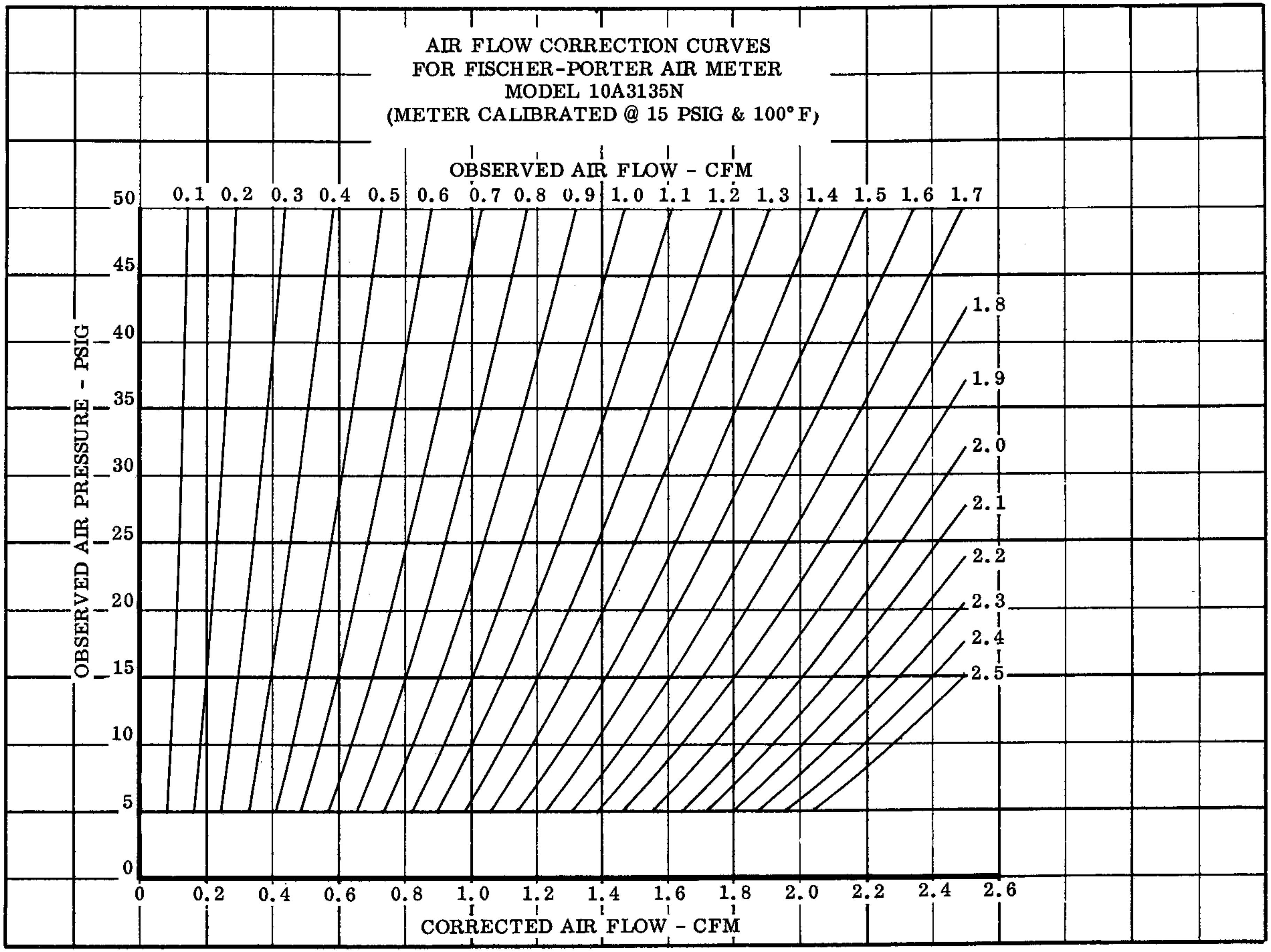

AIR FLOW CORRECTION CURVES
FOR FISCHER-PORTER AIR METER
MODEL 10A3135N
(METER CALIBRATED @ 15 PSIG & 100° F)
OBSERVED AIR FLOW - CFM
0.1 0.2 0.3 0.4 0.5 0.6 0.7 0.8 0.9 1.0 1.1 1.2 1.3 1.4 1.5 1.6 1.7
1.8
1.9
2.0
2.1
2.2
2.3
2.4
2.5
OBSERVED AIR PRESSURE - PSIG
50 45 40 35 30 25 20 15 10 5 0
0 0.2 0.4 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

