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Technical Report **R-008** EVAPORATIVE COOLING OF INTERNAL COMBUSTION ENGINES

NY 012 015-3

24 January 1958

U. S. NAVAL CIVIL ENGINEERING

RESEARCH AND EVALUATION

LABORATORY

PORT HUENEME, CALIFORNIA

EVAPORATIVE COOLING OF
INTERNAL COMBUSTION ENGINES

by

E. J. Beck

24 January 1958

FINAL REPORT

U.S. NAVAL CIVIL ENGINEERING
Research and Evaluation
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SUMMARY

OBJECT OF PROJECT

To make reasonably certain the efficient starting and operating of military construction equipment in extreme low temperature (down to -65 degrees Fahrenheit). This task was related to a project with the same purpose in high temperatures.

OBJECT OF SUBPROJECT

To develop a cooling system for internal combustion engines that will provide uniform heat and oil temperatures throughout the engine and thereby result in more efficient engine operations under low temperature conditions.

OBJECT OF THIS REPORT

To present the results of testing, studying, and developing variations of boiling-condensing cooling systems which maintain constant engine temperatures as a result of boiling the coolant rather than by using thermostats, covers, and other controls.

RESULTS

Cooling of internal combustion engines by allowing the coolant to boil and be condensed in a closed cycle is concluded to be a superior method, with many beneficial side effects other than constant temperature without mechanical control. Possible limitations of the system on particular types of engines are discussed at length, as well as a number of commercial, developed, and proposed variations of the basic application.

ABSTRACT

This paper deals with the desirability and feasibility of cooling internal combustion engines by allowing the coolant to boil, then separating and condensing the vapors in a closed cycle at atmospheric or slightly higher pressure.

The general conclusion reached is that cooling by boiling is in practically every respect a superior process, and will prevent the accelerated wear and sludging associated with cold ambient temperature or light loads. Some potential fuel savings are seen through reduction of auxiliary pumping and fan power.

It is further concluded that the system can not be applied without hesitation to any and all engines, but that the process whereby specific engines may be examined for vapor cooling is simple and direct. Some tentative design data for application are proposed.

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figure

WEAR IN THE INTERNAL COMBUSTION ENGINE

An extensive survey of the literature indicates that the most probable common source of difficulties in the operation of engines is associated with what are, for an engine, relatively low temperatures in the cylinder wall and crankcase areas. These difficulties, while not readily detected, probably are the source of most of the severe rapid engine deterioration causing need for early replacement. Overheating, while less common in commercial engines, causes more spectacular failures in the form of valve burning, ring sticking, and seizing. Many of the difficulties related to overcooling result from excess air velocities over engines and faulty thermostat action.

The particular phenomenon giving the most trouble is that of condensation within the cylinder area from too low cylinder wall temperatures and from cold walls in the oil sump, which lead to sludging of the oil charge and formation of acids from the combination of products of combustion and condensed water. The trouble can most readily be corrected by the modification of the cooling system to insure that no point on the combustion area walls falls below the local dew point. This has been, with current engine outputs, of the order of 195 to 200 degrees Fahrenheit. To be effective and prevent acute problems from overcooling, minimum cylinder wall temperatures with any cooling system should be of that order, but probably not over 450 to 500 degrees Fahrenheit.

Of the many possible changes in cooling systems investigated, that of using water (or any antifreeze solution of water) and allowing the water to boil to establish the operating temperature, appears to have the most merit. The process of boiling, with extremely high local heat transfer rates, offers all the desirable characteristics in a cooling system as the need is presently seen. The single question of maximum temperature control in the cylinder wall is shown to be well below the critical temperatures established in the literature and in test. Heat transfer per unit area at critical points near the valve seat is shown to be of the order of one-ninth that of the critical value for typical boiling processes. Beneficial side effects of reduced pump and fan power requirements, automatic regulation of flow rates to achieve the cooling, and possible elimination of the pump in most applications are demonstrated. Ready use of the heat reject with no possibility of starving the engine to the point of overcooling is practical. For extremely low (sub-zero) temperatures, completely adequate engine temperatures based on the boiling point of conventional permanent type antifreeze solutions are possible even in lightly loaded or idling engines. This is contrary to the popular conception that the engine does not furnish enough heat to keep itself warm, but needs auxiliary heat from combustion heaters.

The basic cooling systems do not provide for sufficient heat reject to the oil to keep its bulk temperature in the engine in a desirable operating temperature of over 180 degrees Fahrenheit. It is concluded that to exclude condensation entirely the oil temperature should be at least 212 degrees Fahrenheit in the crankcase. No definite upper limit for oil is demonstrated or found to be established in the literature, but extended tests of diesel engines powering generator sets with the oil at 240 degrees Fahrenheit showed this to be entirely reasonable for 2-stroke cycle high-speed diesel engines on extended full load runs. A method of automatically heating to and cooling above the boiling point of the coolant (212 degrees Fahrenheit for water, and about 220 degrees Fahrenheit for 60 percent ethylene glycol, 40 percent water solutions), is devised. It was demonstrated in practical field equipment under both desert and arctic conditions for both gasoline (small truck) and diesel (crawler tractor) engines, as well as generator sets powered with both large and small 2- and 4-stroke cycle diesel engines.

A beneficial side effect of letting the engine boil accrues from the fact that water-to-air personnel heaters will be receiving coolant (a mixture of liquid and steam) at a higher temperature, thus improving the output. The need for some modification in the location or inclusion of a small circulation pump on those units from which the coolant circulating pump of the engine is omitted is seen as a complication which possibly could be avoided by changing the heater to reduce friction loss, and by increasing the hose sizes and shortening their lengths.

In short, the only disadvantage in the boiling of an engine seen at this time is in possible steam binding in selected engines with tortuous passages, small block-to-head porting, etc. Solutions in the form of hose bypasses are suggested, although for most applications selection of engines with adequate passages is suggested. Steam binding could result in such high temperatures in the valve areas of "L" head engines as to prevent proper valve cooling by the valve seat metal. Only one example is cited in the experimental work, and the particular engine, a 4-cylinder utility truck engine, gave excellent service with boiling condensing applications when installed in a truck. It gave trouble only when used to drive a generator at 1800 revolutions per minute.

Design criteria and method of approach are established and illustrated. Critical areas of temperature measurement within the engine to determine the suitability of the engine are described. The minimum instrumentation necessary to determine the quality of installation is discussed. It is concluded that the system is directly applicable to virtually all stationary and generator applications, with some special attention necessary to those units using water-cooled manifolds to insure water flow by arrangement, not pumping.

Application to mobile equipment has for some time been limited by the demand of buyers. Current commercial systems are available which could be applied to a limited number of units employing particular engines. Development history of a system suitable for application to all stationary and mobile installations is given, termed the "NAVCERELAB Boiling-condensing Cooling System." Methods of preventing freezing of condensate in condensers at very low temperatures are discussed and illustrated.

Automatic temperature control of the oil near the boiling point of the condensate (normally water) is discussed as desirable, and a method of accomplishing this without mechanical control, again using the boiling point as the "thermostat" is illustrated. The literature is incomplete in the respect that an "ideal" oil temperature is not discussed. Evidence was found that 180 degrees is probably a very satisfactory temperature while 290 degrees is too high. The optimum is projected in this report as being at or slightly above the local boiling point of water, or safely high enough to avoid retention of condensation formed on the upper crankcase walls. Oil temperatures as high as 290 degrees Fahrenheit are discussed for spark ignition engines operating under desert conditions. Controversial opinions may be found on the effect of raising oil temperatures on the life of an engine and particularly the pistons. One source alleges that raising the engine temperature will raise the oil temperature and decrease the temperature difference between the oil and the piston crown, making piston failure more likely. Another source, on the other hand, contends that raising the oil temperature slightly increases the flow with viscosity decrease, thereby aiding piston crown cooling. In increasing the temperature of SAE 30 weight oil from 170 to 210 degrees Fahrenheit, the kinematic viscosity will decrease from 23 to 12 centistokes, an appreciable amount. The difference in temperature between the oil and the piston crown with the latter at about 600 degrees Fahrenheit, on the other hand, would be reduced from 430 degrees to 390 degrees, or a comparatively small percentage. The stand of the latter in favor of raising the oil temperature a small amount and lowering the viscosity, appears to be the more credible, so long as temperatures high enough to cause severe oxidation of the oil are avoided.

Part I. INTRODUCTION

Chapter 1

THE INTERNAL COMBUSTION ENGINE

The modern high-speed, high-output internal combustion engine is a remarkable device. Its growth, production, care, feeding, and retirement have been the basis for the second largest industry in a mechanized world since the start of the 20th century. The production, care, feeding, and retirement of people alone surpasses the internal combustion engine in economic importance. Machines powered by engines have been the basis for development and mechanization of many other major industries as, for example, construction, food production, and the tourist trade. An internal combustion engine is an improbable device because it subjects its parts to the most severe mechanical, thermal, and chemical stresses conceivable, all with an amazingly high reliability. The expected life varies with application from as little as about 100,000 miles for automobile engines to as great as the equivalent of several million miles in certain well-regulated stationary applications. This difference in life has been the subject of countless investigations into design, maintenance, feeding, and environmental conditioning practices to attempt to extend the useful life.

Controlled Variables

Most of the investigations of the internal combustion engine have had as controlled variables the temperature in the area of the cylinders and valves. The selection of a suitable temperature for the apparent environment of the cylinder area by individual manufacturers appears to be based largely on convenience in handling a particular cooling agent for liquid-cooled engines, and entirely unrelated to the temperatures occurring in air-cooled engines.

It might be hypothesized that there is an optimum temperature for each engine part and material which would give highest efficiency and extended life. An extensive literary search has not disclosed any unified set of temperatures. Rather, the variation in operating temperatures of engine parts is so great as to suggest that internal combustion engines, and the materials associated with their construction, are largely insensible to their environment, and will tolerate any but the most unreasonable treatment.

An ancient adage regarding the cooling of internal combustion engines relates that they are cooled not to improve their operation, but to prevent their destruction. In the theoretical consideration of heat engines, the nature of the process is such that any heat removed from the working fluid except as useful work tends to reduce the maximum efficiency attainable. Perfect cycles used for instructional and descriptive processes involve expansions without heat transfer. The loss in efficiency associated with heat rejection is generally appreciated even in nonengineering circles.

Examined microscopically, practically every facet of engine design and material selection is based, either from theory or practice, upon the heat transfer at trouble spots to maintain reasonable temperatures. Macroscopically, cooling of the cylinder-valve area and crankcase lubricant by suitable media appears to receive the bulk of attention. The two generally used media, water and air, are chosen on the basis of cost, weight, and the preference of the designer. Reciprocating aircraft engines are, in general, cooled by air, and any penalties for the use of air are accepted as necessary in the selection of materials, fuels, and lubricants, all of which must be premium at the very high outputs achieved. Light and medium duty automotive and industrial applications, on the other hand, usually employ a liquid coolant in the cylinder-valve area, with random air cooling of the crankcase. In very heavy duty automotive and industrial applications, or where installation and cost demand longer life, the crankcase lubricant is usually selectively cooled, or heated, in a suitable area of the cylinder coolant circuit. In some extremely heavy duty applications, oil is methodically cooled in its own section of radiator, similar to the practice in aircraft.

Interdependent Variables

The total engine process in an engine is so complex as to defy complete separation of a parameter to study the effect of its variation on the whole. A small fraction of the large number of variables studied by enterprising investigators is reflected in the bibliography. Among the better recognized variables affecting the general health of an engine are the following:

1. Cylinder wall temperature.

2. Variation in cylinder wall temperature, circumferentially and longitudinally.

3. Fuel characteristics, among which are viscosity, stability, burning characteristics, self-ignition characteristics, end boiling point, ash content, volatility, heat of evaporation, and sulfur content, to name some of the more obvious.

4. Compression ratio.

5. Ignition timing, or injection timing in diesel engines.

6. Volumetric efficiency, which in turn is related to carburetion, fuel characteristics, manifolding, controlling (throttling), surface roughness, valve area, lift, timing, etc.

7. Valve temperatures.

8. Combustion chamber deposits and their composition.

9. Charge density, which in turn is affected by volatility, manifolding, valve action, size and lift, cylinder and manifold temperatures, air temperature, humidity, filter pressure loss, etc.

10. Lubricant, its viscosity, volatility, stability, oxidation resistance, crankcase charge temperature, detergency, foaming characteristics, etc. The crankcase charge temperature in turn is related to ambient temperatures, volume of flow, contact with heated parts, coolant temperature, use as coolant for engine parts, such as, valve guides, rocker arms, etc., in the particular engine, as well as conditioning accomplished in coolers, heaters, engine enclosures, winterfronts, etc.

11. Vitiation of the oil charge, which is dependent upon the variables listed above under 10, plus filter characteristics and condition, engine condition, blowby, ambient dust, location of

air intake, manifold and ducting condition, fuel burning, dilution, cylinder oil film temperatures, chemical activity of engine materials, individual parts temperatures, crankcase ventilation, lubricant compounding, etc.

With such an imposing list of interdependent variables, it might be hypothesized that testing of the effects of varying any particular one might yield results of questionable value insofar as the results apply to service conditions. This is almost universally admitted by the investigators and is the basis of justification for expensive, extended service test under fleet conditions.

Because of the immense economic importance of engine life, the number and variety of laboratory and field test efforts to isolate deleterious environmental factors affecting engine life are great. Further, there is a general reluctance, if not inability, to discuss the effect of variation in environmental parameters on engine hygiene except in a two dimensional plot. This paper will follow established practices in this respect.

However difficult it may be to correlate the findings of the diverse investigations reported in the bibliography, a parameter common to practically every case is some measure of the "engine temperature." Most frequently this is the maximum mass coolant temperature in liquid-cooled engines, and either spark plug base or cylinder wall temperature in air-cooled engines. Some of the more recent papers report cylinder wall temperatures of liquid-cooled engines. Where available, cylinder wall temperatures are preferred. They are usually measured by fine wire thermocouples imbedded in the wall. In the experimental work reported here, the thermocouple is approximately at the mid-point of the cylinder wall and held in place with a small slug of peened-solder. By carefully controlling the depth, surface temperatures can be calculated more exactly than they can be measured. For most purposes this calculation is an unnecessary refinement.

Over-Cooling

In the earliest liquid-cooled engines, difficulties with sludging and rapid wear were recognized as being attributable to over-cooling at light loads and cool ambient temperatures. In the low output engines of the day, circulation of the coolant was frequently accomplished by gravity with boiling or near boiling common at high loads or warm weather. The model "T" Ford utilized this system. At least one farm tractor with gravity circulation of coolant is still made. Gravity circulation also is used in a number of heavy duty, long life, horizontal single cylinder industrial engines. In the simplest of liquid-cooled engines, the coolant was retained in an open hopper around the cylinder, usually horizontal. The coolant was allowed to boil at atmospheric pressure, the steam escaped with most of the rejected heat, and the coolant was replenished by some available supply valved to maintain a set level in the hopper. More imposing systems, utilizing the same boiling technique for cooling, employ ample but simple condensers which return the condensate by gravity to the hopper. They are available on single cylinder gas and diesel engines, and are common in Europe. A particularly novel and new European design incorporates the condenser in a hollow flywheel, with integrally cast fins. Condensate collection is in the periphery and return to the separation area is accomplished by means of an impact scoop, similar to a pitot tube. Velocity of the rotating liquid in the flywheel periphery is converted to pressure in the tube as the condensate builds up to the level of the pickup tube.

As engine outputs increased, gravity circulation of the coolant through a radiator imposed severe limitations in design. Requirement for flow area in the connecting hoses was very high, and boiling at high loads with high ambient temperatures was inevitable. Because the configuration of engine-radiator in use did not provide for ample flow, flow was accelerated by use of centrifugal pumps. While the gravity system was largely self-regulating, the more capable forced-circulation cooling system was not. The literature indicates that boiling-condensing, self-regulating systems were well regarded and the benefits generally recognized at this stage of development. With the advent of the liquid-filled thermostat, the problems associated with overcooling on forced circulation seemed to be eliminated. Simple systems employing ample fans were developed and are currently still the rule on most engines, both mobile and stationary, except those cooled through a heat exchanger and a second reject liquid, e.g., marine applications.

In spite of the apparently theoretically correct practice of using thermostats, rapid engine wear, sludging, high crankcase acidity, etc., attributable to overcooling of specific areas under light load and low temperature conditions, or both, have been the rule rather than the exception. The use of thermostatically controlled shutters, fabric or metal winterfronts, variable pitch or electrically clutched fans is common, and probably should be much more so. Many cases of heavy sludging have been attributed to improper functioning of thermostats. Certain new engines cooled for very high output have extremely long warmup times as installed in passenger vehicles.

This paper is related to the investigation of the advisability, practice, possible difficulties, and gains in using some form of liquid-cooling system in which temperature control is achieved by allowing the coolant to boil, separating the steam produced, condensing it, and returning the condensate to the engine, thus preventing overcooling.

Part II. "NAVCERELAB BOILING-CONDENSING SYSTEM"

Chapter 2

DESCRIPTION

The simplest cooling by boiling system is conceived to be the open hopper (shown in Figure 1) surrounding the horizontal single cylinder engine, with continual loss of steam during operation. The commercial application, the investigation of which was begun in 1951 by the U. S. Naval Civil Engineering Research and Evaluation Laboratory, Port Hueneme, California, is schematically shown in Figure 2, and was applied to the first test unit, a 30 kw generator set driven by a 3-cylinder, 2-stroke cycle unit (shown in Figures 2A, 2B, and 2C). The basic concept of cooling an internal combustion

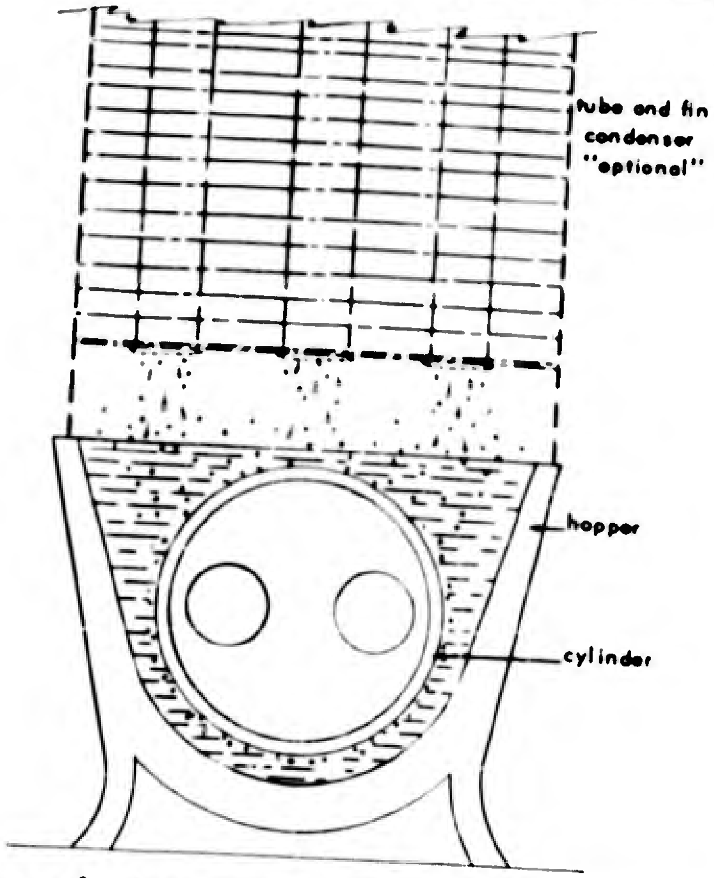
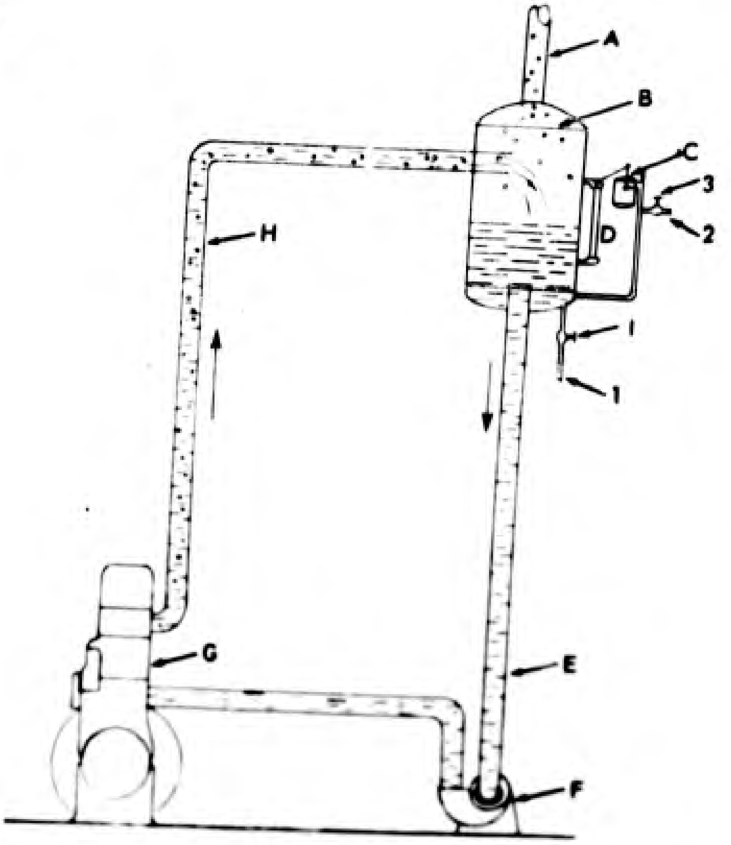


Figure 1. Schematic of hopper-cooled engine cooling system.