Figure 37

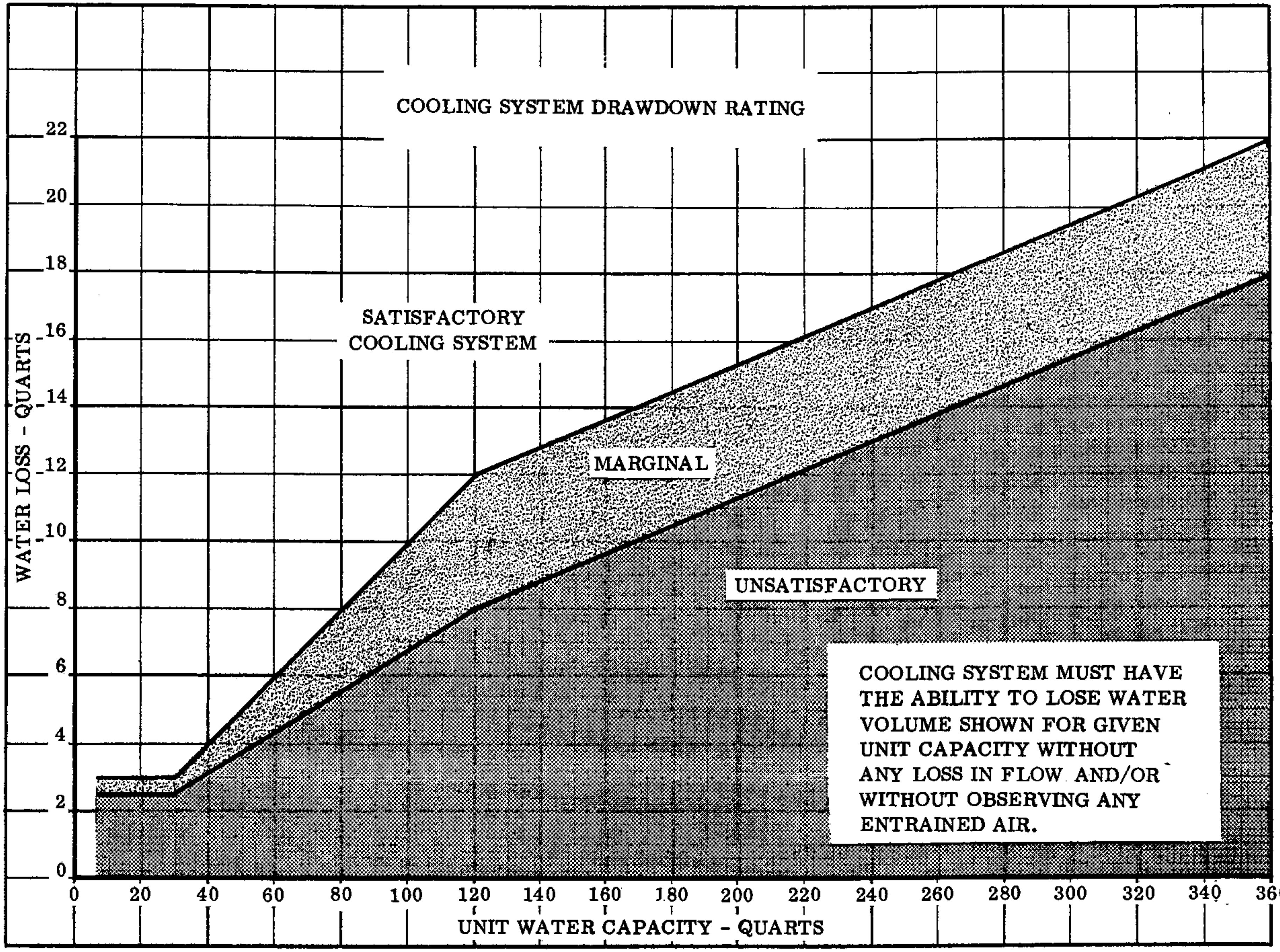

COOLING SYSTEM DRAWDOWN RATING
SATISFACTORY COOLING SYSTEM
MARGINAL
UNSATISFACTORY
COOLING SYSTEM MUST HAVE THE ABILITY TO LOSE WATER VOLUME SHOWN FOR GIVEN UNIT CAPACITY WITHOUT ANY LOSS IN FLOW AND/OR WITHOUT OBSERVING ANY ENTRAINED AIR.
WATER LOSS - QUARTS
0
2
4
6
8
10
12
14
16
18
20
22
UNIT WATER CAPACITY - QUARTS
0
20
40
60
80
100
120
140
160
180
200
220
240
260
280
300
320
340
360