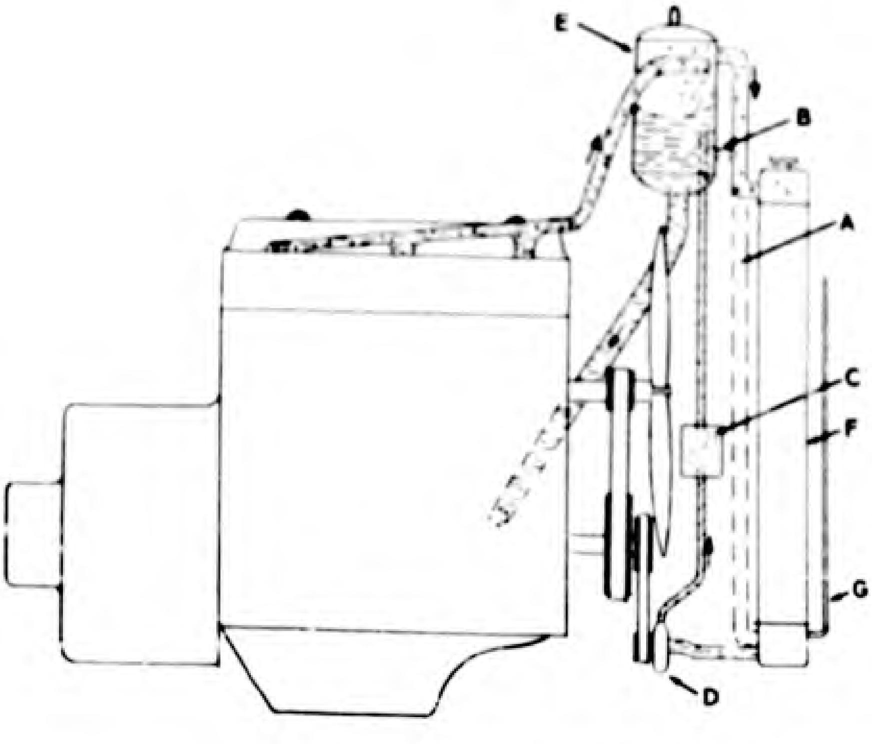


Nomenclature

A steam line to atmosphere or condenser
B steam separator
C automatic water make-up valve and low water level actuating device
D gouge glass
E water into engine
F engine water pump
G engine water jacket
H water out of engine
I blow down valve

1 blow down line
2 make-up water line
3 make-up water line valve

Figure 2. Schematic of commercial application of boiling-cooling system.



Nomenclature

A alternate steam line for cold weather tests
B suggested metered bleed-off for cold weather
C oil cooler, cooled by condensate
D small belt-driven centrifugal pump
E cylindrical separator
F original radiator
G air vent

Figure 2A. Schematic of commercial boiling-condensing system applied to portable generator set.

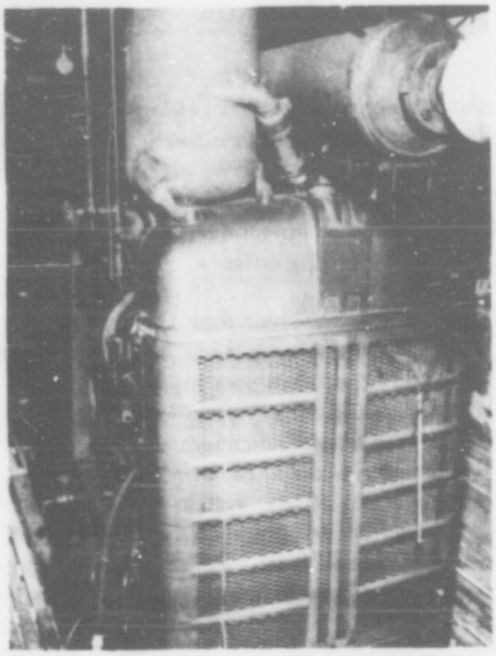


Figure 2B. Generator set (Figure 2A) during hot box tests.

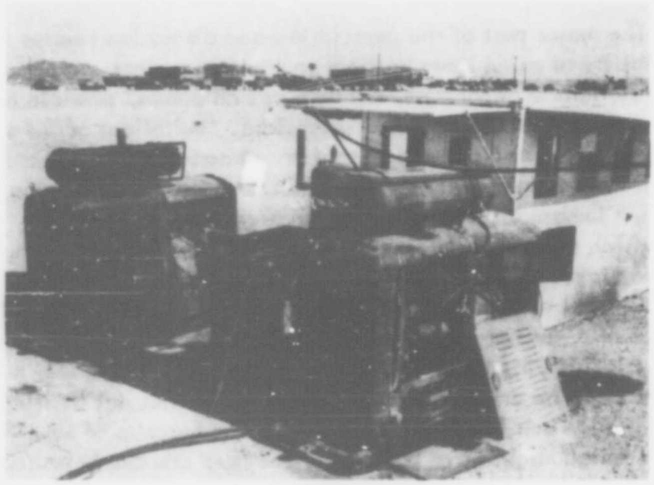


Figure 2C. Yuma setup for generator tests (Figure 2A).

engine by boiling is not patentable, having been a part of the art since the early days of engine development. Certain features of specific systems related to the method of fan control, use of reject steam for driving cooling fans, use of percolating arrangements for condensate pumping, use of fan-condenser combinations as described below, etc., are protected by patents.

In 1952, it became apparent that a successful system utilizing boiling-condensing, while having merit in retrieving condensate lost in high ambient operations, thus possibly solving some of the difficulties in that area, would offer particularly attractive advantages under light load or low ambient temperature operation, or both. For some time it has been considered for all-temperature operation, with the view to develop a simplified system for the entire spectrum of ambient temperatures from -65 to +125 degrees Fahrenheit. Test conditions have been varied from +140 degrees Fahrenheit in hot boxes to -65 degrees Fahrenheit under laboratory conditions, and from +110 to -42 degrees Fahrenheit under field conditions. Since 1954, the experimental work has been largely in the low temperature area because of project emphasis and personnel time limitations. A limited cold weather laboratory and paper study program has been continued during this time.

The major part of the description and discussion relates to the advisability of using some boiling-condensing system, possible configurations of separators, condensers and pumps, possible uses of reject heat, possible critical applications, limitations of the system, design criteria, and probable benefits. Among the systems discussed will be what is termed the "NAVCERELAB Boiling-condensing Cooling System," Laboratory design (shown in Figures 3, 3A, 3B, 3C, and 3D), which represents what is conceived to be a uniquely simple, advantageous system aiming toward complete temperature conditioning of an internal combustion engine under any condition. Specific features of a number of commercially available systems are discussed without identifying the supplier. The contents of this paper are not considered to be critical of proprietary items, thus not limiting its distribution. It is believed that all peculiar features of specific systems which might have patentable features are amply protected, and that distribution of what might normally be considered to be proprietary information can serve only to benefit the patentees.

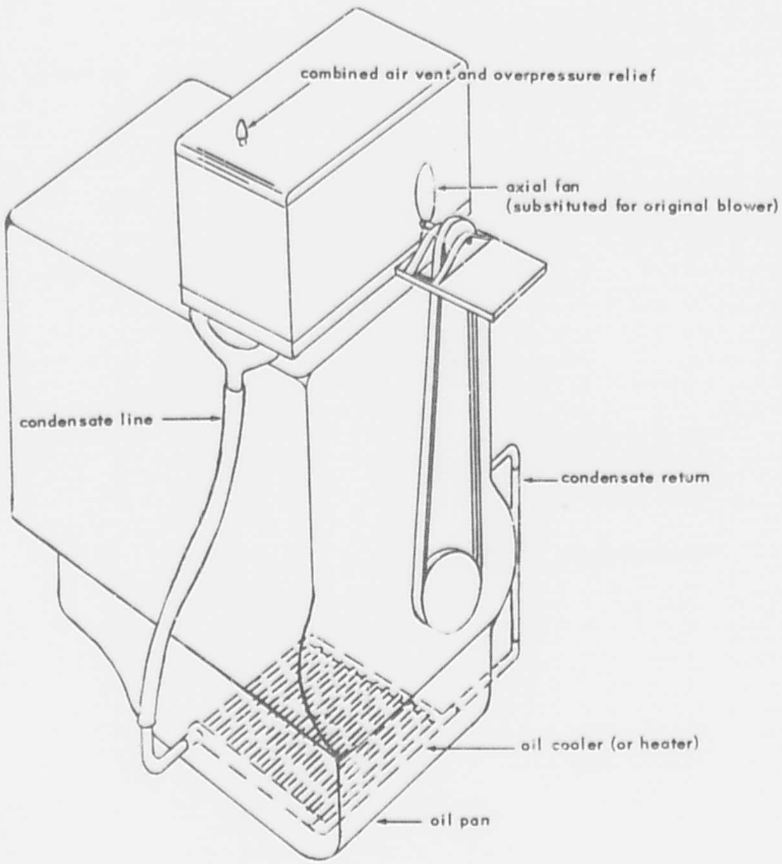


Figure 3. Schematic of Laboratory design of a boiling-condensing system.

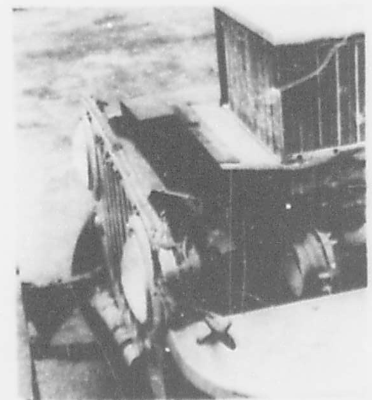


Figure 3A. Boiling-condensing system (Figure 3) applied to 1/4-ton truck ("Jeep").

Figure 3B. Rear of condenser on "Jeep" mounted unit (Figure 3A).

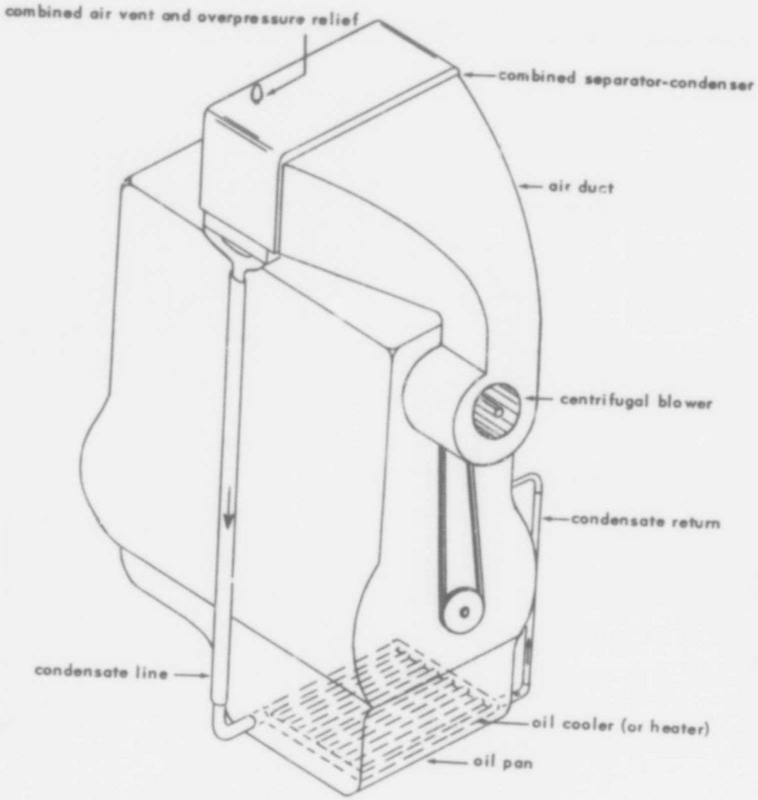


Figure 3C. Schematic of boiling-condensing cooling system (Figure 3) with centrifugal blower.

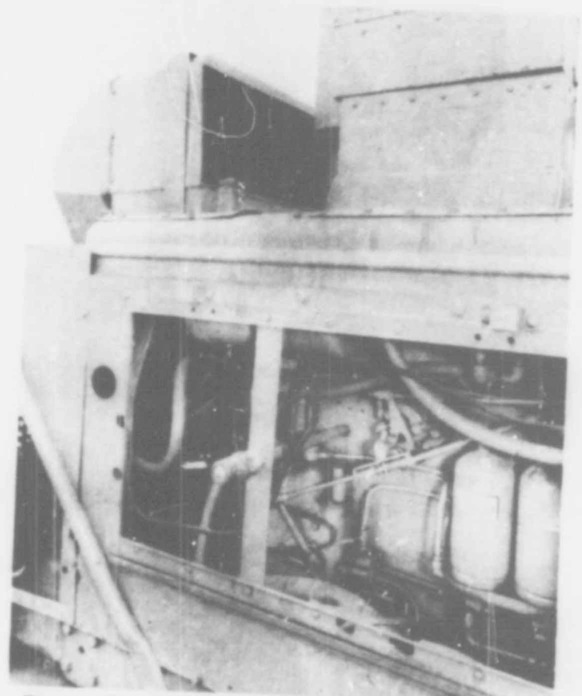


Figure 3D. Boiling-condensing cooling system (Figure 3C) applied to diesel crawler tractor.

Cooling by Forced Convection

The general dimensionless equation for cooling a surface by forced convection,[1]

$$N_{Nu} = C(N_{Re})^n (N_{Pr})^a, \text{ or} \tag{1}$$

$$\frac{hD}{k} = C\left(\frac{DG}{\mu}\right)^n \left(\frac{c_p \mu}{k}\right)^a \tag{1a}$$

describes the rate of heat flow from or to a surface from a fluid without change of state of fluid, and in general allows the selection of suitable fluids, velocities, etc., for either air- or liquid-cooled engines as well as miscellaneous heat exchanger surfaces, vehicle surfaces, building surfaces, etc. Once the cooling (or heating fluid) is known, the necessary mass rates of flow (G) can be calculated, providing that allowance is made for evaluation of the physical properties at a suitable temperature, and providing the flow is known to be turbulent.

This equation allows the prediction that dense fluids of high heat capacity (^cp) and high thermal conductivity (k) will offer rapid heat transfer. If we solve the equation for (h), the thermal conductivity per unit area per temperature difference per single film per unit time, it is seen that the significant physical dimension (D) for the shape being cooled (or heated) and viscosity (μ) appear in both numerator and denominator to different powers, so their importance in promoting heat transfer depend upon the experimental exponents assigned. For many practical cases,

$$h = 0.023\,\frac{k}{D}\left(\frac{DG}{\mu}\right)^{0.8}\left(c_p\,\frac{\mu}{k}\right)^{0.4} \tag{1b}$$

to a close approximation.

In discussing the case at hand, and the effect of coolant velocity on the water side of the cooled engine area, we can consider that the temperature at which the physical properties of the coolant are evaluated remains essentially constant, by virtue of thermostatic control. For a given system or specific area within the system, the value of "h" will vary as (G), the mass rate of flow to the 0.8 power. A typical characteristic is plotted to an arbitrary scale in Figure 4.

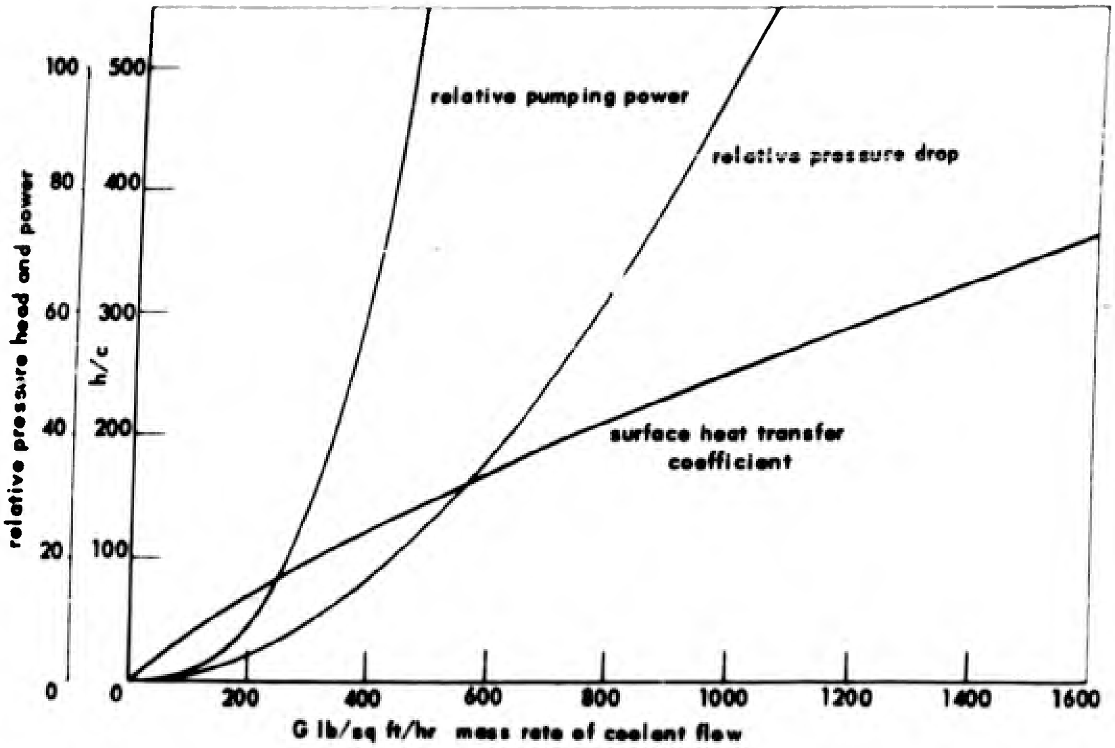


Figure 4. Graphic representation of poor economy of achieving high heat transfer rates by forced convection.

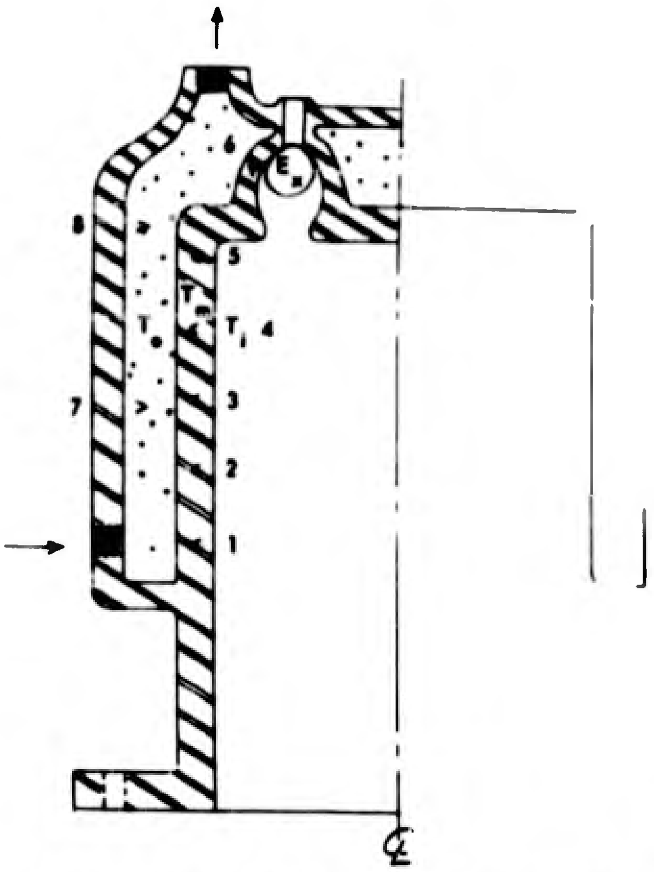


Figure 5. Schematic of diesel engine cylinder used in heat rejection investigations.

In the same figure and again to an arbitrary scale are plotted typical pump pressure and power characteristics. From inspection of the curves it can be seen that while the effectiveness of cooling increases only approximately as the mass flow (which in turn is proportional to the velocity), the pump head and power requirements increase very rapidly. For a given system with a pump of reasonable quality, speed, and capacity, a point will be reached beyond which cooling can not economically be accomplished.

If we assume, in a simple cylindrical coolant space typical of the engine shown in Figure 5, temperature increase recommended by Marks[2] we obtain:

Maximum b. hp at 1800 rpm	8
Temp rise coolant °F	10
Coolant flow, gph/b. hp	50 (lb/hr 3275)
D' (calculated) * ft	0.1375
Area cooled, sq ft	0.50
k, Btu/hr/°F/ft^2/ft	0.388
c_p, Btu/lb/°F	1.01
μ, lb/hr/ft	1.005
G, lb/ft^2/hr	104,600
Q, heat reject, Btu/hr	33,000

Substitution of the above values in equation (3) yields a heat transfer coefficient at the metal-water interface of 204 Btu per hr per degree Fahrenheit for the single film.

Solving the equation $Q = -hA \Delta t$ for Δt (2)

$$\Delta t = \frac{-Q}{A\ h} = \frac{33,000}{(204)\ (0.5)} = 323\ ^\circ F; \qquad (2a)$$

$$\text{surface } T = 323 + 155\ ^\circ F = 478\ ^\circ F \qquad (3)$$

The above example assumes uniform heat transfer over the entire cylinder combustion chamber area. Since it is known from experiment and induction that the upper cylinder area and exhaust valve area receive and therefore reject more heat than the lower cylinder area, what appears to be an unreasonable temperature difference, metal to coolant, must further be aggravated in the actual engine. Measurement[3] of cylinder temperatures at the metal-coolant

* W. H. McAdams, p. 201

interface on many engines reveals that cylinder temperatures at the top of the ring travel of 122 degrees Fahrenheit at reasonable loads are quite possible, using copious quantities of very cool water.

From the example it is deduced that the heat transfer in typical engines can not be explained mathematically in terms of forced convection alone. However, the rates of heat transferred from the cylinder walls to the coolant can easily be explained in terms of boiling at the interface and recondensation in the stream. Measurements[1] of electrically heated tubes of heat transfer with reasonably high flux rates show that the surface temperature of the metal is practically independent of velocities and coolant (water) temperature. Temperature differences, metal to coolant, are comparable to those obtained with heat transfer to a boiling liquid at the local saturation temperature. At very high velocities and low heat flux, the surface temperature is related to the value calculated by forced convection equations similar to those above. If, as will be developed at length later, there is a minimum surface temperature below which the engine is actually overcooled and damaged, it appears expedient to select coolant temperatures sufficiently high, and velocities sufficiently low to avoid cooling below some selected value. A cursory study of the art of air-cooling, which will not be covered in this paper, indicates that air-cooled engines with adequate cooling systems at high temperature and load invariably suffer at low ambient temperatures and load. The major effort in improving variable-load air-cooled engines is in devising blower and baffling control to prevent rapid overcooling with load and air velocity change.

It appears, then, that a coolant-cooling system combination should be sufficiently flexible so that it will warm the cooled surfaces which might normally run cold at low temperatures, i.e., transfer heat continuously to the lower cylinder walls from the high heat flux areas around the exhaust valve. At higher loads it should respond immediately to cool all surfaces to a previously determined acceptable value, preferably without mechanical adjustment, throttling, variation in air flow, etc.