Figure 38

EFFECT OF VEHICLE SPEED ON COOLING

8V-71T ENGINE
N70 INJECTORS - 2100 RPM FULL LOAD
26" 6 BL x 2 3/4" P.W. FAN

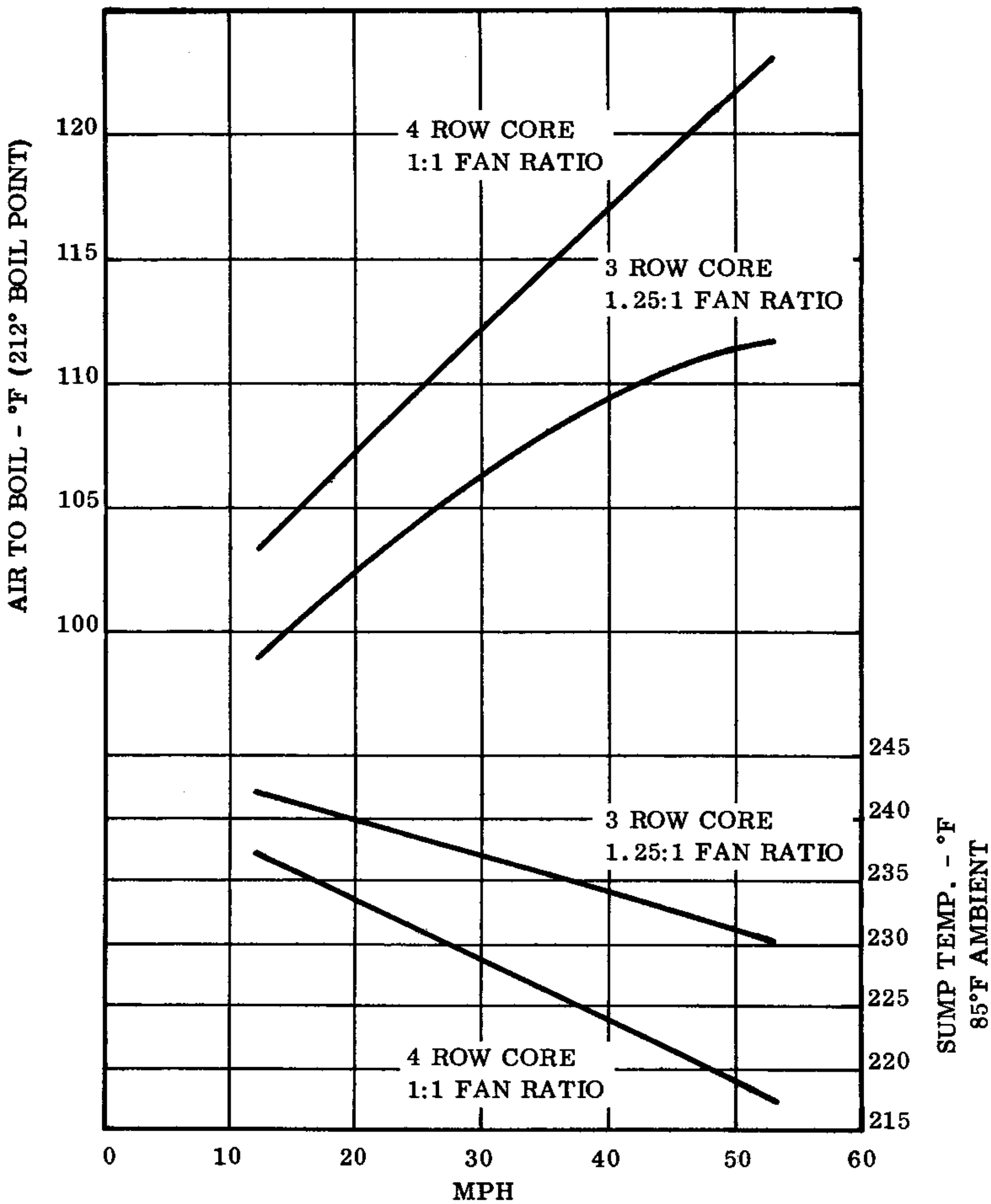


Figure 39

COOLING COMPARISON

6V-53 TRANSIT MIX

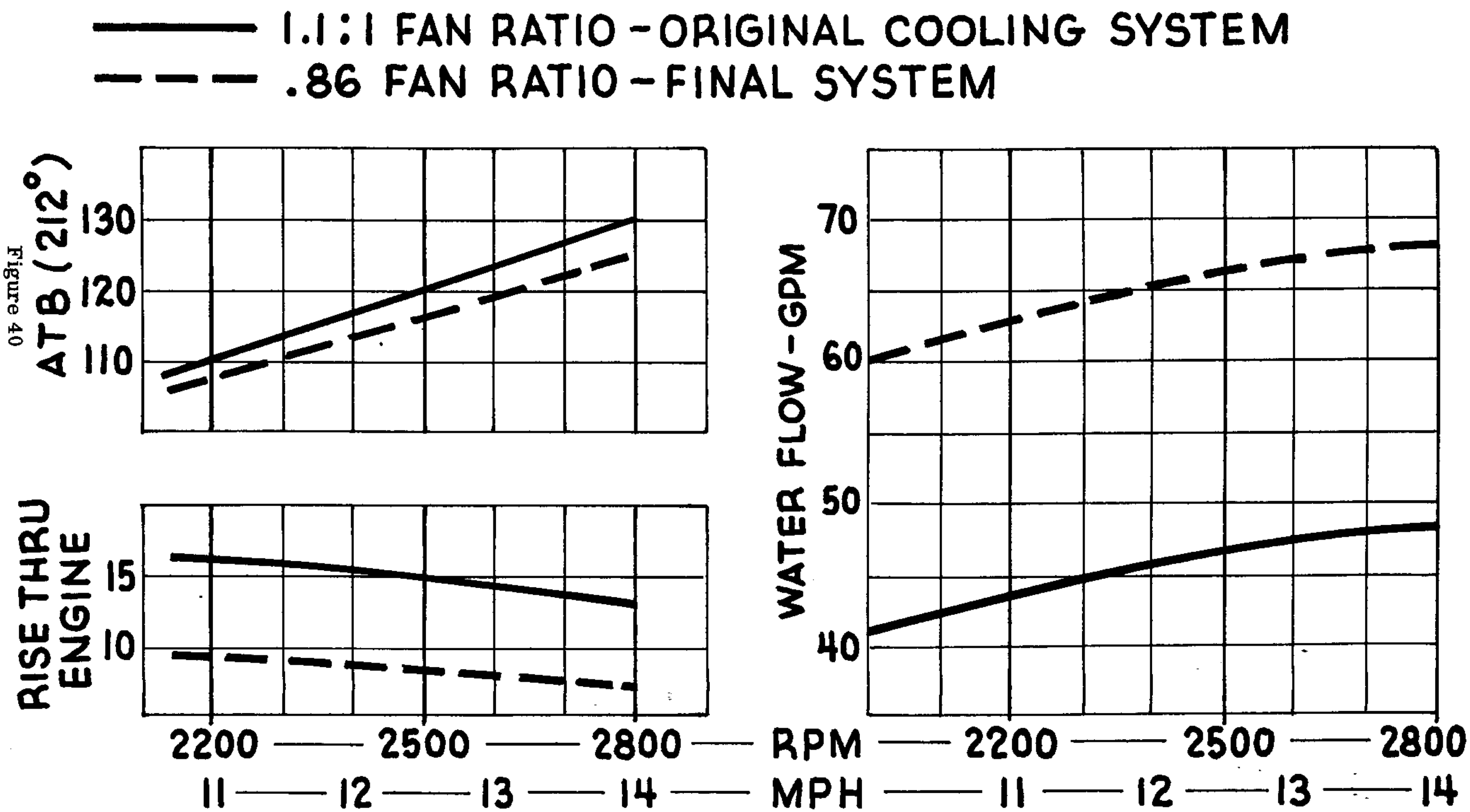


Figure 40

DDA AUXILIARY ENGINES APPROVED BY LLOYD'S REGISTER OF SHIPPING (CONT'D)

Engine	Injector	BHP	Full Load
8V-71N	N70	240	1800
8V-71T	N90	295	1500
	N90	335	1800
12V-71N	N70	320	1500
	N70	390	1800
16V-71N	N70	431	1500
	N70	513	1800
12V-149	130	600	1500
	130	700	1800
12V-149T	165	825	1500
	165	975	1800
12V-149TI	180	890	1500
	180	1050	1800
16V-149	130	800	1500
	130	930	1800
16V-149T	165	1100	1500
	165	1300	1800
16V-149TI	180	1190	1500
	180	1400	1800

Detroit Diesel Allison
Division of General Motors Corporation

AIR CLEANER SYSTEMS
FOR
DETROIT DIESEL ALLISON ENGINES

APRIL, 1972.

Engineering Technical Data Dept.

ENGINEERING BULLETIN No. 39

18 SA 0206

ENGINEERING BULLETIN NO. 39

ISSUED APRIL 1972

DATE REVISED	PAGE REVISED	GENERAL DESCRIPTION OF REVISION
7-10-72	6	New picture for Figure 5.
	45	Scale change.
	46	Scale change.

CONTENTS

INTRODUCTION

The purpose of the air cleaner is to protect the engine from air borne dust and the resultant wear caused by abrasives. The piping or ductwork is designed to flow the air from the cleaner to the engine. Collectively they make up a system which imposes a restriction (pressure drop) to the otherwise free flow of air entering the engine. Therefore, it is necessary to consider the cleaning efficiency of the air cleaner, and the restriction to airflow of both the cleaner and the ducting when planning the system. Low airflow restriction and high cleaning efficiency are the keys to a successful air induction system.

TYPES OF AIR CLEANERS

Two general types of air cleaners exist, the oil bath type and the dry type. Oil bath cleaners may be classified as light duty or medium duty, while the classifications of the dry type include light duty, medium duty, heavy duty, extra heavy duty, and extra heavy duty - extended life.

OIL BATH AIR CLEANERS

Description

Oil bath air cleaners, Figure 1 are designed to clean the incoming air by a constant oil washing process. Several designs are available, but they all consist essentially of a wire screen element supported inside a cylindrical housing which contains a pool of oil - the oil bath - directly below the element. Air drawn into the cleaner passes downward towards the oil bath, and undergoes a complete change in direction when it impinges on the oil. During this reversal in flow direction, much of the foreign matter in the incoming air is carried into the oil by its own momentum where it is trapped and settles to the bottom. An oil mist is found in the air as a result of its impingement on the oil bath, and many of the remaining dust particles are collected by these oil droplets. The droplets then adhere to the wire mesh, and cleaned air emerges at the top of the cleaner elements; the separated droplets drain back into the oil bath.

Oil bath cleaners are suitable for certain applications where dust conditions are not severe.

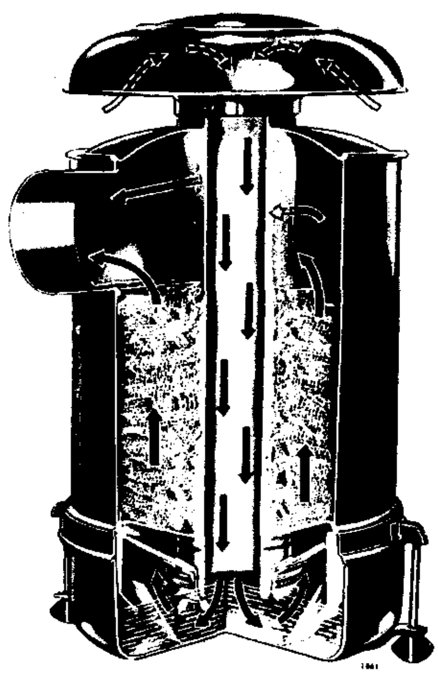


Fig. 1 - Oil Bath Air Cleaner

Characteristics of Oil Bath Air Cleaners

1. Oil bath cleaners are designed for specific maximum and minimum airflows. If used below the minimum design rating, lower cleaning efficiency will result and more dirt will enter the engines; if used above the maximum design rating, oil will be pulled through the cleaner and be carried into the engine.