The mechanism of surface boiling, with the coolant at the local boiling temperature, appears to have this characteristic. Referring to Figure 6, it will be seen that remarkably high rates of heat transfer are possible without high velocities in the coolant stream. In fact, within certain physical limits, the entire turbulence necessary to

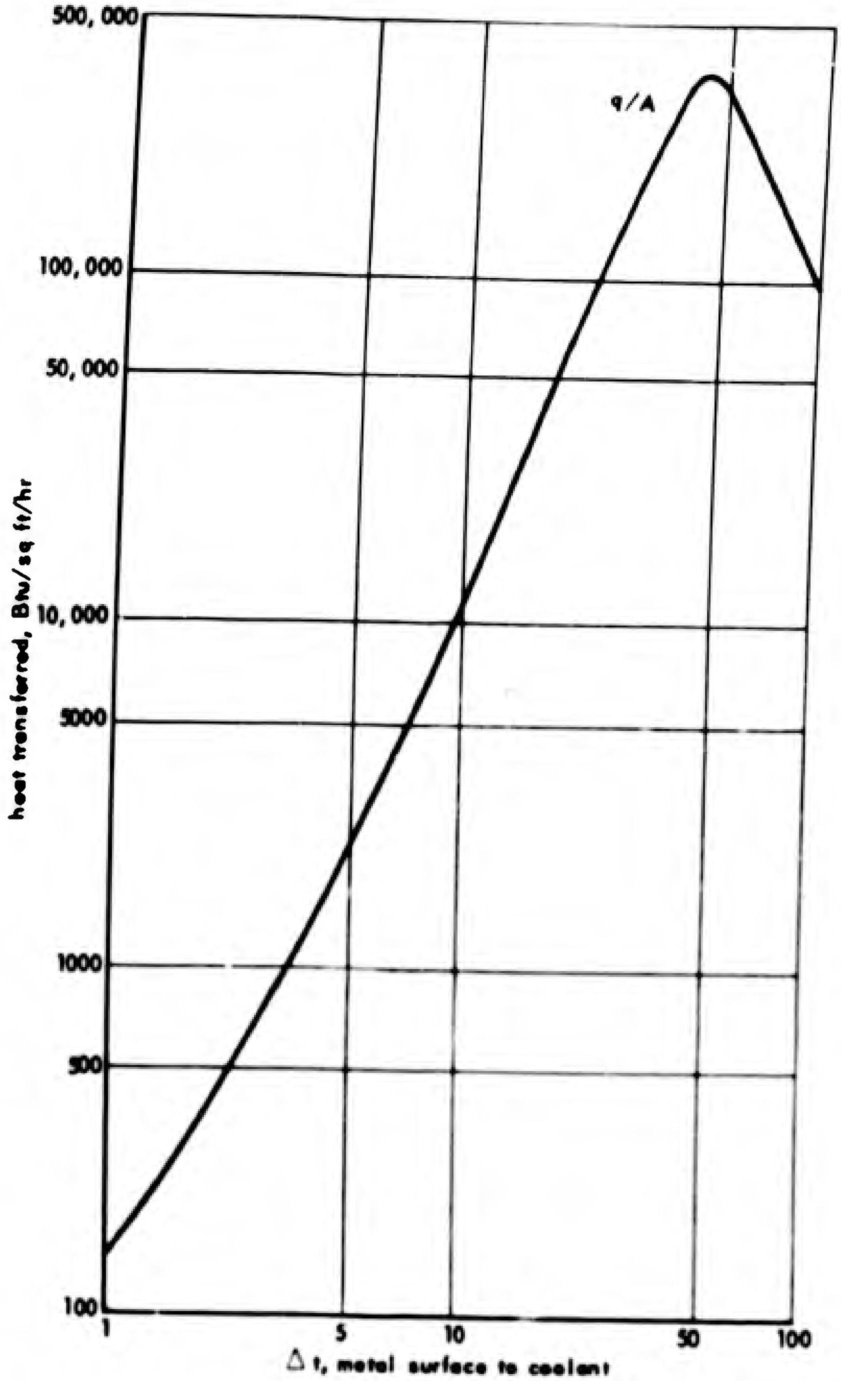


Figure 6. Graph* of water boiling at 1 atm outside a horizontal submerged tube.

* by W. H. McAdams, 1942, p296.

achieve the otherwise unattainable heat fluxes is accomplished as a result of steam bubble growth, detachment and release from the surface. Critical values of from 8800 to 10,000 for "h" for a variety of configurations are reported by the same source. For lower saturation temperatures (and pressures), critical rates are proportionately less and for higher pressures, higher. Further, the successive curves for higher pressures are displaced to the left, with result that for a given heat flux the temperature difference, saturation temperature to metal surface, is lower for higher pressures. Presumably, heat transfer would be accomplished at an extremely high heat flux at the critical temperature and pressure with practically no elevation of the metal wall temperature above the saturation temperature. Under these conditions, bubble size would reduce to an infinitesimal, and the liquid and steam specific volumes would be the same. In the various experimental equipment reported,[1] the maximum, or critical heat transfer was usually accomplished at the same approximate temperature, about 45 degrees above the saturation temperature.

In the 8-hp diesel engine used as an example above, if heat were rejected as steam at atmospheric pressure (or under a slight water head) at the same rate, 33,000 Btu per hr as projected, and at a uniform rate over the entire heating surface, the heat flux would be:

$$\frac{Q}{A} = \frac{33,000}{0.43} = 77,000 \text{ Btu/sq ft/hr} \qquad (4)$$

From Figure 6 the necessary Δt, metal to saturation temperature, would be about 12 degrees Fahrenheit, and h, the film coefficient of heat transfer $\frac{77,000}{12}$ = 6400 Btu/sq ft/hr/degree Fahrenheit.

Such phenomenal heat transfer is, of course, obtained with limitations. In transferring the 33,000 Btu per hour, neglecting radiation losses, some 34 lb of saturated steam is produced, and must be removed from the area of formation to prevent superheating the steam, with local heating to an undesirably high metal temperature. In applying cooling by boiling to a particular engine, the physical characteristics of the coolant passages become very important. Primarily, it is important that no area of potential high heat flux have an overhead pocket without a clear vertical egress to the next higher level and on into the cooling ducting. Discussing the three common arrangements of valve, head, and block, it can safely be predicted that the cast-as-a-unit block, typified by the small engine used in the example (see Figure 5), will probably be most easily and surely cooled by boiling. Without a separate head, the steam has a clear path with overhead pockets quite unlikely. Next, the overhead valve engine, most common in industrial diesel engines, should normally present no problems. Third, the "L" head engine, frequently found with comparatively small coolant passages between the block and head, and with many possible locations for overhead pocket, appears to require closest inspection. Experiences with all three types of blocks will be discussed later.

Quite aside from the desirability of cooling by boiling, or more exactly, cooling by boiling with the coolant at the local saturation temperature, the desirability of such a system from a heat reject standpoint will be considered. As has been shown, heat fluxes and satisfactory metal surface temperatures can probably be reliability achieved without high coolant velocities achieved through pumping. It is,however, necessary that coolant be available for replacement upon detachment of a bubble. Using an open vessel (shown in Figure 7),

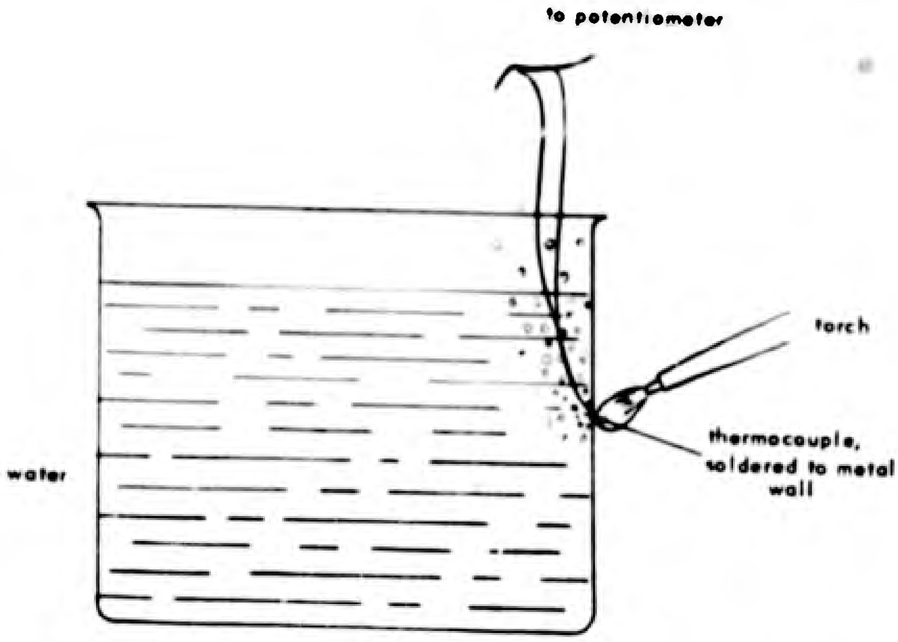


Figure 7. Sketch of simple apparatus for demonstrating cooling ability of a boiling liquid.

a simple experiment can be conducted with a minimum of equipment by soldering a fine wire thermocouple to the inside surface of one side, and playing a high temperature acetylene flame on the outside. Immediately upon application of the flame, local boiling will occur, and the temperature recorded for the inside surface will rise to about 218 to 222 degrees Fahrenheit, depending upon the location of the flame and its intensity. As the water in the container becomes heated, little if any change in the measured surface temperature can be detected. If we then cover the thermocouple area with an inverted cup-shaped shield to prevent the upward flow of steam, an immediate rise in temperature will be noted over the value recorded for that with unobstructed steam flow.

Depending upon the positioning and size of the shield, thickness of the parent metal, and its heat transfer characteristics, considerably higher temperatures than the original can be obtained. In severe cases it might be possible to restrict the escape of steam to the point where the metal could be damaged.

A still more spectacular and simple experiment can be made using a torch and an unwaxed paper cup of the type used for hot beverages. When filled with water and heated from the side, local boiling immediately occurs, and the outside layers of paper begin to char and sometimes burn. Successive layers of charred paper are removed by the flame velocity, but the tissue thin layer forming the inside surface stays intact, without discoloration. It is suggested that the reader avail himself of one or the other or both of the above simple experiments to obtain some personal feeling of the tremendous cooling capabilities of boiling liquids.

Heat Transfer

While mathematically very complicated,[4,5,6] the explanation for the phenomenal heat transfer possible with boiling is simply explained. Heat must be transmitted to an infinitesimally thin layer of water immediately adjacent to the metal. If not removed, this layer and adjacent layers become superheated with respect to the saturation temperature. Heat transfer through the quiescent layers adjacent to the heating surface is in accordance with the basic equation:

$$Q = \frac{k A \Delta t}{x}, \qquad (5)$$

where x is the material thickness, or in the case of transfer to liquids, the stagnant film thickness.

Equation (3) describes the situation in which the thickness of the stagnant film is decreased by turbulence related to the velocity of the moving stream. The major measure of the thin film thickness under forced convection conditions is the Reynolds number, which describes mathematically the turbulence or scrubbing condition. Another important case is that in which the expansion properties of the heated fluid and the change in density set up local currents, reducing the stagnant film thickness. The term fluid is used rather than liquid, since the mechanism is applicable to gases as well as true liquids. Chang[4] shows mathematically that boiling is a special case of natural convection heat transfer with phase change.

With high heat flux, particular points on the heating surface, because of some particular physical or chemical characteristic, become nuclei for steam bubble formation. This can be observed conveniently by heating water in a glass vessel. At the first evidence of boiling, steam bubbles will emanate rapidly from one or a few spots. Over the rest of the heated area, transfer from the superheated layer can be detected by convection currents. As the heating rate is increased or the bulk of the fluid being heated approaches the boiling point, the number of nuclei increases until steam bubble formation seems general. At very high rates, steam formation can be virtually continuous until the entire area becomes blanketed in an insulating layer of steam. If the heat flux is maintained, the situation proceeds unstably to burnout. Heating rates of this order can not be achieved by low temperature flames in the simple apparatus described above, unless special configurations are contrived to prevent water from reaching the heated surface after the steam bubble is released. The simplest example of steam insulation can be observed by heating a plane metal horizontal surface to a relatively high temperature, then dropping water on it in small quantities. Boiling of the drops is accomplished so rapidly that it seems to dance on the surface, and is intermittently suspended above the heating surface. If the surface is no longer heated, and the drops are periodically placed as the surface cools, the "life" of the drop will gradually decrease. At some point in the plate cooling process, the drops will reach the metal surface, spread, and rapidly evaporate. It should be pointed out that the typical stove or hot plate heating surface does not burn up for lack of evaporative cooling. It reaches some stable temperature through heat loss by convection and radiation. True burnout of the surface can only be achieved in experimental apparatus in which heat can be applied to a restricted area in large, controlled quantities, as to a small electrically heated wire or tube.

Referring the above thinking to an engine, it does not follow that local failure to destroy an engine part through faulty cooling is necessary for a cooling failure. Distortion of a part dependent for cooling on an adjacent part, for example, a valve, overheating of fuel or lubricating oil which comes into contact with a particular heated area, change in physical properties over a period of time with shortened life, among many, can result from a "hot spot", whatever the cause. Further, actual burning or melting in an engine is most unlikely, because the heavy sections of metal with high heat conductivity will relieve the situation in a trouble area. More likely, moving parts depending, for example, upon heat transfer during their dwell period to surrounding stationary parts, will probably become overheated if satisfactory mass block temperatures are not maintained. The most severe area and most likely to become "overheated" in this respect is the exhaust valve seat area. The highest velocities, highest mass flow rates in the engine of hot gases occur in the exhaust valve port. By nature its complicated shape is most likely to occasion unsatisfactory heat transfer. It is for this reason that in many engines of the "L" head engine design, the entire coolant supply from the radiator is distributed at high velocity through sheet metal nozzles aimed at the exhaust port area, coolant side.

Commercial Application

The determination of whether or not a given commercial engine, developed for cooling with water or an antifreeze mixture, can be satisfactorily cooled in the simplest of boiling-condensing systems (shown in Figure 8) is easily accomplished. If the temperature limitations of known trouble areas can be determined or deduced, the engine can be simply instrumented with a few thermocouples in that area and operated at the expected load, preferably with a controlled loading device such as a dynamometer or generator. By connecting the coolant lines to a variable elevation tank in the manner shown (see Figure 8), the limiting head "H", as well as steam production through evaporative loss, etc., can be determined directly.

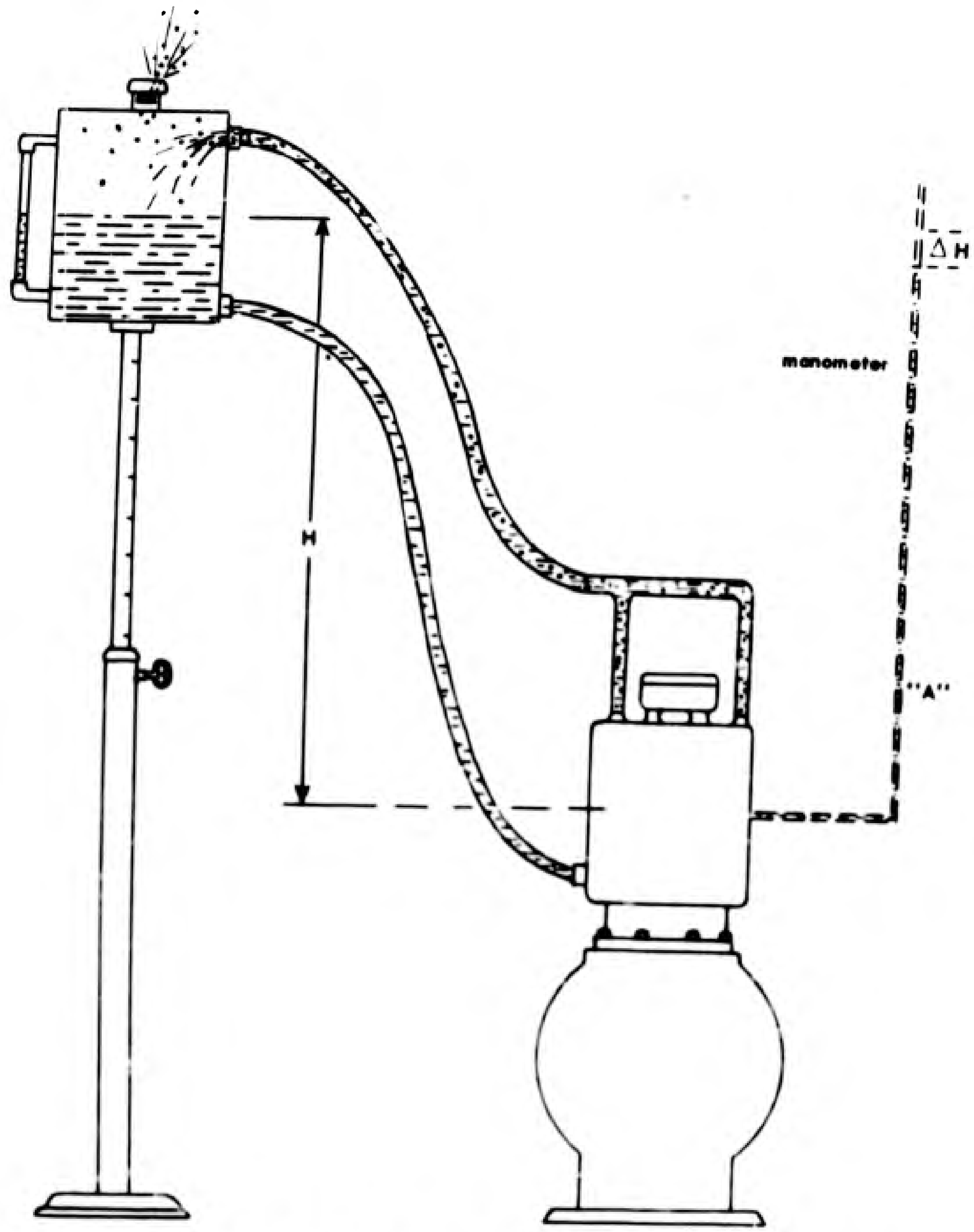


Figure 8. Schematic of simplest boiling-condensing cooling system with variable elevation separator.

If high velocity at certain points is necessary in conventionally cooled engines, it might be hypothesized that certain installations will require these or similar velocities at the same points. If not, it seems appropriate to remove the water distribution tube for the tests when the pump is removed, since it can only complicate the passage of steam and water. Certainly the engine with adequate passages between head and block, valves-in-head, and simplest coolant passages in all areas will present less restriction and chance for trouble. Diesel engines, usually of valve-in-head design, with low fuel-air ratios, high expansion ratios and lower exhaust gas temperatures, are generally recognized as having lower heat rejection than gasoline engines of similar power. They may be expected to present little if any difficulty since they are generally "well-cooled" in their original design. The valve-in-head configuration has a distinct advantage in that steam produced in the valve port area need not proceed through the coolant passages between the block and head.

For engines without collecting manifolds for the hot water, but with simple collection at the front of the engine at the thermostat housing, steam congestion near the engine outlet appears to be possible. This could be relieved by tapping the head immediately adjacent to the coolant passages between the head and block, particularly on the valve side. Fortuitously, the common valve-in-block "L" head engine on which this is most likely to occur usually has large horizontal areas available. This would give a direct unobstructed path for the steam formed in the block to proceed vertically without meandering horizontally through the head. In critical applications, and again probably most frequently in gasoline burning engines where "heating" is most likely to be severe, enlargement of existing coolant passages or placing of new, critically placed passages, appears to be desirable. Enlargement may be restricted by the openings in head gaskets which are frequently undersize. Provision of additional holes, except in production, would entail special head gaskets, or special care in fitting doctored gaskets. "O" rings in simple machined seats in the block or head would be particularly appropriate, and are used in some well cooled diesel engines.

Part III. EXPERIMENTATION AND EVALUATION

Chapter 3

DISCUSSION

In determining the merits of allowing an engine to boil as a practical cooling method, the advantages and problems in three areas will be discussed separately:

(1) Cooling of engine components per se.

(2) Disposal of waste heat external to the engine.

(3) Suitability of waste heat for auxiliary uses, e.g., preheating fuels, heating buildings, temperature conditioning of lubricant charge, etc.

Cooling of Engine Components

A great deal of experimentation and measurement has been done to determine the effect of temperature on parts and lubricants. Work was done by the Laboratory, in addition to that reported by other investigators, on a number of engines, including two comparable 2-stroke cycle diesel engines,[6] a 4-cylinder gasoline engine driving a 15-kw generator, a 1/2-ton truck "Jeep" engine, a 4-stroke cycle diesel engine powering a crawler tractor, and several identical 1-cylinder, 4-stroke cycle diesel engines driving centrifugal pumps or 2-1/2 kw generator sets. The best instrumentation and most closely controlled runs were in the latter units, and results on these will be discussed in detail. This engine is that discussed in the hypothetical heat transfer examples above. It is a heavy duty diesel engine of 21.5 to 1 compression ratio, 3-1/2 in. by 3-5/8 in. bore and stroke, 1800 rpm (driving a 110 volt single phase generator). It features forced lubrication and high pressure solid injection of fuel in the Bosch type system, and a cylindrical precombustion chamber. The water-cooled cylinder (see Figure 5) is cast in one piece from iron, and has no removable head. The piston assembly is inserted from below before the block is assembled to the cast aluminum crankcase.

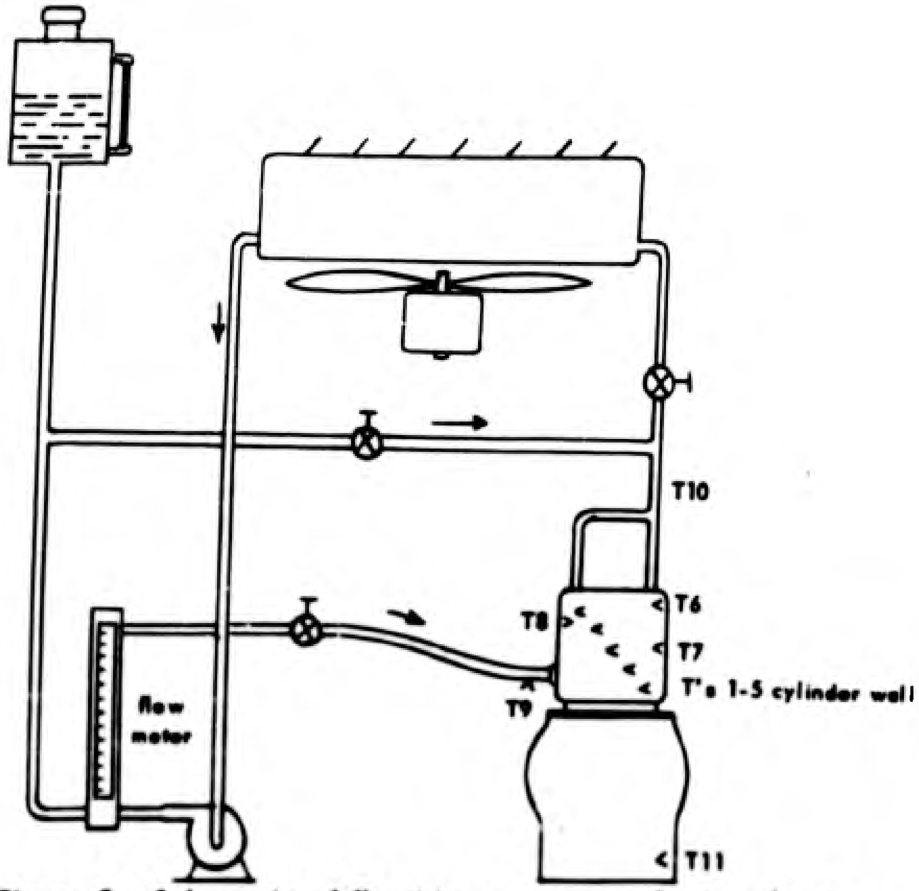


Figure 9. Schematic of flexible experimental setup showing thermocouple locations.