2. In general, efficiency increases from 96% at low airflow to a maximum of 98.7% at near rated airflow on AC fine test dust and AC coarse dust respectively.

3. Lint or fibrous materials will plug up an oil bath cleaner resulting in oil pull over conditions because of channeling.

4. Oil pull over caused by overspeed, bounce, angular operation, and high dirt level in the oil cup must be considered.

5. The inlet must be protected to prevent the intake of water by either splash or direct rain, otherwise emmulsification, plugging, and pull over can occur.

6. Operating temperature must be considered so that the proper oil viscosity is maintained. Since the oil in these cleaners performs a washing action, it is imperative that the oil be kept fluid at all times. Too light an oil may pull over, too heavy an oil may not function. Generally the air cleaner will function properly with the same viscosity oil as is used in the engine.

7. The oil bath cleaner will handle oil vapor and for this reason is used on applications when the crankcase breather fumes must be directed into the air cleaner; it will also handle exhaust soot.

8. The restriction across an oil bath air cleaner does not increase appreciably as the dust level builds up, and therefore cannot be used as a guide for maintenance intervals; the cleaners must be serviced on a time basis depending on the amount of dirt that settles in the oil cup. Maintenance is required more frequently than for dry type cleaners.

Classification of Oil Bath Air Cleaners

Light Duty Oil Bath Cleaners: These cleaners are recommended only for light delivery trucks operating on paved streets, and industrial power units and generator sets inside buildings.

Medium Duty Oil Bath Cleaners: These cleaners are suitable for highway trucks, farm equipment, air compressors, generator and power units as well as other applications where fairly heavy dirt conditions exist. Compared to the light duty types, these cleaners have larger dust holding capacity.

DRY TYPE AIR CLEANERS

Description

Dry type air cleaners are designed to filter the air through replaceable elements constructed of a specially treated filter paper. The elements are usually pleated in various forms to provide the maximum filtering area. They may be of the single stage type with paper only as the filtering medium, or this may be combined with a precleaning section to form a two-stage cleaner. Some dry air cleaners also incorporate a paper safety element which is in service with and on the cleaner side of the primary, or main, cleaner element. It serves to protect the engine in case of accidental perforation of the primary element and also during servicing of the primary element.

Characteristics of the Dry Type Air Cleaners

1. Dry type cleaners operate at higher efficiency than oil bath cleaners at all airflows. Their operation is not affected by extreme ambient temperature conditions. Regardless of the size, classification, or airflow, the filtering efficiency is usually high, viz., 99.9 percent on all types of dry dust and fibrous material.

2. Exhaust carbon and oil vapor seriously affect the life of dry cleaners and, where encountered, provisions must be made to avoid them.

3. Restriction to airflow increases as the element becomes dirty. Therefore a low initial restriction is desirable for longer element service life. However, a single stage cleaner with a large paper area or a two stage cleaner with less paper may have comparable service life even though the two stage cleaner will have higher initial restriction to flow because of the nature of the first stage inertial action.

   Occasionally a dry cleaner and an oil bath cleaner are used in series. This is done to protect the engine in case the dry cleaner element is punctured. This is poor practice and not recommended because of the resulting high system restriction caused by the addition of the oil bath cleaner. Use of a dry type cleaner with a safety element is recommended for this type of situation.

Classification of Dry Type Air Cleaners

Light Duty Dry Air Cleaners: These cleaners, Figure 2 are those that consist of a small paper or felt element only. They are often used under the hoods of passenger cars or light duty delivery trucks. Though they perform with the same high efficiency that characterizes the dry air cleaners, they have a relatively small filtering area which limits their service life.

Medium Duty Dry Air Cleaners: These cleaners are usually small two-stage cleaners, or single-stage cleaners having a filter medium of increased area over that used in the light duty classification.

Heavy Duty Dry Air Cleaners: These cleaners, Figure 3 have large filtering or element areas together with mechanical precleaners which make them adaptable to conditions of much greater dust concentration. Some makes of heavy duty dry air cleaners do not include a mechanical precleaner but simply employ an oversized filter element to accomplish the same end.

Extra Heavy Duty Dry Air Cleaners: These cleaners, Figure 4 are used for service where dust conditions are so heavy that frequent servicing is required. They have large paper filters coupled with high efficiency mechanical precleaners, Figure 5 and they may also have safety elements for increased reliability.

Extra Heavy Duty Extended Life Dry Air Cleaners: These cleaners are used where service life exceeding 1000 hours is required under extreme dust conditions.

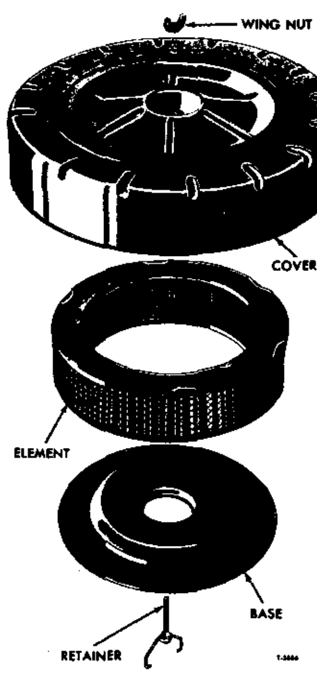


Fig. 2 - Typical Light Duty Dry Air Cleaner

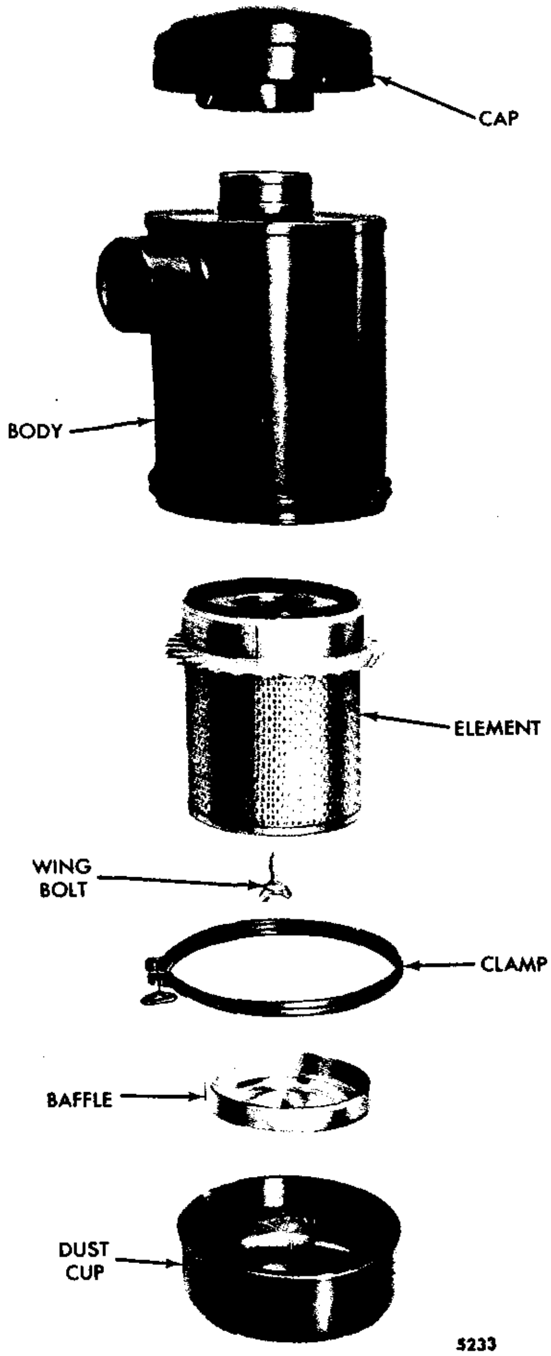


Fig. 3 - Typical Heavy Duty Dry Air Cleaner

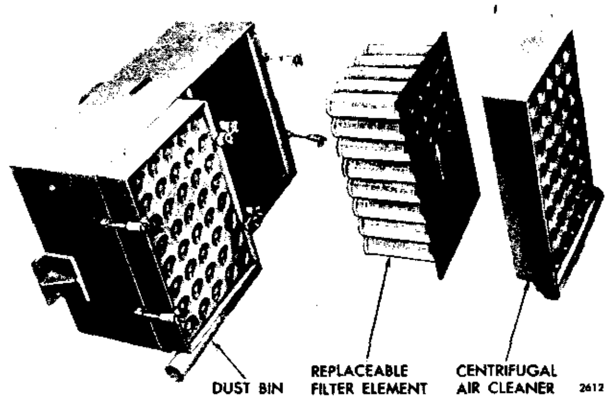


Fig. 4 - Typical Extra Heavy Duty Dry Air Cleaner

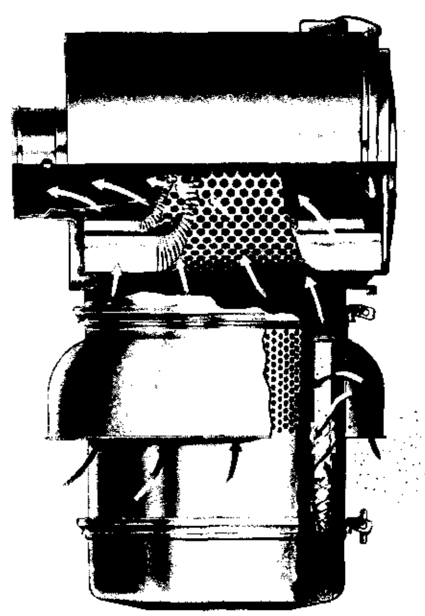


Fig. 5 - Typical Extra Heavy Duty Extended Life Air Cleaner

PERFORMANCE COMPARISON - OIL BATH VS. DRY CLEANERS

As mentioned before, oil bath air cleaners usually operate in an efficiency range of 96% to 98.7%, depending on the airflow through the unit, as shown in Figure 6.

Dry type cleaners operate at 99.8 to 99.9% efficiency regardless of the airflow. This is also shown in Figure 1. Since the engine is only sensitive to the amount of dirt which is allowed to pass through the air cleaner, it can be readily seen that dry type cleaners are generally to be preferred over oil bath.