During the controlled tests, the unit was equipped with eleven 22-gage copper-constantan twisted (shown in Figure 9), soldered thermocouples. These were imbedded in locations as follows:

One through five were imbedded in the cylinder walls with the junctions approximately 1/8 in. below the surface in 1/4-in. wall. They were spaced at 3/4-in. intervals vertically from bottom to top, as follows: Thermocouple five was at the location of top ring travel at top dead center,

Six was in the exhaust valve seat area approximately 5/16 in. from the valve face,

Seven in the coolant, lower block,

Eight in the coolant, middle block,

Nine in the coolant, in duct to bottom of block (normally coolant into engine), Ten in the coolant, in duct from top of block after connecting "T" (normally coolant out of engine),

Eleven in the bulk crankcase lubricant.

These temperatures were read using a calibrated sensitive potentiometer calibrated to 1 degree Fahrenheit and readable to 1/2 degree. This instrument proved particularly valuable in the course of the experiment in that fluctuations and rapid rates of changes could be continuously sensed by the recorder in balancing the potentiometer bridge. A sense of pinpoint temperature control by boiling-cooling was obtained, in contrast with continual variation with forced convection and low coolant temperatures in those tests with the coolant well away from the boiling point. Similar variations have been observed elsewhere and recorded on fast response electronic recording oscillographs.

A series of tests at varying loads, coolant outlet temperatures, coolants and coolant velocities were run. Normal flow was considered to be the usual case, wherein the coolant passed upward by the cylinder and out of the top of the engine. One series was run with forced reversed flow. Two coolant mixtures were used, fresh tap water, and a mixture of 60 percent ethylene glycol and 40 percent tap water. The engine was cooled with each coolant at conventional Port Hueneme temperatures, about 70 degrees Fahrenheit and in the cold chamber at -25 degrees Fahrenheit. Values from these tests were correlated with similar readings taken from identical units at Fort Churchill, Manitoba, Canada, at sub-zero temperatures, and at Yuma, Arizona, at high temperatures. In both cases the field units were exposed to the elements of wind, sun, and snow. Runs were made with the engine boiling water at 70 degrees Fahrenheit ambient, varying the height of the steam separator. One series of runs was made using gravity or "thermosyphon" flow, the normal arrangement in the commercial water-cooled generator unit. (See Figure 9 for the complete apparatus arranged for forced convection and Figure 8 for the setup on boiling runs).

Of the several prototype installations made (see Figures 2A, 3D, and 5), the greatest amount of data collected under simulated field conditions was taken on the unit reported,[6] a 30-kw diesel driven generator set (see Figures 2A, 2B, and 2C). The unit was operated successfully at 140 degrees Fahrenheit in a hot room, 95 to 110 degrees Fahrenheit in the sun at Yuma, in the cold room at -65 degrees Fahrenheit, and for a short time at Fort Churchill, and was exposed to snow, wind, and ambient temperatures as low as -35 degrees Fahrenheit. The pump used to circulate coolant during a portion of the hot room tests was later removed and thermal circulation employed. At all except low temperatures, it was operated successfully on standard heavy distillate diesel rather than premium fuels recommended by the manufacturer. The clear condition of the exhaust during the hot room tests is illustrated by Figure 10. The same excellent combustion was typical of low temperature operation. The stabilized oil temperatures at full load are shown in Figure 11. Typical cylinder temperatures[6] are shown in Figure 12. It is worthy of comment that the original radiator of the generator was clearly shown to be approximately 3-times larger than necessary when used as a condenser at 140 degrees Fahrenheit, full load, in the hot room tests. At -65 degrees Fahrenheit, only about 1-1/2 in. of the radiator (at the top) was functioning as a condenser. This type of 2-stroke cycle diesel engine can not normally be operated satisfactorily at very low temperatures without excessive smoking, because of the excessive cooling of the central portion of the cylinder by the incoming air charge. Even while boiling, there was evidence of mild overcooling in the intake port area.

Several installations were made on 1/4-ton truck engines, 4-cylinder "Jeeps", (see Figures 3, 3A, and 3B). The installation shown in the three figures was used extensively in the Port Hueneme area for general runabout service without a fan for 6 months. Under these conditions, the engine could be idled indefinitely, without loss of steam, boiling consistently providing it was pointed into the prevailing wind. Still without a fan, it was shipped to Fort Churchill for simulated field work as a runabout. Here again, no evidence of failure to condense was observed except when the unit was parked with the condenser not faced into the wind. It was idled for extended periods and driven at high speeds on the grades available. During 5 weeks it was operated approximately 1700 miles by NAVCERELAB and Army personnel. It was fitted with an engine preheater which also preheated the oil. With its 6-volt starting system it could not be started without preheating below about zero degree Fahrenheit. The oil, roughly equivalent to an SAE 5 W grade, MIL-O-10295, was not changed during the 1700 miles of operation under these most severe conditions. Some 4 quarts were added, for although the engine did not normally use appreciable amounts of normal viscosity oil, it did use some of the more volatile product. The engine coolant and oil temperatures over a 2-day period of intermittent operation,

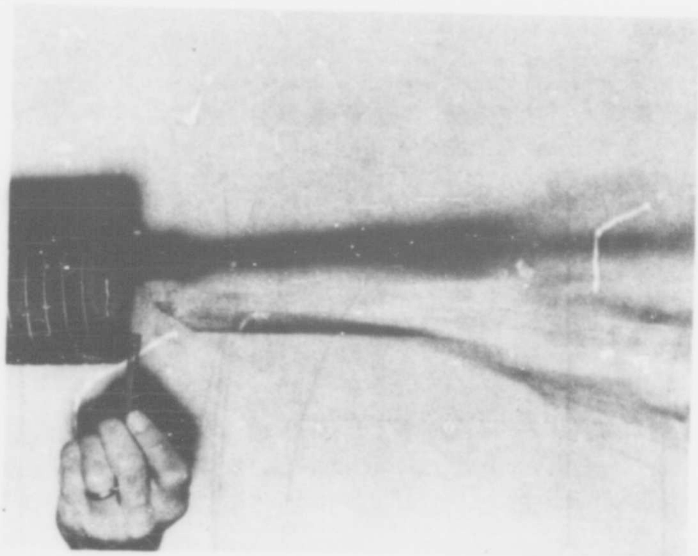


Figure 10. Exhaust from diesel generator at high ambient temperature. Engine cooled by boiling.

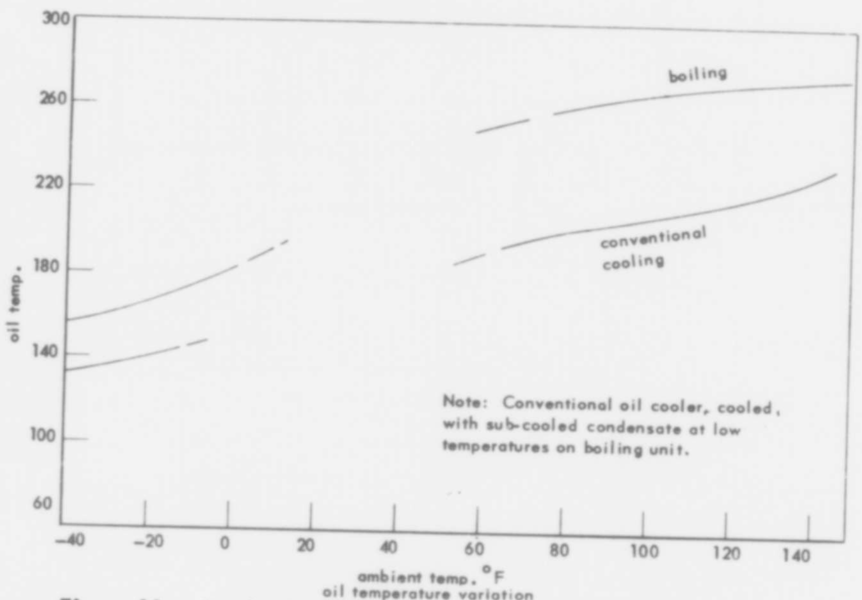


Figure 11. Graph of collected data on high- and low-ambient temperature tests.

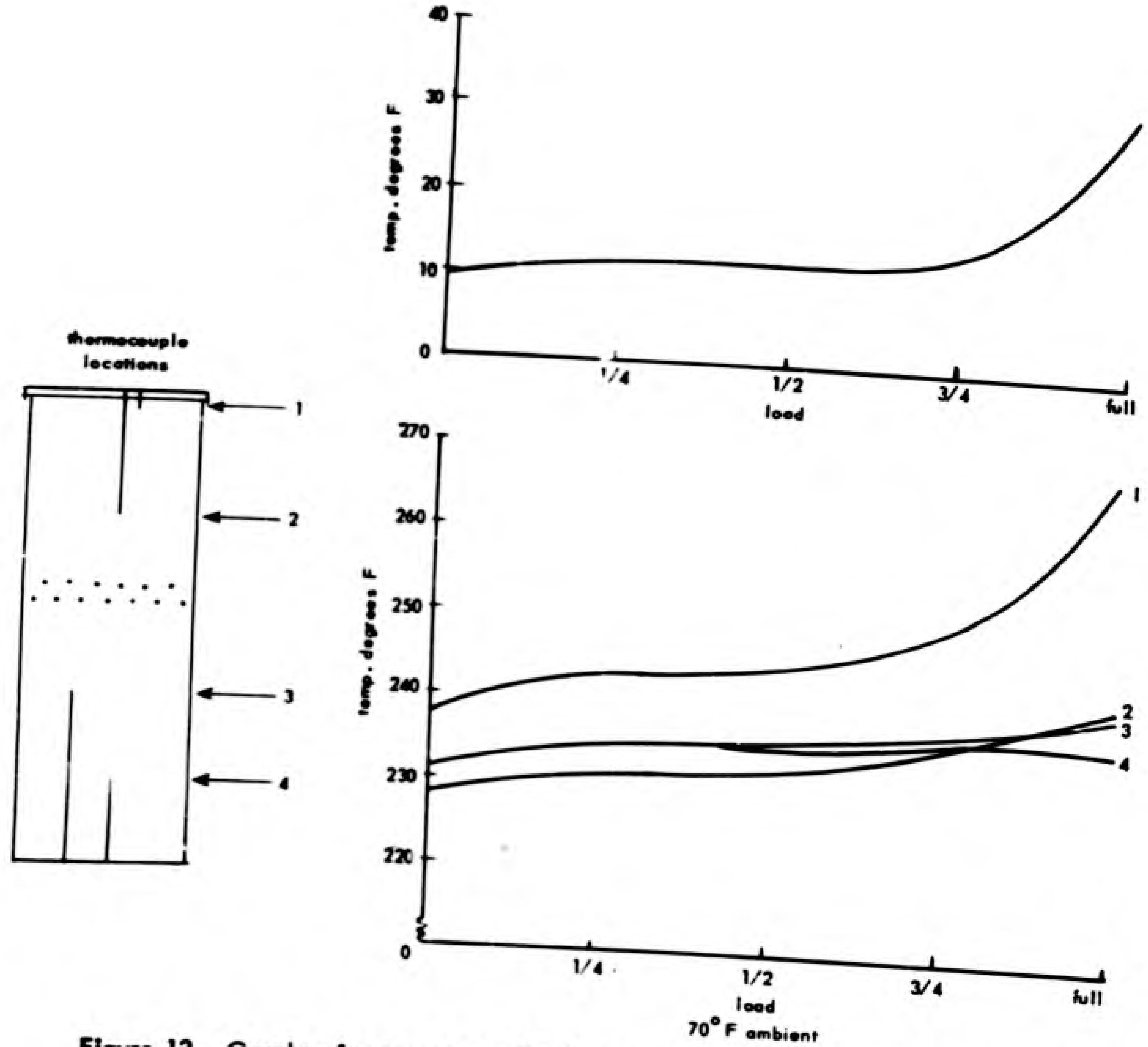


Figure 12. Graphs of temperature distribution along cylinder wall of 2-stroke cycle diesel engine with boiling-condensing. Diagram shows thermocouple locations.

cooling, and preheating are shown in Figure 13. The excellent condition of the engine and crankcase are shown in Figures 13A, 13B, and 13C. There was no evidence of scoring, significant wear, sludging or corrosion. Except for the small amount of sediment (shown in Figure 13D), the drained engine underside had the general appearance of having been steam cleaned and wiped with clean oil. All rings were entirely free. At this time the engine had a total of some 23,000 miles of light duty operation of which 7500 miles had been with boiling-cooling. This unit, with fan (see Figure 3) was later operated in and around Yuma by station personnel for one summer. During this period some difficulty was experienced with steam loss from the prototype condenser, but not while the condenser was in good repair. The same unit with former prototypes was operated to and from the Sierra test site of the Laboratory at an elevation of about 8000 feet, across the desert, hauling heavily loaded cargo trailers, etc. One system used in winter mountain operation is shown schematically, Figure 13D.

Applications virtually identical to that of Figure 3 were made on the same engines powering 3-phase 15 kw generators at 1800 revolutions per minute. Steam production appeared to be excessive in this application, with the result that pressure usually built up in the condenser, necessitating shutdown. One unit could satisfactorily be operated for short periods until the collection of noncondensable vapors in the condenser caused pressure build-up, again requiring shutdown. This particular type of "L" head engine, spark ignited, appears to be the most critical and questionable application. While it is believed that the installation could be made to be satisfactory changing the block to head coolant passages, providing more frequent steam outlets from the head than the single thermostat opening, etc., this was not done.

The installation made on a large diesel tractor (see Figures 3C and 3D) gave excellent results at high and low temperatures. At 110 degrees Fahrenheit, with a steady, maximum drawbar load, the condenser constructed proved slightly undersize, with a pressure build-up over a 2-hr period. Performance of this unit at very low sub-zero temperature was particularly spectacular. Combustion was so clean at all loads that operation of the engine could not be detected by observation of the exhaust stack, to as low as -57 degrees Fahrenheit.

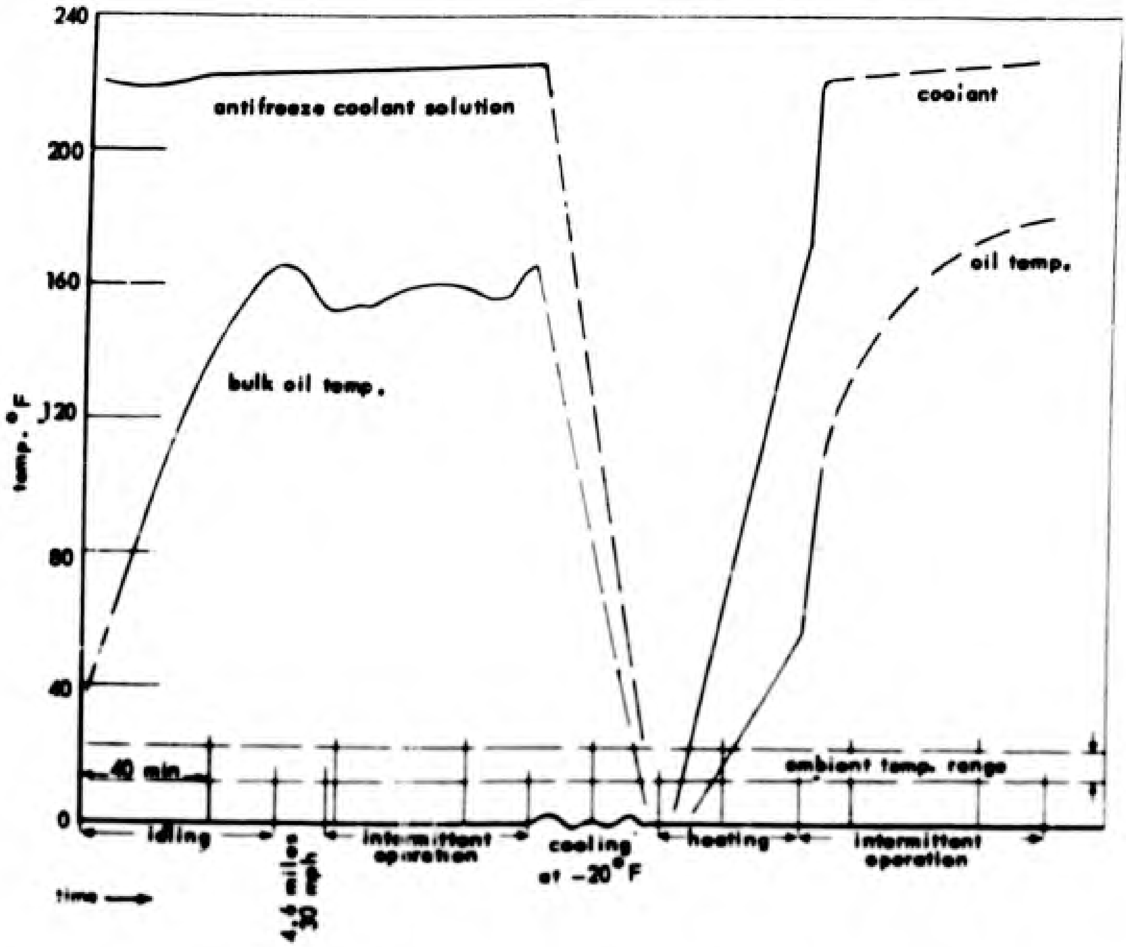


Figure 13. Graph of engine coolant and oil temperatures, "Jeep" installation used in sub-arctic.

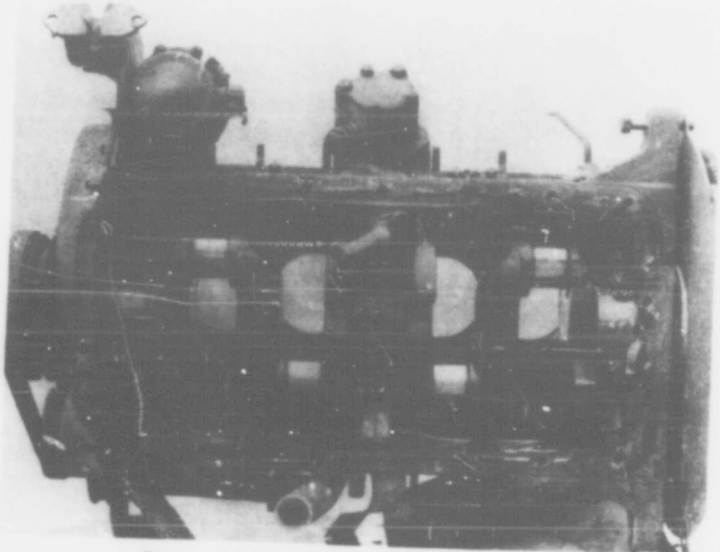


Figure 13A. Condition of engine.

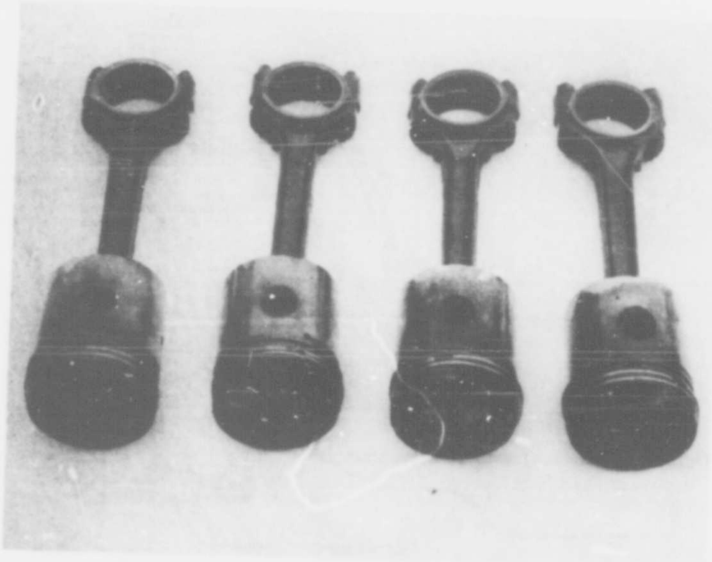


Figure 13B. Condition of piston assembly.

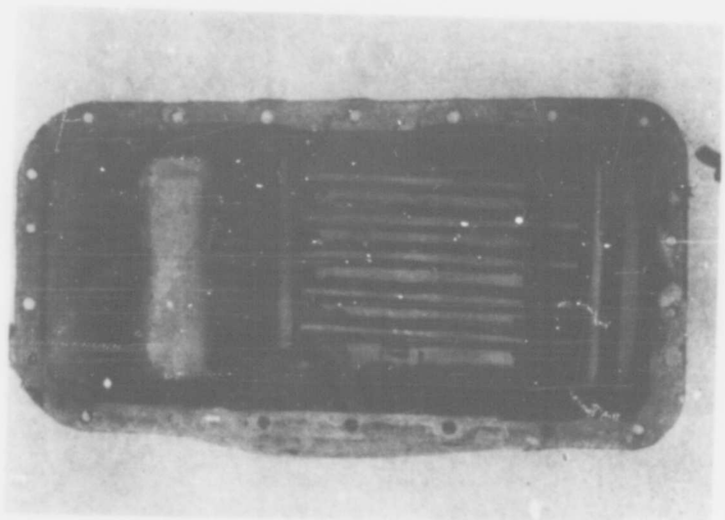


Figure 13C. Condition of oil pan.

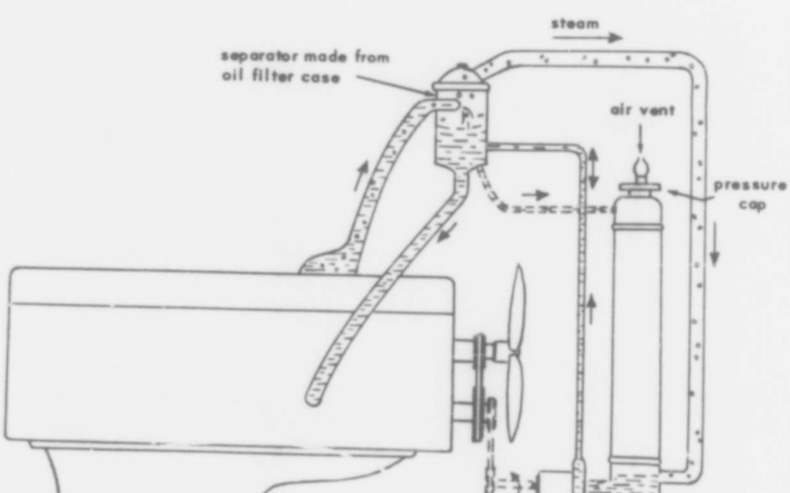


Figure 13D. Schematic of radiator condensation system.