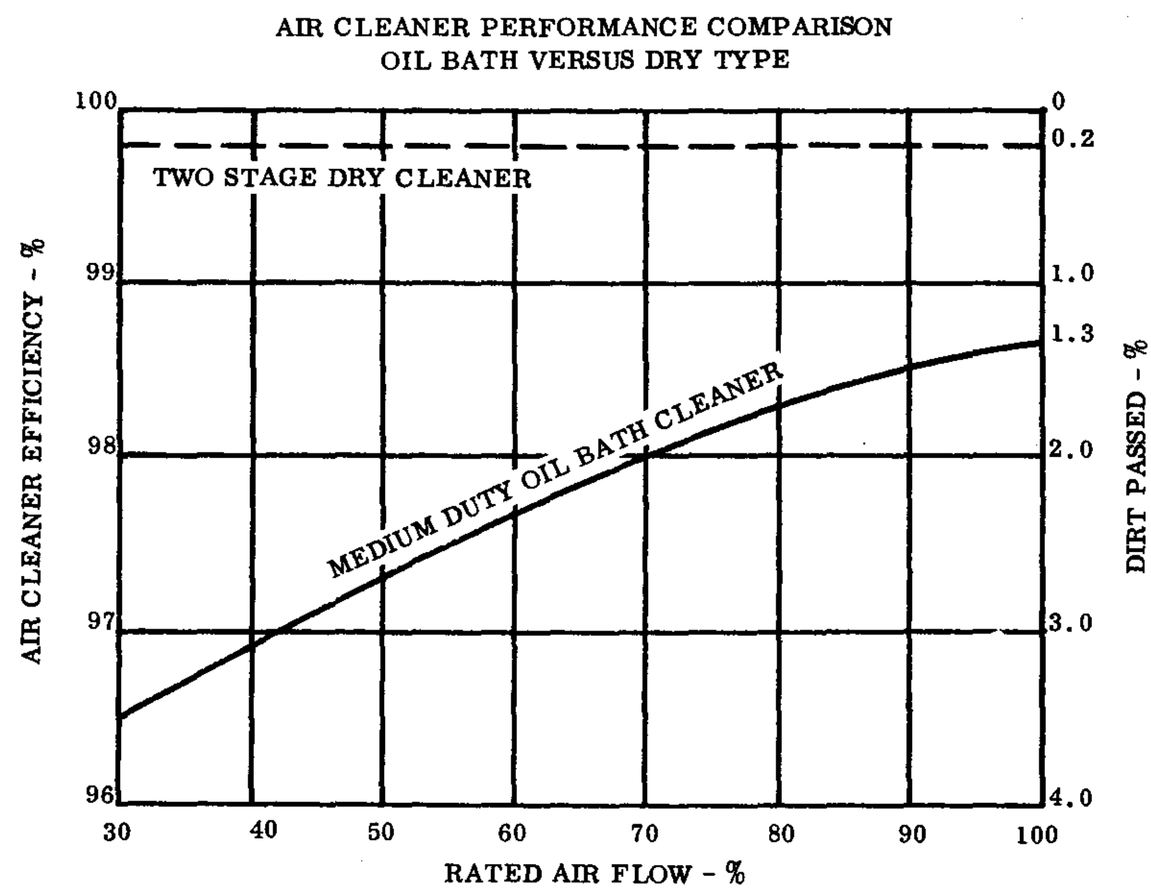


Figure 6

Note: The higher and constant efficiency of the dry type cleaner compared to the oil bath type where the efficiency decreases with decreasing airflow.

PRECLEANERS

A precleaner is a mechanical air-dirt separator which functions via the centrifuge aspirator principle. The purpose of a precleaner is to reduce the burden of air cleaning, which must otherwise be done in the main section of the air cleaner, so that the main cleaner will function for longer periods of time.

Precleaners are not recommended for use with oil bath air cleaners because they do not add to the efficiency of such a system but do add to the airflow restriction. They can perform a useful service ahead of a single stage dry type air cleaners where a beneficial increase in service life can be realized.

AIR CLEANER RATINGS

Air cleaners are rated for flow based on their developed flow restriction or drop. As the airflow rate, in cubic feet per minute (cfm), is increased, the restriction, measured in inches of water--water gage (w.g.), increases. The airflow rating of the cleaner is established at restrictions of 8", 10" or 12" w.g. as the case may be.

GENERAL CATEGORIES

There are two general categories for air cleaners: "Highway" and "Off-highway - Industrial". Highway truck air cleaners are usually subjected to finer types of dust than off-highway vehicles and other types of industrial equipment. For this reason, air cleaners for highway vehicles are tested and classified using laboratory fine dust (AC fine), and the test life obtained is interpreted in miles of air cleaner life.

For off-highway vehicles and other applications, air cleaner ratings are based on coarse dust (AC coarse), and the test life obtained is interpreted in hours of air cleaner life.

Air cleaner classifications are discussed in Section II.

TEST LIFE OF DRY TYPE AIR CLEANER

The selection of an air cleaner which will require as little maintenance as possible is recommended. In order to determine what the comparative operational life of various dry type air cleaners is, and estimate their performance on the job, laboratory test data may be used as a basis.

Test life, as determined by laboratory test, is the time required to raise the initial airflow restriction across the cleaner itself to 20 inches water gage while feeding AC test dust at a continuous rate of .025 grams per cubic foot of air at the rated airflow. The tests shall be conducted in accordance with SAE J726b. (The dust feed rate of .025 grams/cu. ft. is the point of zero visibility and represents a very heavy dust concentration. It is a standard in air cleaner test works.)

The following tables show the relationship of laboratory test life hours to air cleaner usage classification.