Several installations were made on single cylinder diesels identical to that discussed at length (see Figure 8) driving pumps, generators, etc. Observed temperatures at all ambient temperatures agreed well with all the data presented. Variations in oil temperatures with ambient temperatures were in excellent agreement with laboratory test values. The usual installation was equivalent to that of Figure 3C, except no provision was made for heating or cooling the oil selectively toward or near the boiling point of the coolant. Generator sets equipped to boil used fresh water as coolant at high ambient temperatures. Antifreeze solutions, 60 percent ethylene glycol, 40 percent water, were used at below freezing temperatures. The first units caused some difficulty from loss of steam from leaks caused by vibration. Later units used at Fort Churchill had the condenser isolated, and gave excellent results. Two different tests were made at the Canadian installation in successive winters of approximately 500-hr duration each, with programmed loading from no load to full load in each case. The first year the engine was lubricated with MIL-O-10295, with severe scoring and wear (shown in Figure 14). It should be noted that the bulk crankcase oil temperature for this and the subsequent test was the same within experimental limits with variations in temperature in the range from -20 to -40 degrees Fahrenheit. The following year the same unit, rebuilt, was subjected to the same conditions but lubricated with Navy symbol 9110 oil, a detergent diesel oil with an equivalent viscosity to SAE 10 weight. In both cases the engine was started by preheating the coolant electrically. The wear and scoring for the second test is shown in Figure 14A. Comparative measured wear rates for the two tests are given in the following table:

Single Cylinder Engine Wear Tests Boiling-Condensing Cooling Programmed Variable Loading 500 hr No Load to Full Load		
Part	Lubricant MIL-0-10295 WEAR, in.	Lubricant, Navy Symbol 9110 WEAR, in.
Rocker arm shaft	0.0004	0.0001
Cylinder, top	0.002	0.002
Crankpin journal	0.0005 to 0.001	0.0002 to 0.0000
Piston ring land, all rings	0.0005	0.0005
Note: This is roughly equivalent to 13,000 miles at 25 mph, 1600 revolutions per minute (equivalent cylinder wear, 0.00014/1000 miles).		

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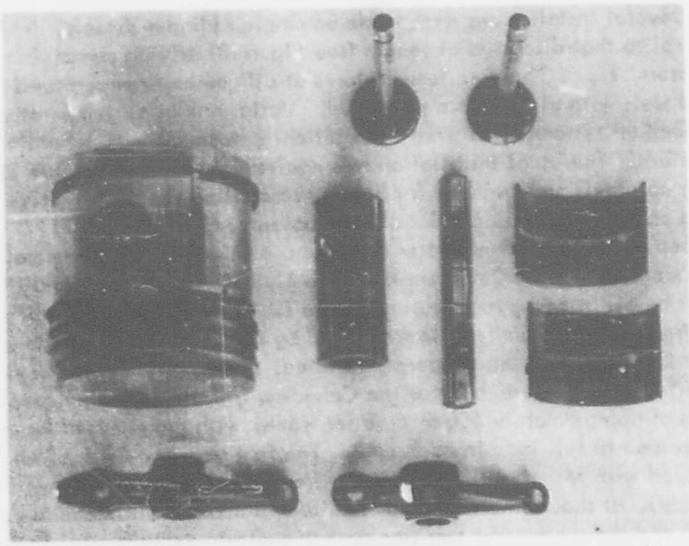


Figure 14. Piston assembly parts showing severe scoring and wear (lubrication with MIL-O-10295 oil).

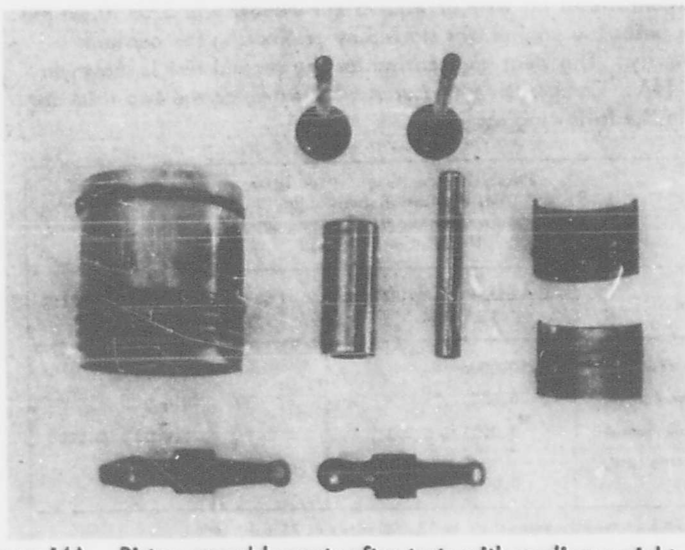


Figure 14A. Piston assembly parts after tests with medium weight oil (lubricant Navy symbol 9110 oil, SAE10).

While it is outside the purpose of this report to discuss lubrication, this wear is intermediate, i.e., somewhat higher than predicted and determined[7] for comparable cylinder temperatures. It is concluded that the viscosity of both lubricants is low for optimum lubrication, and that the recommendation of the manufacturer of SAE 30 weight at higher ambients might well have been used, inasmuch as preheating was necessary to start the engine with the lighter oils.

No difficulties from overheating occurred that could be traced to the boiling-condensing system except in connection with the 4-cylinder engines driving generator sets, discussed above.

Typical temperatures for forced convection runs with ethylene glycol mixtures and water, normal flow, are shown in Figure 15, with coolant in thermocouple 9 at 188 degrees Fahrenheit ± 2 degrees. A pronounced convergence in thermocouples 1 to 5 along the cylinder wall with the ethylene glycol solution is less noticeable with water. Values obtaine n boiling tests under similar loading were plotted at 25 gal per hour, flow was not measured and these were spotted only for comparison. Marked divergence with the ethylene glycol solution taken at an ambient of -25 degrees Fahrenheit is not explained. Variation from this trend will be seen later. Changes in length of hoses, pipe fittings, etc., were found to cause major shifts since they restricted the normal gravity flow. Figure 16 shows similar data for water introduced at 150 degrees Fahrenheit again at full load. The same gathering of points, i.e., lessening of the temperature difference along the cylinder wall is noticeable with both water and ethylene glycol solution, with decreased flow rates. Again, it is most noticeable with ethylene glycol solution as is the spreading during boiling with that coolant. Figure 17 shows similar data with 118 degrees Fahrenheit coolant introduced at full load. The tendency for the temperature spread along the cylinder wall to increase at high flow rates is less pronounced. This tendency to overcooling at high flow rates is even more noticeable in the first set of data for reversed flow of coolant at 118 degrees Fahrenheit inlet temperature (shown in Figure 18) in that even the highest cylinder temperatures are below the minimum 194 degrees Fahrenheit temperature desirable[7] established by the literature and corroborated in general. On the other hand, this reverse flow system, which would appear to have merit for extremely high output, critically cooled gasoline engines, yields good to excellent temperature spreads along the cylinder wall ΔT, exceeding in excellence only by boiling under optimum conditions.

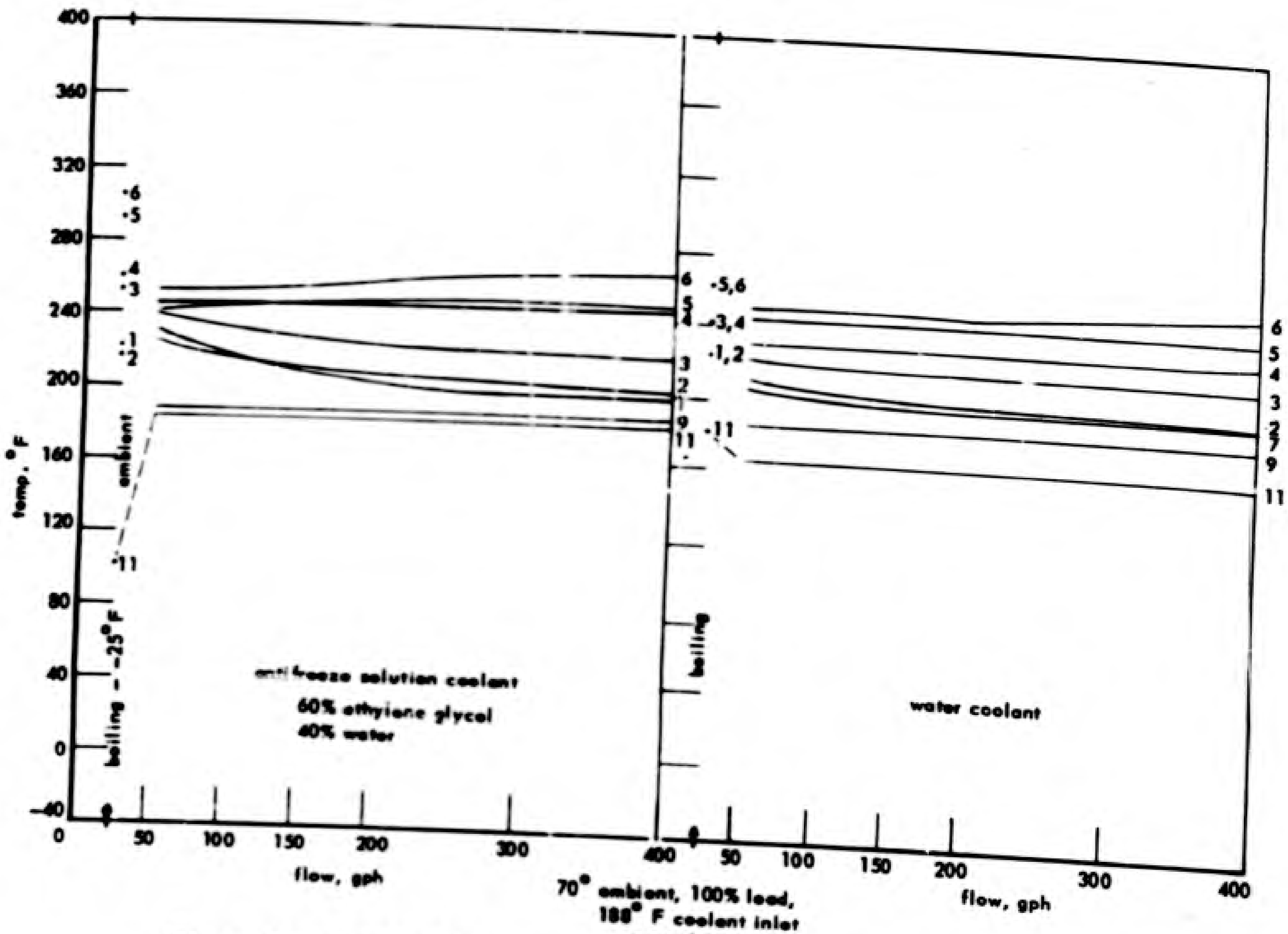


Figure 15. Graphs of typical engine temperatures for forced convection.

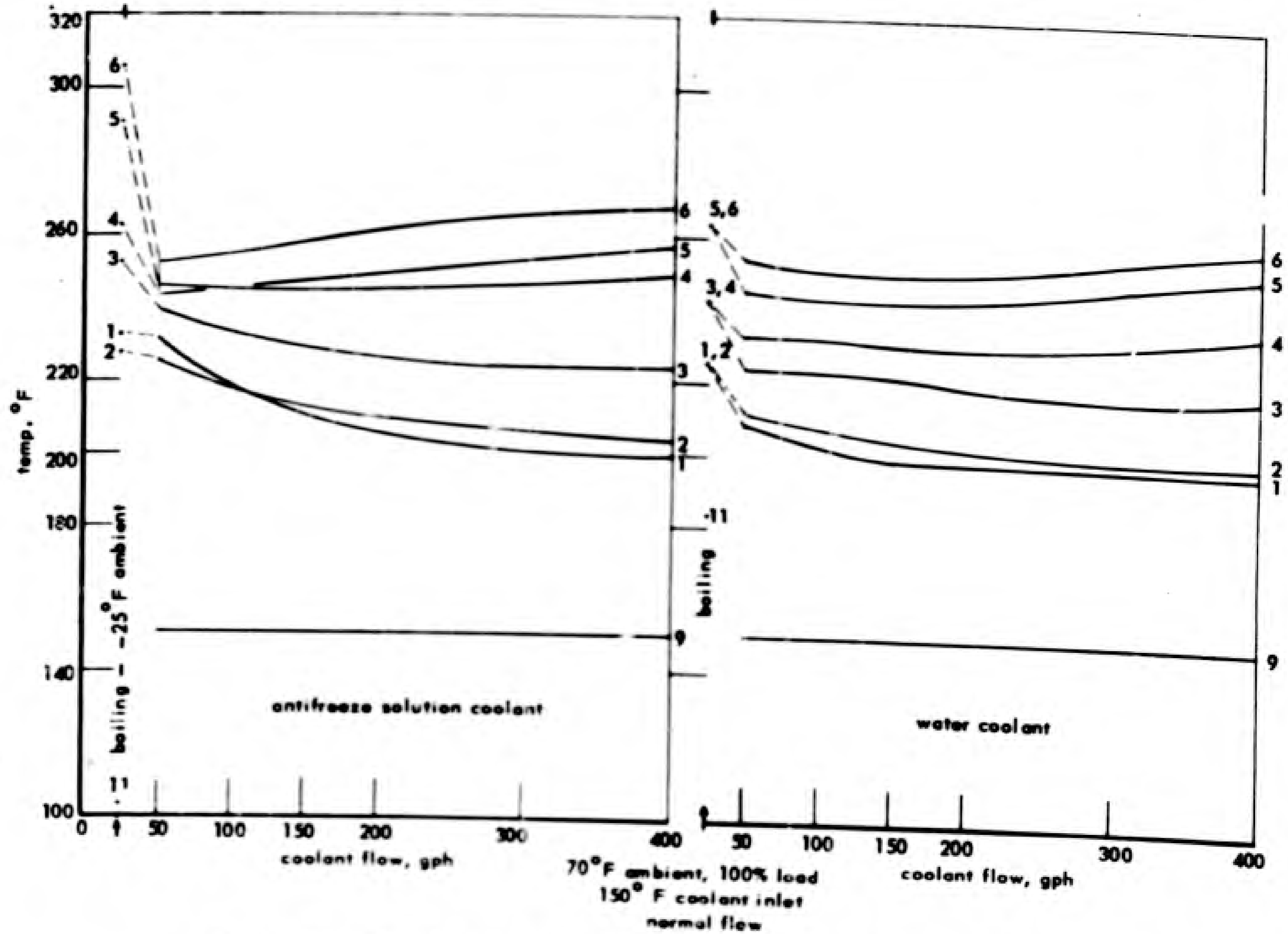


Figure 16. Graphs of typical engine temperatures for forced convection.

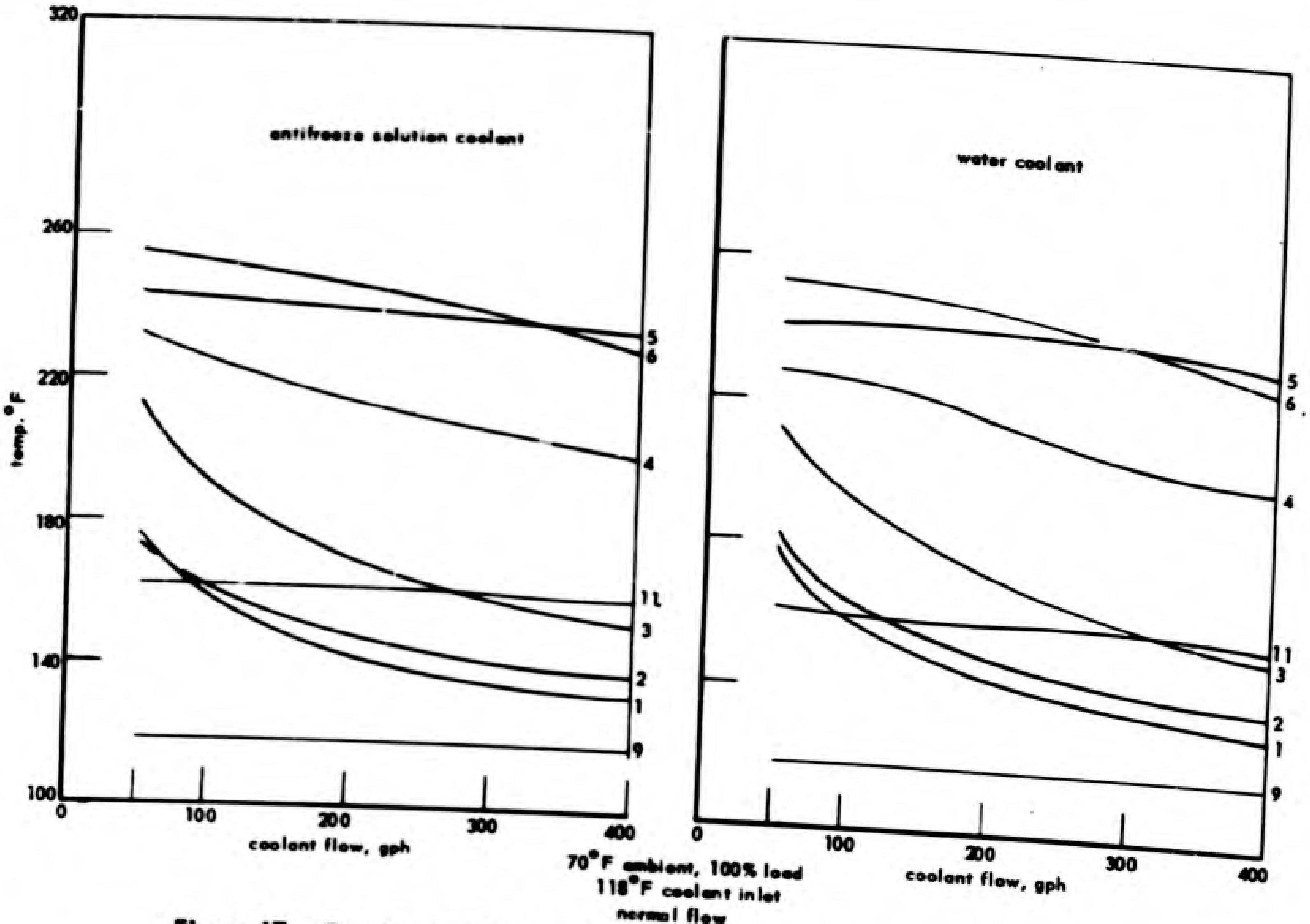


Figure 17. Graphs of typical engine temperatures for forced convection.

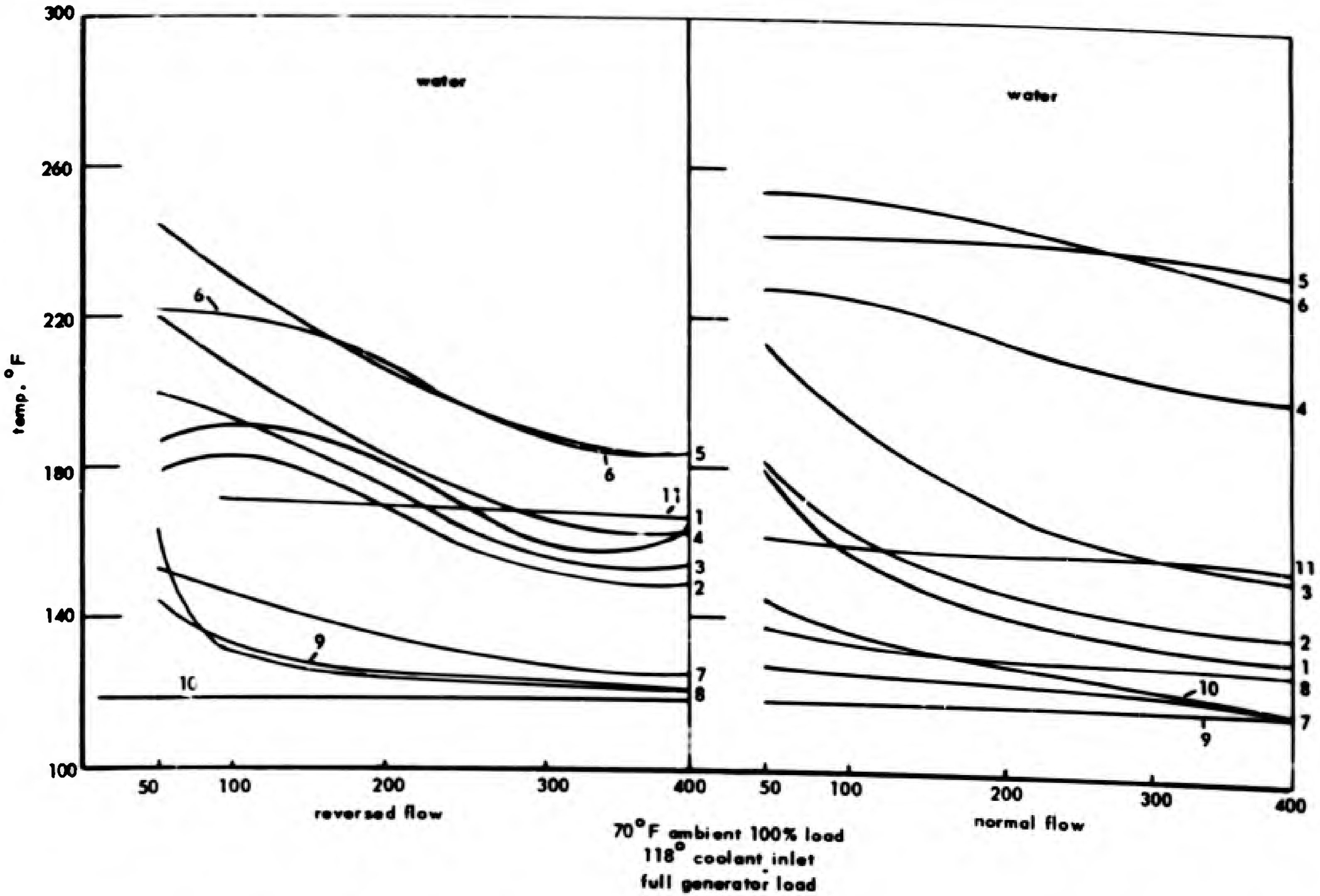


Figure 18. Graphs of typical engine temperatures for forced convection at high flow rates.

The data presented in Figures 15 through 18, inclusive, are for full generator load with varying flow rates, temperature of inlet of the coolant and flow rate. With coolant above 150 degrees Fahrenheit, inlet, satisfactory cylinder wall temperatures in terms of the previously selected 200 degree Fahrenheit minimum, are obtained. At lower coolant temperatures (see Figure 17) and lower loads (shown in Figure 19), however, severe overcooling is the rule. For brevity, complete

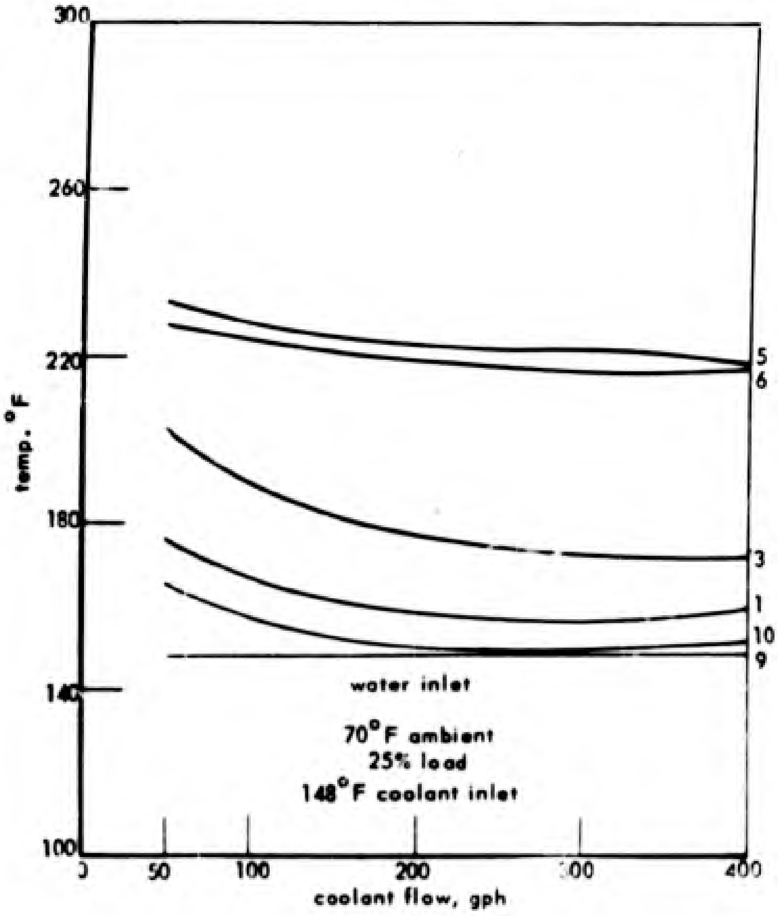


Figure 19. Graph of typical engine temperatures for forced convection at low coolant temperatures and low loads.

comparisons at full load only are shown, but the pattern is complete for light loads and coolant temperatures with forced circulation, at both normal (70 degrees Fahrenheit) and low (-25 degrees Fahrenheit) ambient temperatures. Restated, overcooling appears to be the rule except at optimum loads, coolant temperatures, and low flow rates. The remainder of the paper is devoted largely to determining the probability of better performance and the possibility of overheating with high temperature coolants.

A comparison of temperature spreads, ΔT, for a variety of conditions at full and 1/4 generator load are shown in Figure 20. Without deliberating the individual curves at length, it will be recognized that ethylene glycol gives less satisfactory curves from an academic standpoint than water, that high temperature water yields greater ΔT's in reverse, lower in conventional flow. At high temperatures (compared with sub-zero), thermosyphon or gravity flow gives desirable cylinder wall temperatures and good ΔT's. Unrestricted cooling of the heat exchanger (radiator) will obviously lead to extremes along the cylinder wall, from severe subcooling of the coolant.

From the data presented in Figures 15 through 20, inclusive, it will be seen that, within the limits investigated:

1. High flow rates (as recommended in handbooks) lead to excessive cooling of certain portions of the cylinder walls at either low coolant temperatures or off-loads.

2. When high flow rates are accompanied by low inlet temperatures, severe overcooling of the cylinder walls will occur, in terms of recognized criteria.

3. Optimum or minimum ΔT's were obtained allowing the engine to boil.

4. In no case in the boiling of an engine were any cylinder wall temperatures below the lower limit of 194 degrees Fahrenheit established in the literature.

5. Values of ΔT obtained during boiling usually fall quite closely on an extrapolated line for high temperature coolant flow at low flow rates with the same apparatus, whether under forced pump circulation or thermosyphon.

This is an agreement with reasoning and the literature[4] which mathematically demonstrates boiling heat transfer to be a special case of natural convection and correlates the two with a wave theory. It further demonstrates that overheating of a surface cooled by boiling, with "steam binding" of the surface, is actually tantamount to cooling the surface with a gas, in this case steam, and the surface temperatures obtained can be predicted on that basis.

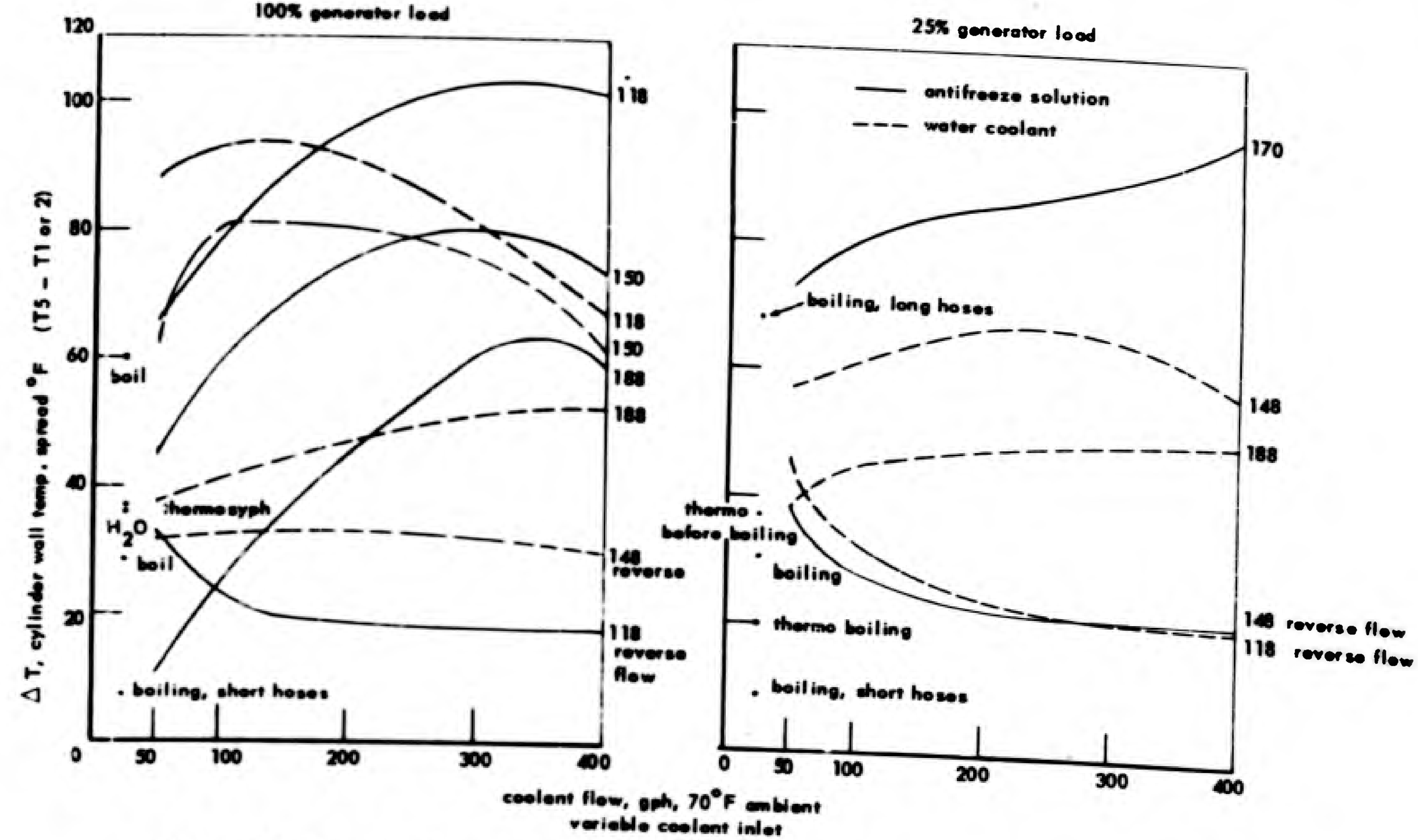


Figure 20. Graphs of typical engine temperature spread for forced convection.

The simple apparatus used in these preliminary investigations was ideally suited to some exploratory excursions varying obvious physical parameters in a simple boiling system. The equipment was further simplified to include only the engine with "T" in the upper manifold, new 5/8 in. internal diameter hoses, and a variable height stand for the small tank with gauge glass used as a separator (see Figure 8). The cap was drilled in a position that would allow the plume of steam to escape, after having made an abrupt turn to separate the water. The cap hole was not in line with the steam-water stream. The escaping steam was assumed to be wet, saturated at local pressure.

Two principal series were run:

1. With hoses sufficiently long to allow vertical adjustment of the steam separating tank height. This series was run with varying load, varying head, engine boiling, with water as the coolant (shown in Figure 21).

2. Extended (2 hr or more) runs were made with the separator at the height of the original tests, with a 28-in. net head, (shown in Figure 22). This corresponds within 1 in. to the second from the lowest position in the variable head runs.

In the variable head tests (see Figure 21), the cylinder wall temperatures did not vary greatly from no load to 125 percent nominal generator load, nor was there considerable variation with change of separator height. There was, however, a surprising variation between the slopes of the plotted values for thermocouple 5, at the top of the ring travel for no load and overload, "E" and "A". ΔT approached a maximum at an intermediate load with the long hose lines. At "F" the ΔT curves for both the variable head and the later short hose, extended runs are given. The data are replotted at "G", against load. The shape of the lower curve, marked "28 in. net head, short hoses" is more as might be expected for a good design. Here, there is a slight increase in ΔT as the engine heat transfer load increases. At intermediate loads, the flow is improved as some near-optimum point of water-steam bubble mixture is approached and temperature at thermocouple 1 rises with increased heat transfer and coolant, Figure 21, "G", point "X". At high loads the flow is even better, insuring scouring of the hotter surfaces, thermocouples 5 and 6. The higher velocity is felt at the location of thermocouple 1, with an increase in metal temperature at the point "Y", (see Figure 22, "E").

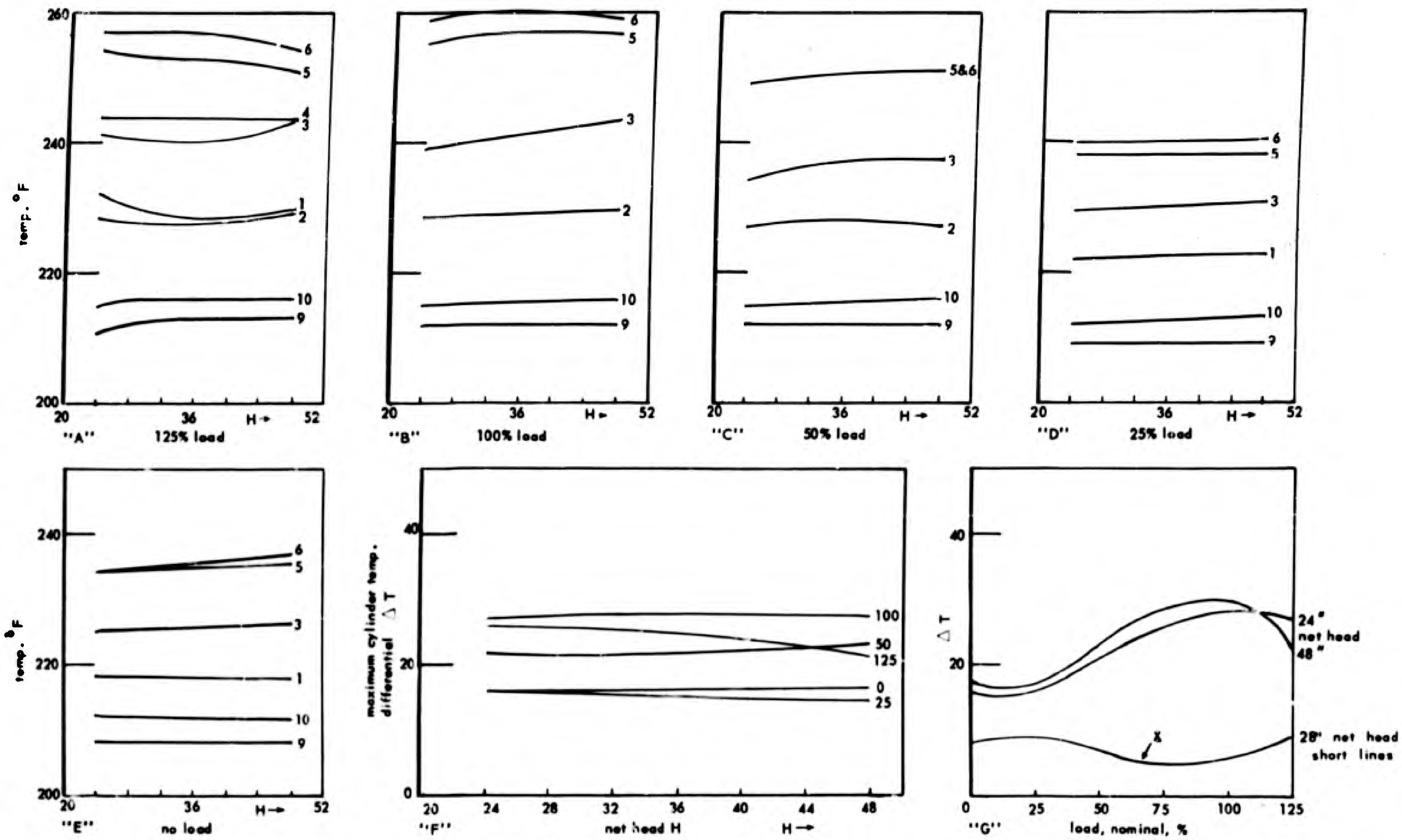


Figure 21. Graphs of typical engine temperatures at varying loads and head, with engine coolant boiling, water coolant.

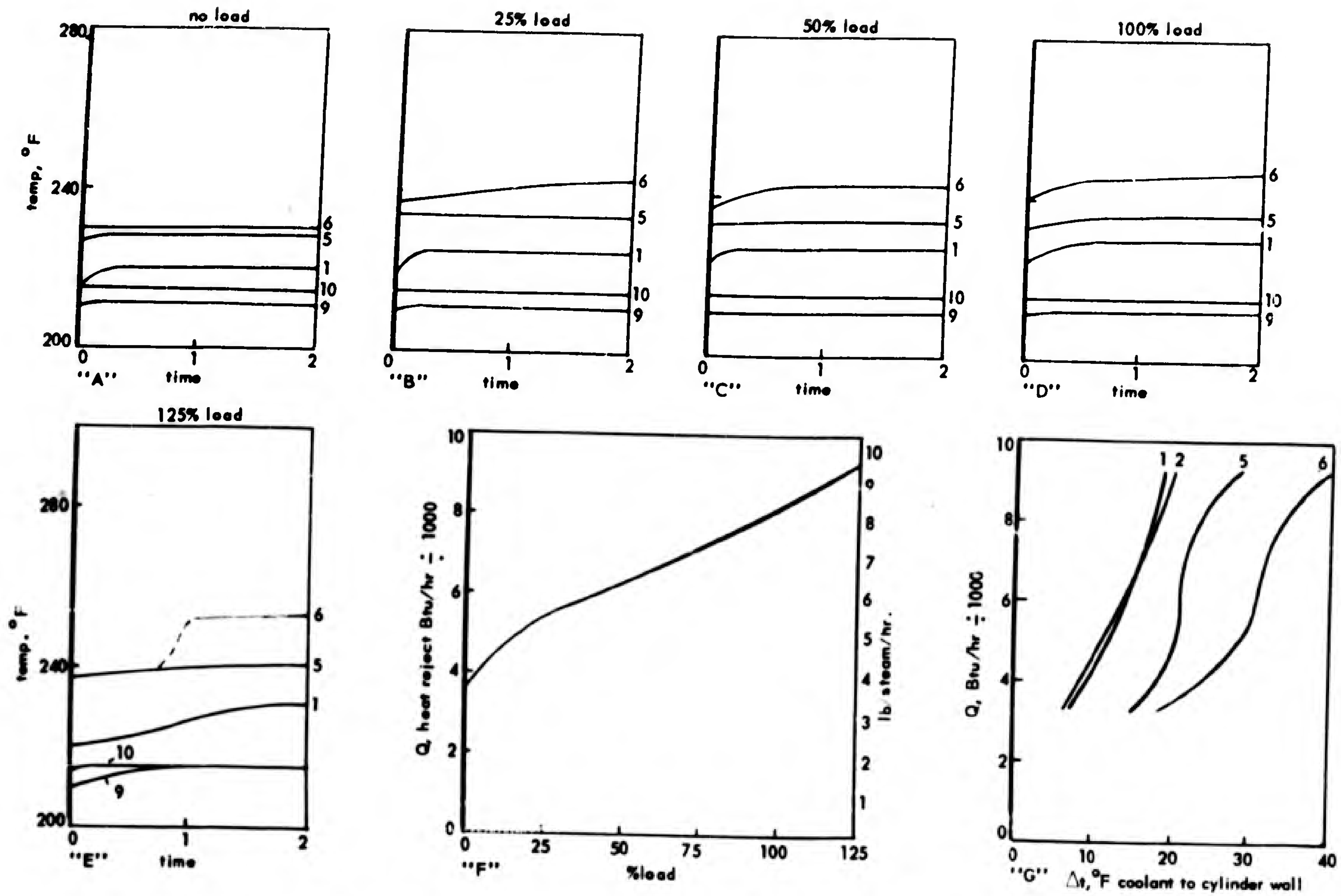


Figure 22. Graphs of typical temperatures with 28-in. net head, engine boiling, variable load.

At "F" (see Figure 22), values of heat rejected to form steam are plotted against load. The error in the measurements is less than the width of the plotted line, and results are gratifyingly linear except at very low load. At "G" these values of heat reject are plotted against values of Δt, where the surface temperature is assumed to be that measured in the metal. Theory and a large mass of experimental data predict that with boiling, "Q" will not be linear with Δt. The spectacular results obtained depend, in fact, on this nonlinearity. The near linearity of thermocouples 1 and 2 as compared with thermocouples 5 and 6, indicate that most of the heat transfer is accomplished on the upper cylinder walls as would be expected. This may be important in locating the water return to the engine and will be discussed under "applications" below.

Figures 15 through 20, inclusive, with wide and vagrant variations in cylinder wall temperature spreads and heat reject with varying flow rates offer sharp contrast with the data of Figures 21 and 22, with boiling water. In the former case of mixed forced convection and local boiling, typical of current practice, both wall cooling and heat reject are seen to be nonuniform functions of several variables. With boiling, on the other hand, they are dependent only on the physical characteristics of the coolant and the load, so long as the engine flow pattern allows free egress of the steam and rapid replacement of steam bubbles with coolant.

The heat rejected to boiling water with varying load is again plotted in Figure 23, and various minimum values, at whatever flow rate they may have occurred from the other forced convection runs, are plotted for comparison. The nature of the physical processes involved at local points in the engine determine the heat transfer. All the random points, for example, did not occur at either high, low, or intermediate flow rates. The random distribution of slopes, without investigation of the local temperatures, leads to considerable misgiving. For instance, heat rejection increased with 188 degrees Fahrenheit coolant at high flow rates at full load (shown in Figure 24) to a maximum flow rate of about 150 gal per hour. At lower flow rates, convection with the special circumstances of film boiling must control. At higher rates, the summation of the local effects was such that transfer by forced convection must have controlled. On further loading of the engine heat reject reached a minimum at the intermediate loads and rapidly increased at high flow rates. With colder 117 degrees Fahrenheit inlet water on the other hand, a characteristic similar to

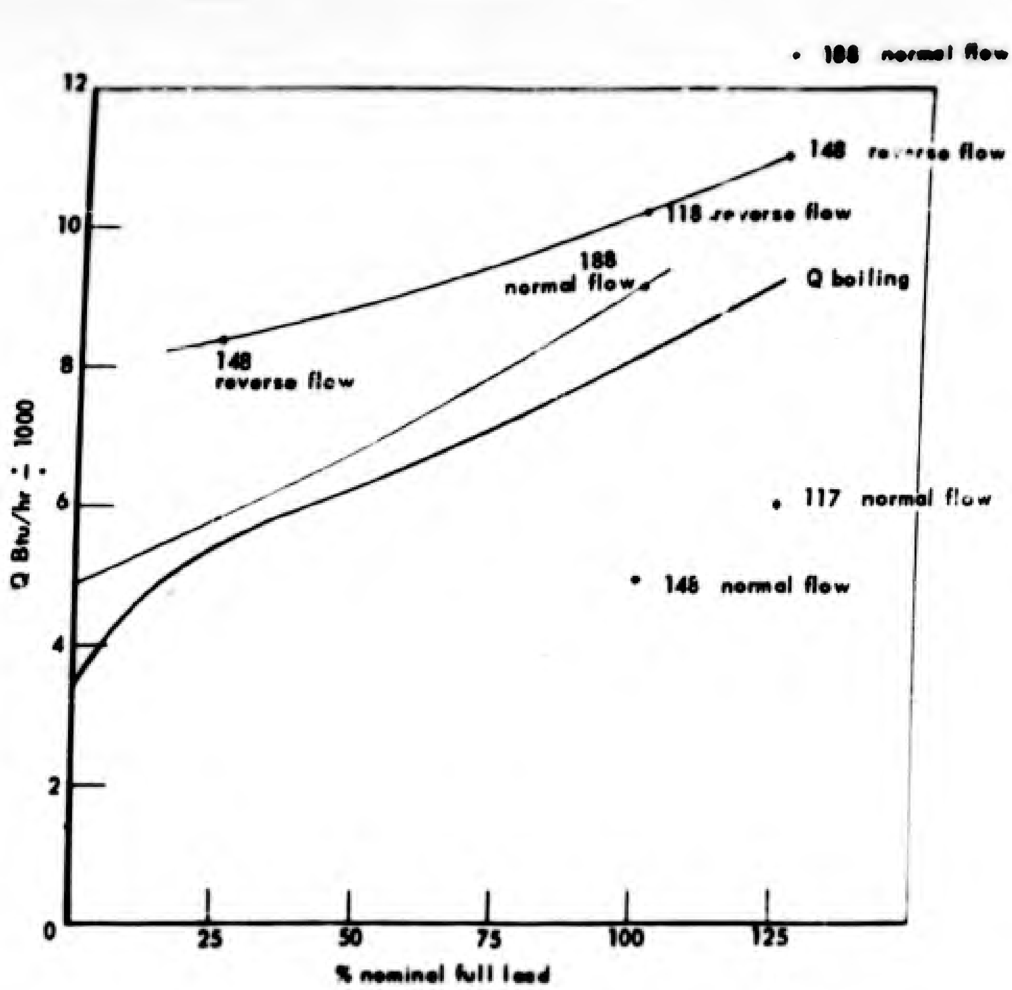


Figure 23. Graph of minimum values of heat rejection with various conditions of forced flow compared with those in a boiling engine.