Table I

HIGHWAY TRUCK TEST LIFE VS. CLASSIFICATION

Classification	Test Life	AC Test Dust
Light Duty	1 Hr.	Fine
Medium Duty	2 Hrs.	Fine
Heavy Duty	4 Hrs.	Fine

To estimate probable air cleaner service life in miles for highway vehicles, multiply the test life in hours by 40,000 miles. For vehicles operating both on and off-highway, multiply the test life in hours by 15,000 miles.

Table II

INDUSTRIAL & OFF-HIGHWAY TEST LIFE VS. CLASSIFICATION

Classification	Test Life	AC Test Dust
Light Duty	3 Hrs.	Coarse
Medium Duty	5 Hrs.	Coarse
Heavy Duty	10 Hrs.	Coarse
Extra Heavy Duty	26 Hrs.	Coarse
Extra Heavy Duty-Extended Life	37 Hrs.	Coarse

Estimates of off-highway vehicle and industrial unit air cleaner service life can be made by multiplying the test life in hours by 30 to determine the number of hours the cleaner may be expected to operate between service periods.

Note that the test life hours in the above tables are strictly relative numbers. However, they do provide a basis for comparing the life expectancy of various cleaners. For example, a heavy duty cleaner which is rated at ten hours test life on AC coarse dust would have twice the service life of a medium duty cleaner rated at five hours test life when used in the same application.

AIR CLEANER SELECTION

To choose an air cleaner, first determine the general category, whether it be an on-highway truck (fine dust) or an off-highway or industrial machine of any description (coarse dust). Second, determine the classification, whether it be light duty, medium duty, heavy duty, extra heavy duty, or extra heavy duty - extended life. See Tables III and IV. Third, select the appropriate cleaner, based on the installation requirement, from the manufacturers recommendations.

AIR CLEANER USAGE CLASSIFICATION

The following Table III (fine dust) and Table IV(coarse dust) are guides for matching the type of operation with the degree of service and the desired service life. Here again the application is relative, and in regions where more severe conditions exist, the service life can be increased by selecting a cleaner in a heavier service category.

Table III

AIR CLEANER USAGE CLASSIFICATION
AC Fine Dust Categories - On-highway Vehicles

Light Duty - 1 Hour Test Life (Est. 40,000 miles service life)
Trucks - City Delivery
- Milk Delivery
- Fork (Shop)

Medium Duty - 2 Hours Test Life (Est. 80,000 miles service life)
Trucks - On-highway
- City Delivery (Extended service)
- Wheel - Mounted Cranes

Heavy Duty - 4 Hours Test Life (Over 100,000 miles service life)
- On the highway trucks for extended life

Table IV

AIR CLEANER USAGE CLASSIFICATION
AC Coarse Dust Categories
Off-highway and Other Vehicles

Light Duty - 3 Hrs. Test Life (Est. 90 hours service life)
- Marine engines
- Mobile and stationary engines in factories, warehouses, etc.
- Cranes, wheel-mounted

Medium Duty - 5 Hrs. Test Life (Est. 150 hours service life)
- Trucks, gravel and redi-mix
- Generator sets
- Air compressors, pumps
- Cranes, wheel-mounted

Heavy Duty - 10 Hours Test Life (Est. 300 hours service life)
- Trucks, off-highway, logging
- Tractors, wheel agricultural
- Motor graders
- Trucks
- Tractors, crawler, small
- Scrapers
- Cranes, shovel

Extra Heavy Duty - 26 Hours Test Life (Est. 780 hours service life)
or
Extra Heavy Duty - Extended Life - 37 Hours Test Life (Est. 1,110 hours service life)
- Scrapers (large and rear engine)
- Rock drills, self-contained
- Cranes and shovels (rough terrain)
- Air compressors (rock drilling and quarrying)
- Tractors, full-tracked, low speed

Note: The applications in Table IV are only approximate. Certain local conditions can affect the application so as to place it out of context. All possible degrees of service should be considered before classifying an application. Local dust conditions, heavy exhaust soot concentrations and long duty cycles influence the selection.

AIR CLEANER AND DUCT INSTALLATIONS

The total air induction system includes not only the air cleaner but all the ductwork and the engine inlet. Even though the air cleaner may be carefully selected, ductwork is an item which requires careful planning for satisfactory results. Since the maximum permissible air inlet system restriction is measured at the engine blower or turbocharger inlet, all restrictions caused by the piping detract from the ultimate air cleaner service period or life.

AIR CLEANER INLETS

Air cleaner inlets, whether underhood or in open areas, must be installed in protective locations. Air inlets must be directed to guard against driving snow, rain, sleet, road dust, high concentrations of airborne dust and other foreign material entering the cleaner.

UNDERHOOD OR ENGINE MOUNTED AIR CLEANERS

Underhood or engine mounted air cleaners require special consideration. Often air cleaners are mounted such that they take in underhood air. Although this is convenient, it is generally undersirable to supply underhood air, especially in warm weather when underhood temperatures may reach 160°F or more. The inlet to underhood cleaners should be ducted to the outside for warm weather operation. The possibility of ingesting oil vapors or exhaust into the cleaner must also be considered.

DUCTING

Duct Material

Ideally the ducting for remote-mounted air cleaners is steel or aluminum tubing jointed with reinforced rubber hose and double clamps. Some types of molded plastic products are also suitable for ducting provided they will withstand the conditions of heat, vibration, and hose clamp loads to which they will be subjected.

Elbows, where necessary, should have as long a bend radius as possible. Right angle jointed (mitered) tubing should not be used because of its higher restriction.