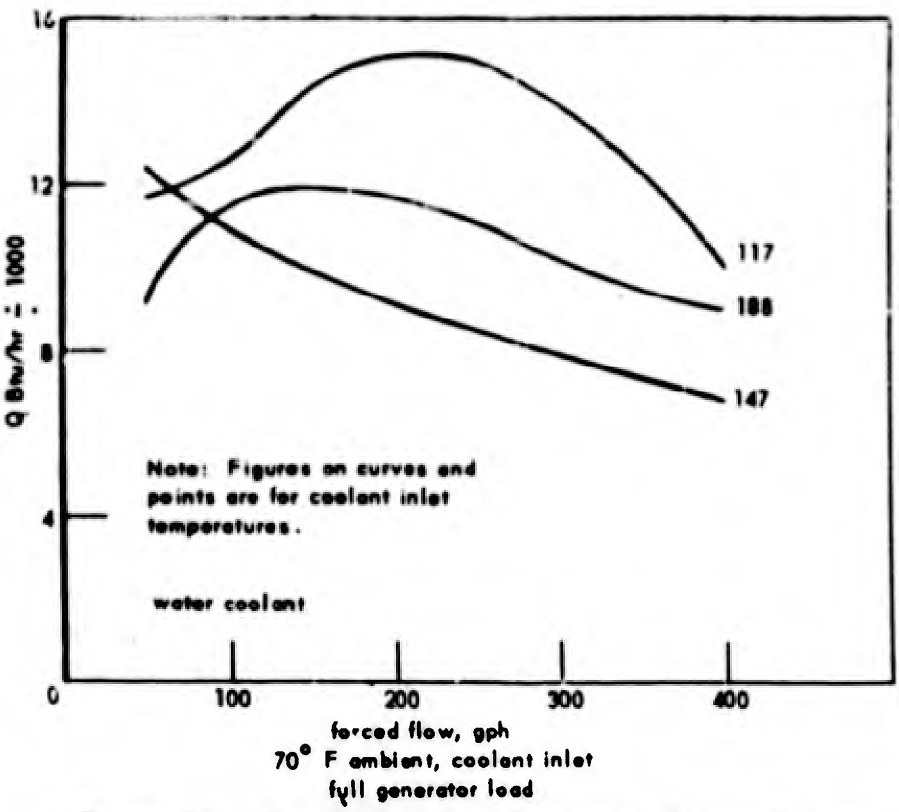


Figure 24. Graph of typical heat rejection variations with forced convection.

that with 188 degree Fahrenheit water, full load, was obtained. Surprisingly, intermediate coolant temperatures at full load gave characteristics of an entirely different nature (see Figures 20 and 24). In this case, the slopes of the curves are probably of less importance than their dissimilarity, i.e., the cooling process must be essentially out of control when we consider local metal temperatures. This, of course, stems from the relative importance of surface or film boiling and forced convection at a particular point. Either condition might yield a highly active cooling condition at the point, but the system decides when it arrives at a particular point just what it will do.

Figures 24A, 24B, and 24C show calculated values of heat rejection for several conditions, loads, and for the antifreeze mixture. Figure 25 shows typical temperature rise plots from which values from

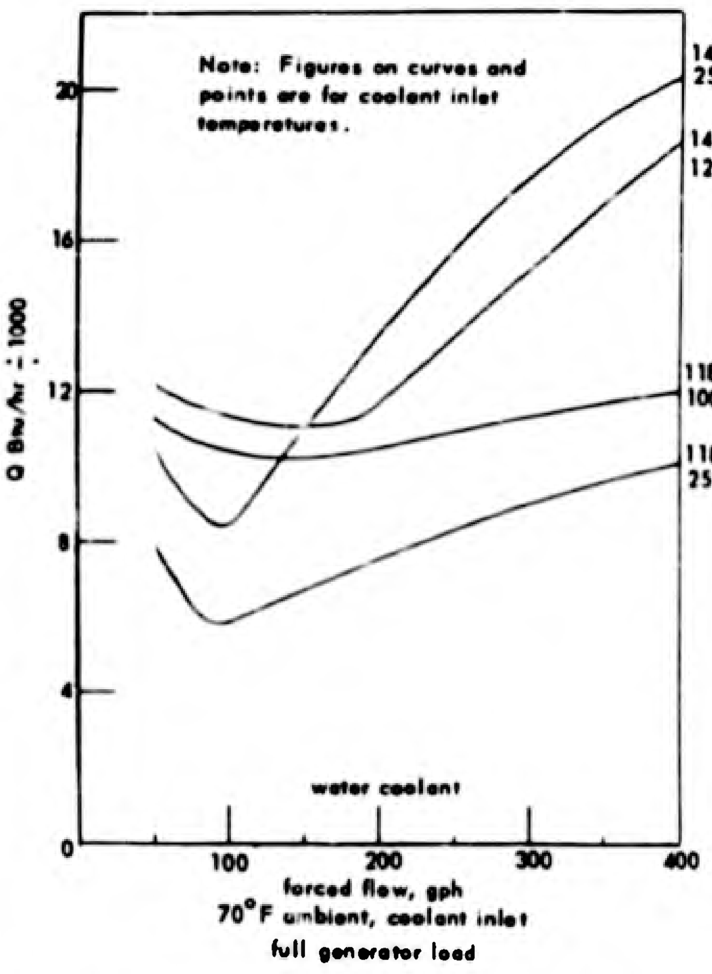


Figure 24A. Graph of typical heat rejections with forced convection.

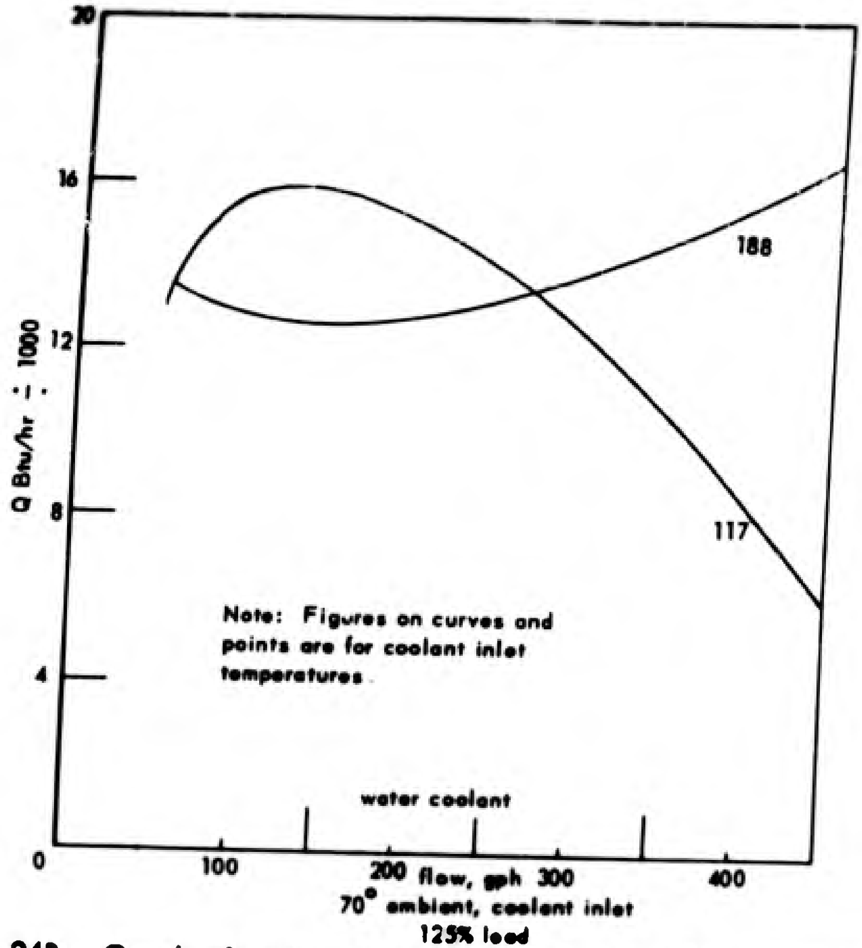


Figure 24B. Graph of typical heat rejections with forced convection.

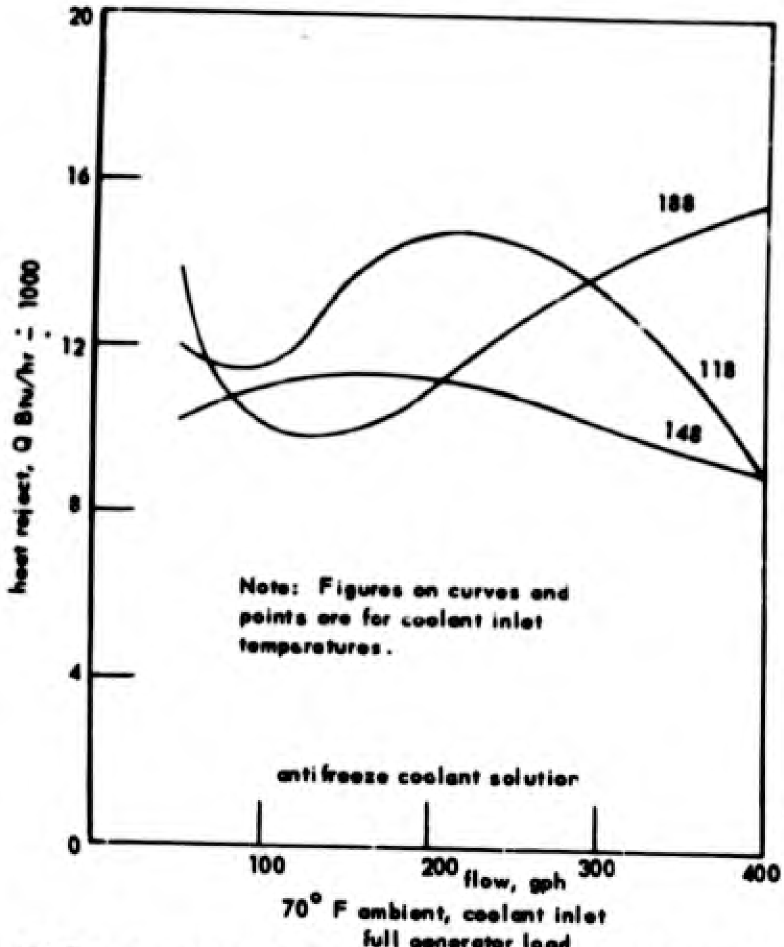


Figure 24C. Graph of typical heat rejection with forced convection.

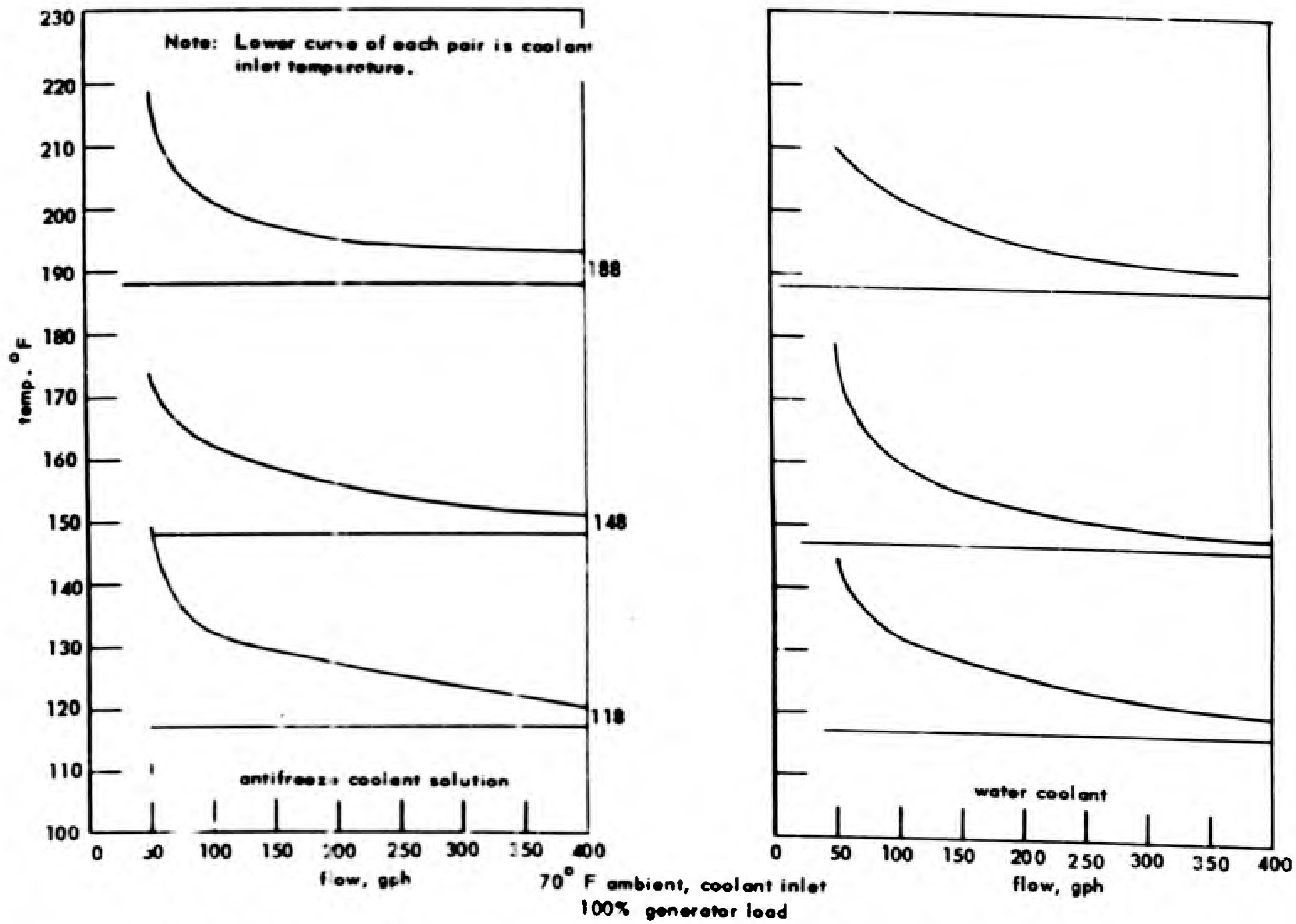


Figure 25. Graphs of typical coolant temperature rises with forced convection.

faired curves were taken to make the calculations of heat rejection with forced convection. The nature of the measurements, the apparatus, and the calculations are such that the values obtained are very accurate for low flow rates. Possible errors at the highest flow rate, 400 gal per hour, are as much as 25 percent because of the small temperature differences involved.

From the above generalizations it can be concluded:

1. Properly accomplished in a well designed system, cooling by boiling appears to have considerable merit over any conventional cooling system utilizing flowing liquid as the coolant, and which depends upon mechanically established and monitered forced convection cooling.

2. A well proportioned system should yield consistently lower cylinder wall temperature spreads, top to bottom, than other conventional systems.

3. The mechanism of local or surface boiling controls certain systems usually considered to be controlled by forced convection.

Lubricating Oil Temperatures

In all the above described investigations, no attempt was made to control the bulk lubricating oil temperature. Since lubricating oil is primarily heated within the bearings and on the under side of the piston, where it is used as a coolant, variation of the coolant temperature will have less effect than the engine load and ambient temperature. In Figures 26, 26A, and 26B, stable oil temperatures are seen to be only slightly influenced by coolant temperatures, flow rates, and loads, but are sharply affected by ambient temperatures. In Figure 26C, the data obtained from these controlled tests at about 70 degrees Fahrenheit and -25 degrees Fahrenheit are plotted with values obtained from similar generator units at Yuma and Fort Churchill. The severe depression of oil temperatures at low ambient temperatures and all loads suggests the dire necessity for some sort of oil heating at all except very high temperatures. This will be treated at length below under "applications."

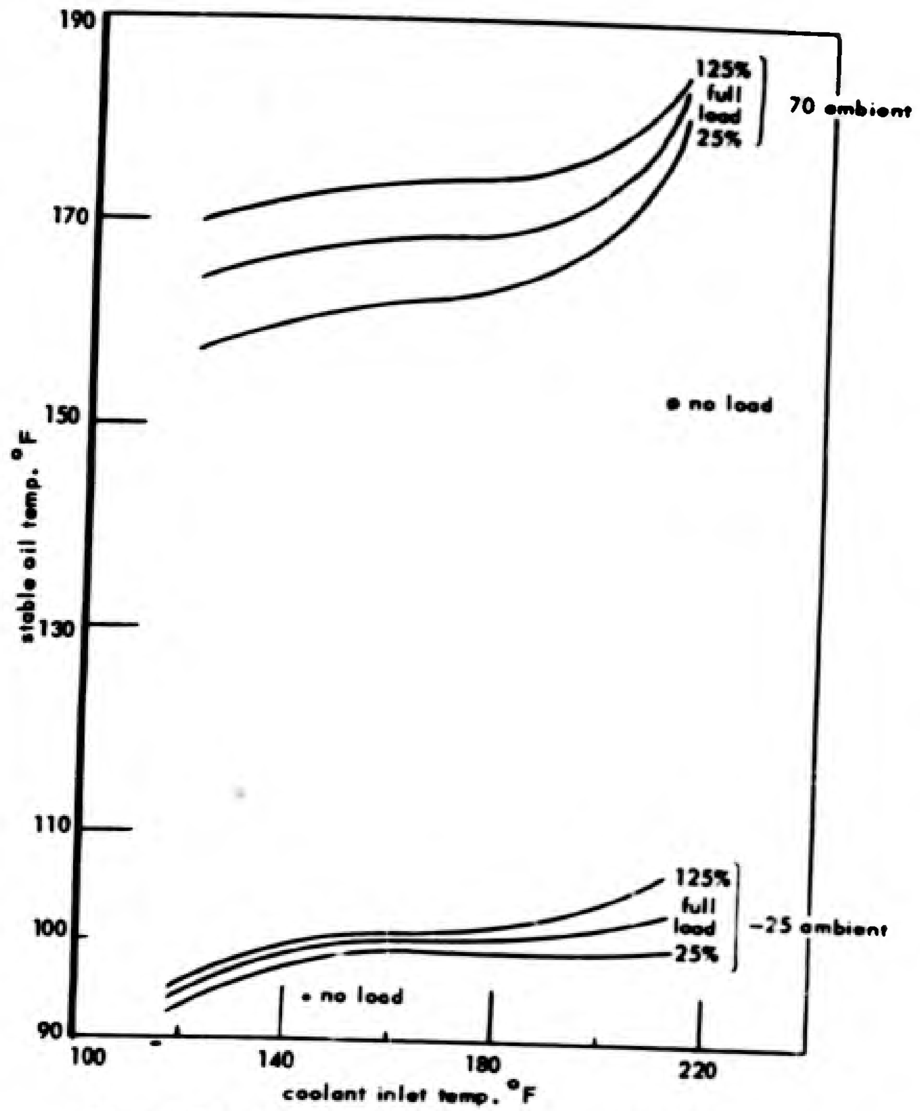


Figure 26. Graph of Influence of coolant temperatures, and loads on oil temperature.

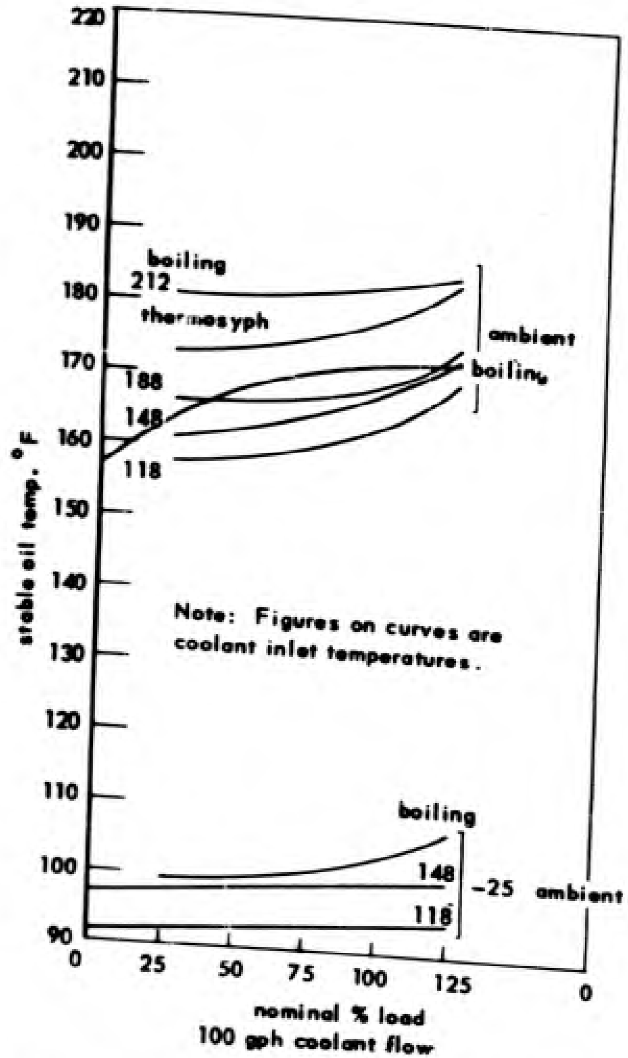


Figure 26A. Graph of Influence of coolant temperatures, and loads on stabilized oil temperature.

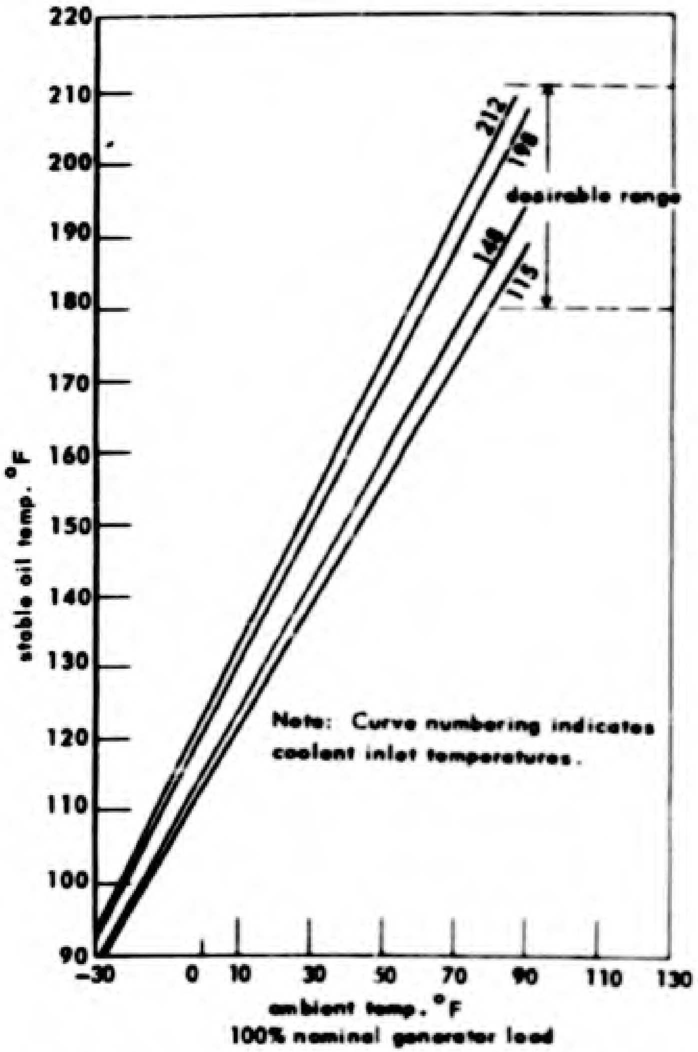


Figure 26B. Graph of influence of coolant temperature and ambient temperature on stabilized oil temperature.

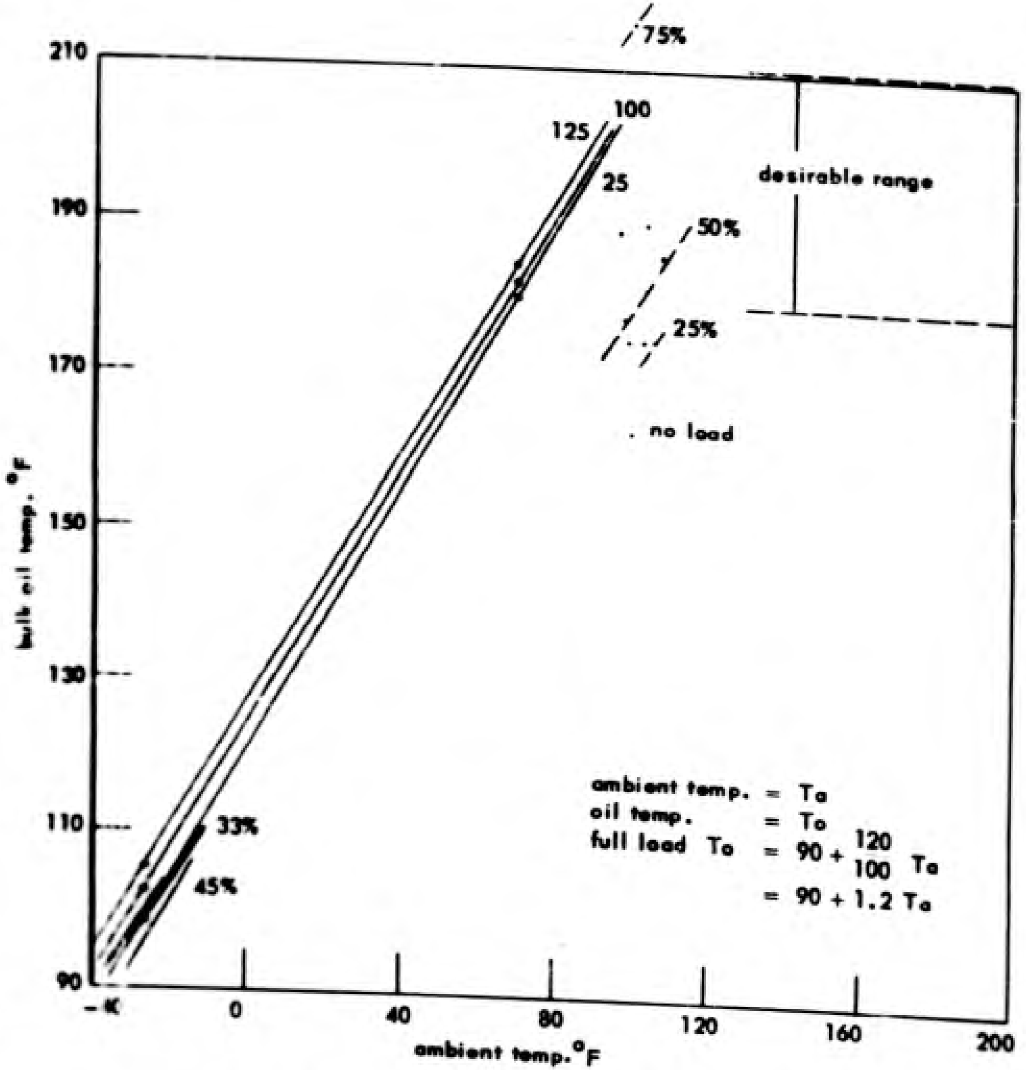


Figure 26C. Graph of variations in stable oil temperatures with ambient temperatures and loads.

While heat rejection, i.e., its scalar quantity, is of only passing interest in the above discussion, it should be noted that no great thermal efficiency increase can be expected based on reduced rejection with boiling. Typical "Q's" at all loads, coolant and ambient temperatures, are not in great variance, except in isolated cases of overcooling. Even here there are inconsistencies. It is of interest that "reverse flow cooling," which gave good ΔT's in comparison with other forced convection arrangements, also gave consistently high heat rejections. The possibilities of such a system in the upper part of the combustion areas with boiling in the lower reaches are very attractive for engines of proven poor cooling in the exhaust valve areas. Mechanically this could be quite simple, and, in a sense, the benefits are partially realized in many existing engines, provided the coolant temperature can be kept near boiling without losing the velocity advantages for severe local cooling of the valves with high velocity.

Before leaving the experimental work, it is worthwhile to consider how the heat rejection and metal wall temperatures agree with those predictable from the work of others.[1] At the nominal 125 percent load, Figure 22, "E", a measured heat rejection of 9190 Btu per hr at cylinder wall temperatures of 231-, 232-, 239-, 240-, 241-, and 253-degrees Fahrenheit exists, reading from the bottom up on thermocouples 1 to 6, inclusive. Considering each thermocouple, 1 to 5, inclusive, as representative of an annular portion of the cylinder with the couple at its mid-point, and thermocouple 6 as representative of much of the head area, an approximation can be obtained. For simplification, the wall thickness is assumed to be 1/4 in. and the top of the combustion area a plane surface. This will obviously lead to some error, but will be compensated for in part in that much of the head area is cooled by the incoming air on the intake side. Using values of ΔT from the literature (reference 1, Figure 164, p 314) as typical of the type of surface involved, by trial and error, values of heat transfer for the various areas are obtained.

In the following calculations it is assumed that the temperature of half the head area is represented by the highest cylinder wall temperature, thermocouple 5. This area is designated 6_a. The other half of the head area is assumed to be represented by the thermocouple imbedded in the exhaust port wall, and is designated area 6_b.

While the numerical approximations are obviously subject to error, especially in the overhead portion of the combustion chamber, because of the simplifying assumptions on area and uniformity of temperature, they do serve to illustrate approximate metal and surface temperatures, areas of major heat rejection, etc. They illustrate particularly well the facility with which a boiling engine can adjust itself to local heat transfer without great changes in metal temperature. Heat flux variations of the order of 7-1/2 to 1 are accomplished with water side metal temperature variations of only 18 degrees Fahrenheit. This will be seen to be nominal variation in terms of, for instance, air cooled engines, in which temperature differences of 400 degrees Fahrenheit are common.

Simple approximate calculations give vagrant heat losses from the cylinder area, hose, and separator at the boiling point of water from combined convection and radiation to be of the order of 700 Btu per hour. When this is added to the measured steam rejection for the nominal 125 percent load, a total heat rejection to the coolant of 10,000 Btu per hr is obtained. By a process of iteration, using the following assumptions, approximate heat losses in the engine are obtained:

Assumptions

Total heat rejected to coolant $Q = 10,000$ Btu/hr

Thermocouples 1 through 5, inclusive, represent median metal temperatures for wall strips 3/4-in. high

Surface temperature with boiling is typical of curves[1] for scaled surfaces, replotted in part in Figure 27

Half the head area, which is considered to be a plane surface, is represented by thermocouple 6, in the exhaust port wall. The other half is represented by thermocouple 5, at the top of the ring travel.

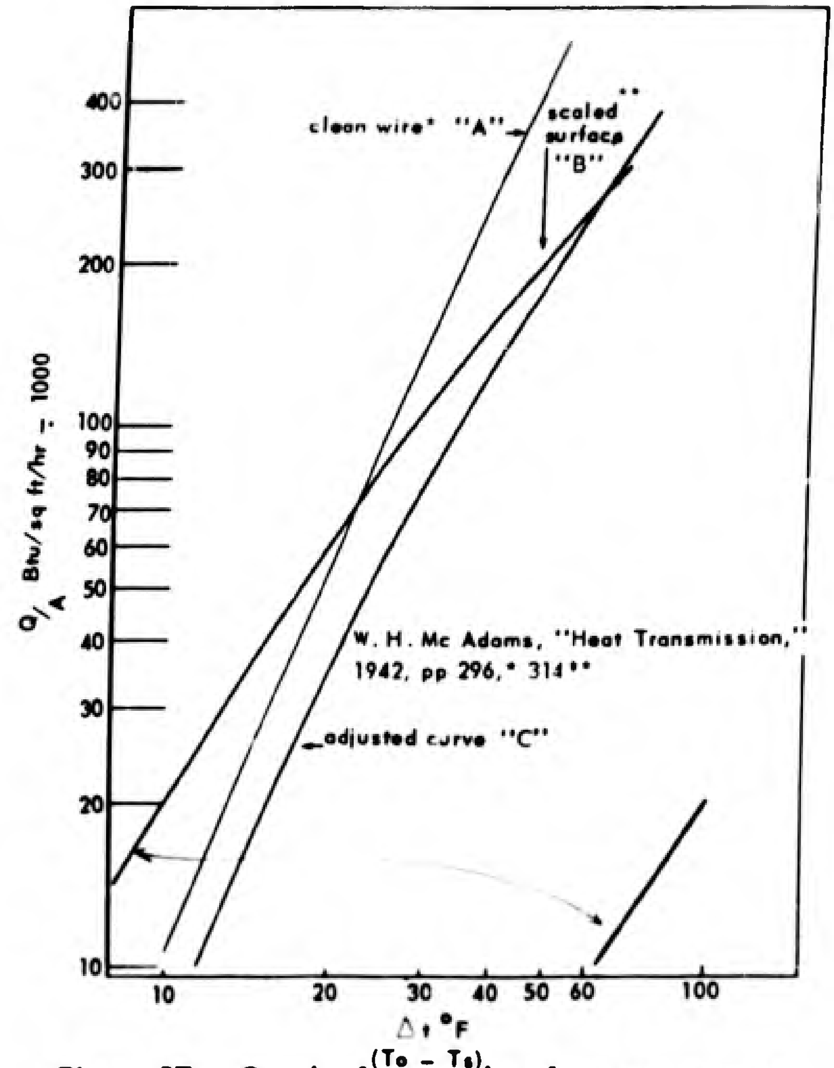


Figure 27. Graph of assumed surface temperature with boiling in engine.

Simultaneous conditions must be met, roughly, as follows:

Summation of

$$Q_1, Q_2, Q_3, Q_4, Q_5, Q_{6a}, Q_{6b} = Q \qquad (6)$$

$$Q_1 = \frac{-KA\Delta T_{m1}}{x_m}, \quad Q_2 = \frac{-KA\Delta T_{m2}}{x_m}, \text{etc} \qquad (7)$$

$$T_{m1} = \frac{T_{o1} + T_{i1}}{2}, \text{ etc.,} \qquad (8)$$

$$Q_1 = \frac{q}{a} A_1, \text{ etc., and values of } \frac{q}{a} \qquad (9)$$

must follow a pattern similar to that of experimentally determined relationships such as those plotted in Figures 6 and 27. No exactly similar situation, i.e., cast iron with scale as found in an engine, was available. By trial, a relationship approximately parallel to "A", from Figure 27,"C" , with some values the same as "B", is plotted from the left portion of (reference 1, Figure 164, page 314) which intersects the curve for the clean wire, "A". The horizontal displacement of the assumed line to the right probably indicates severe scaling and deaeration of the coolant.

Regardless of the accuracy of particular values determined, the overall conclusions appear valid and entirely reasonable.

From the temperature relationship (shown in Figure 28) with local values of $\frac{Q}{A}$, plotted to an arbitrary scale, certain conclusions are very helpful.

Under the conditions of the test:

1. Heat transfer in the lower half of the cylinder is negligible.

2. The highest rate of heat transfer, 59,000 Btu per hr per sq foot, is well below the critical probable value of over 350,000 Btu per sq ft per hour, by a factor of about 6 to 1.

3. Cylinder distortion as judged by wall temperature variation along the cylinder is negligible. It could probably be reduced further by introducing the coolant at or above the approximate mid-point of the cylinder at the location of thermocouple 3, reducing the fluid velocity on the lower half and increasing the cylinder wall temperature in that area.

The above treatment assumes that the relationships are not sufficiently well understood so as to be treated with an equation, therefore, the dimensionless approach. However, an attempt was made to express the heat transfer shown on the dotted "calculated local Q/A" (see Figure 28). The plot on log-log paper of local Q/A for the cylinder areas represented by thermocouples are plotted against the local Δt's (shown in Figure 29). All points fell on a straight line represented by the equation:

$$Q/A = 545\ \Delta t^{2.23} - 6{,}000. \qquad (10)$$

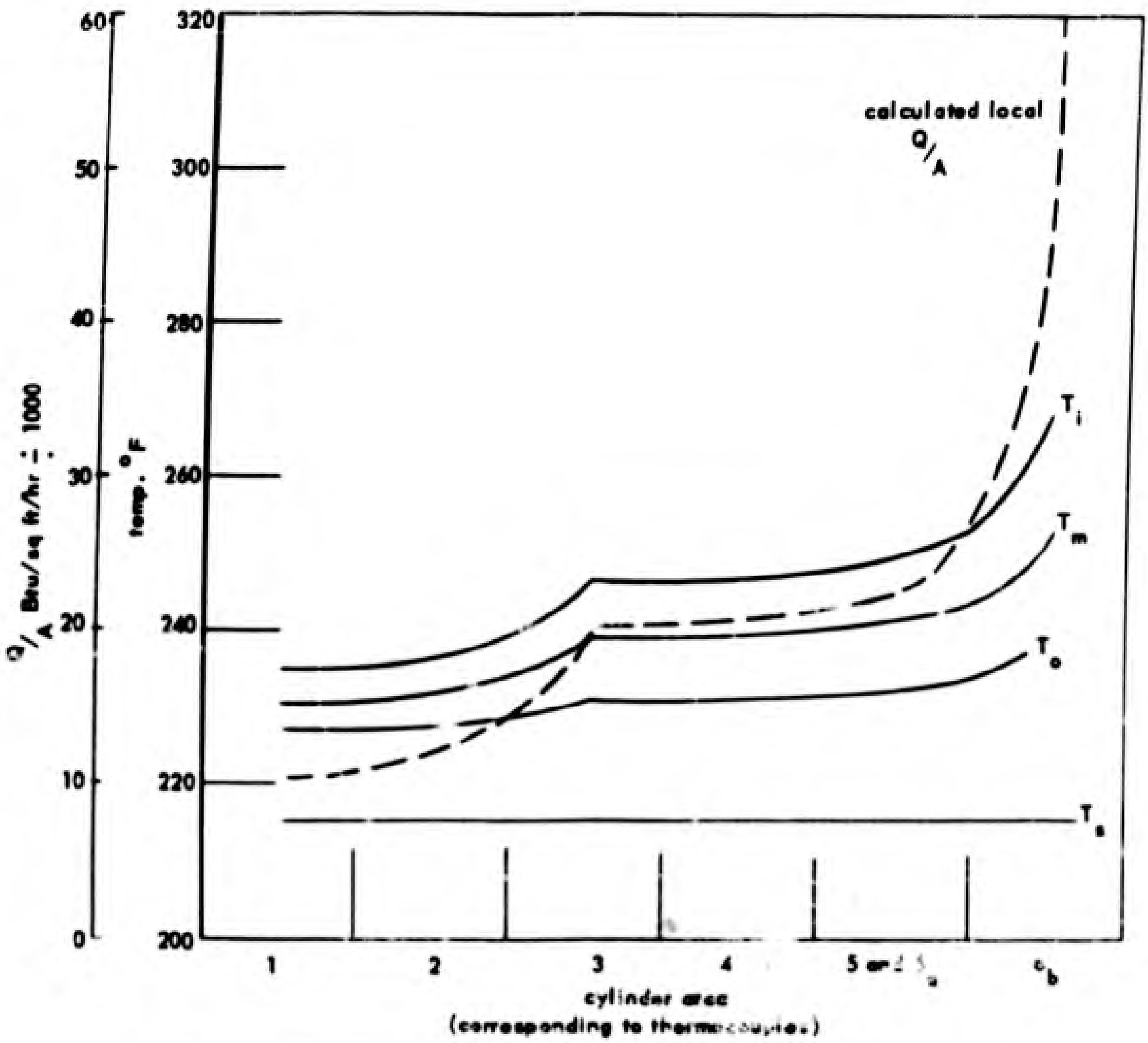


Figure 28. Graph of final trial and error values of engine temperatures and heat flow.

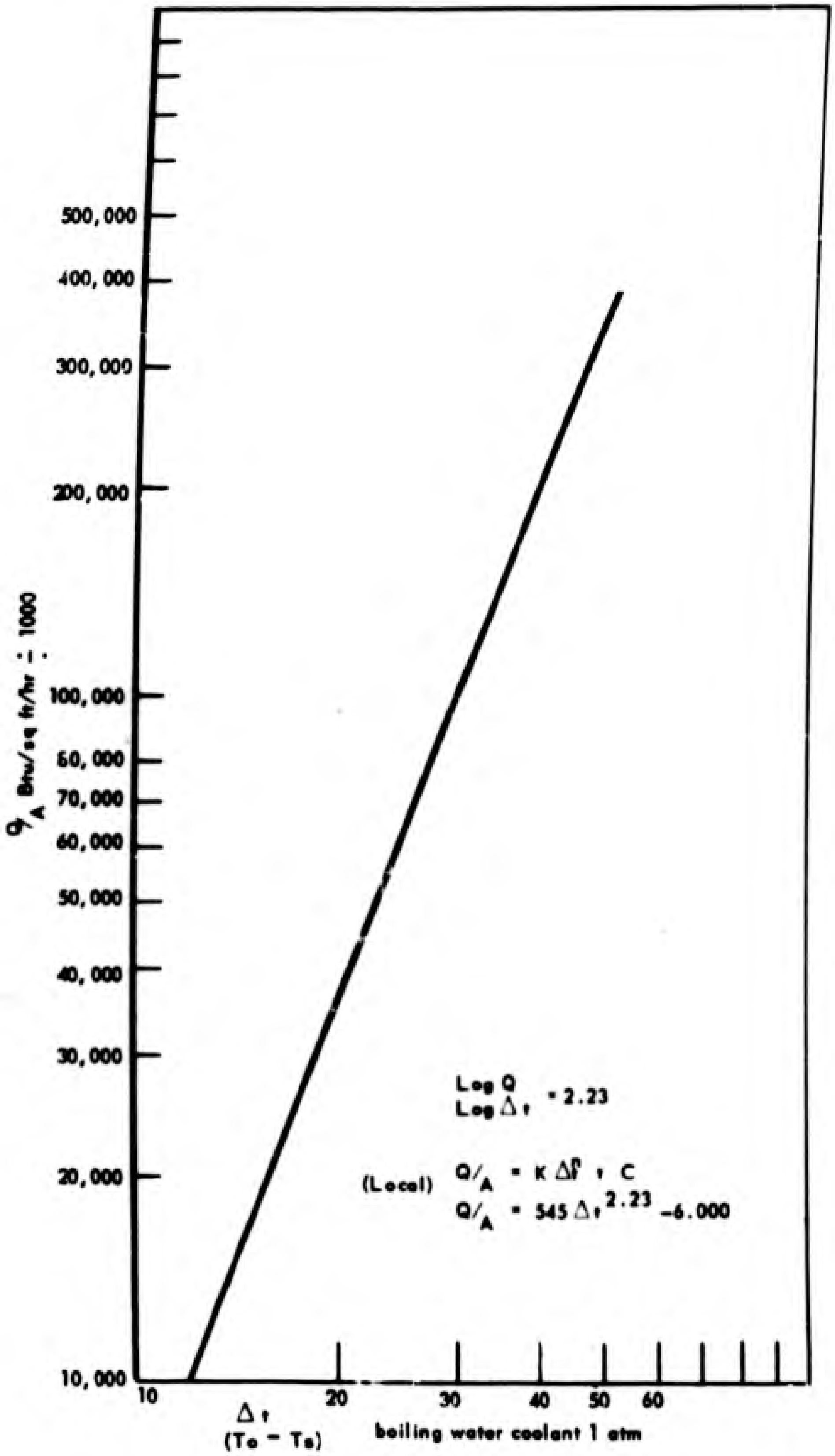


Figure 29. Graph of calculated local heat transfer within boiling engine.

The data shown in Figure 22, "F", are replotted in Figure 30 on a log-log system, and the resultant curve can be roughly represented by two equations:

From low values to 50 percent load,

$$Q = 5750 \text{ (percent load)}^{0.187} - 350 \qquad (10a)$$

From 50 to 125 percent load,

$$Q = 7300 \text{ (percent load)}^{0.51} + 900 \qquad (10b)$$

The discontinuity yielding the two relationships above barely shows in the original plot on rectangular coordinates, Figure 22, "F". Assuming that there is a change in the nature of heat transfer with load and coolant temperature and flow rates, there should be a discontinu'ty in a log-log plot in the total change in coolant temperature versus flow rates. At high flow rates, forced convection cooling should control. The mechanism of cooling-by-boiling becomes dominant at reduced flow rates (shown in Figure 31). These curves are the upper curve from each pair, Figure 25, replotted. As might be expected, the linear portion of the curves on a log-log plot starts at low flow rates with cold coolants, and relatively high rates with higher temperatures of coolant inlet. The final temperature of the coolant should reflect something of the nature of the heat rejection. Ethylene glycol solutions, with relatively inferior heat transfer properties of specific heat, conductivity, etc., should fall as a convective heat transfer media at higher velocities than water. Comparing the curves for water and antifreeze solutions, Figure 31, the apparent points of discontinuity are displaced toward the low flow rates for the antifreeze, the reverse of the expected effect. If the inference from this is correct, the antifreeze must inhibit boiling, and the net effect is that forced convection can be dominant at lower velocities with antifreeze solution than with water as a coolant. There may be serious side effects in promoting over-cooling of the lower cylinder areas at very low temperatures.

Applications to Equipment

The remainder of this report discusses the applications of some boiling-condensing systems to practical equipment.

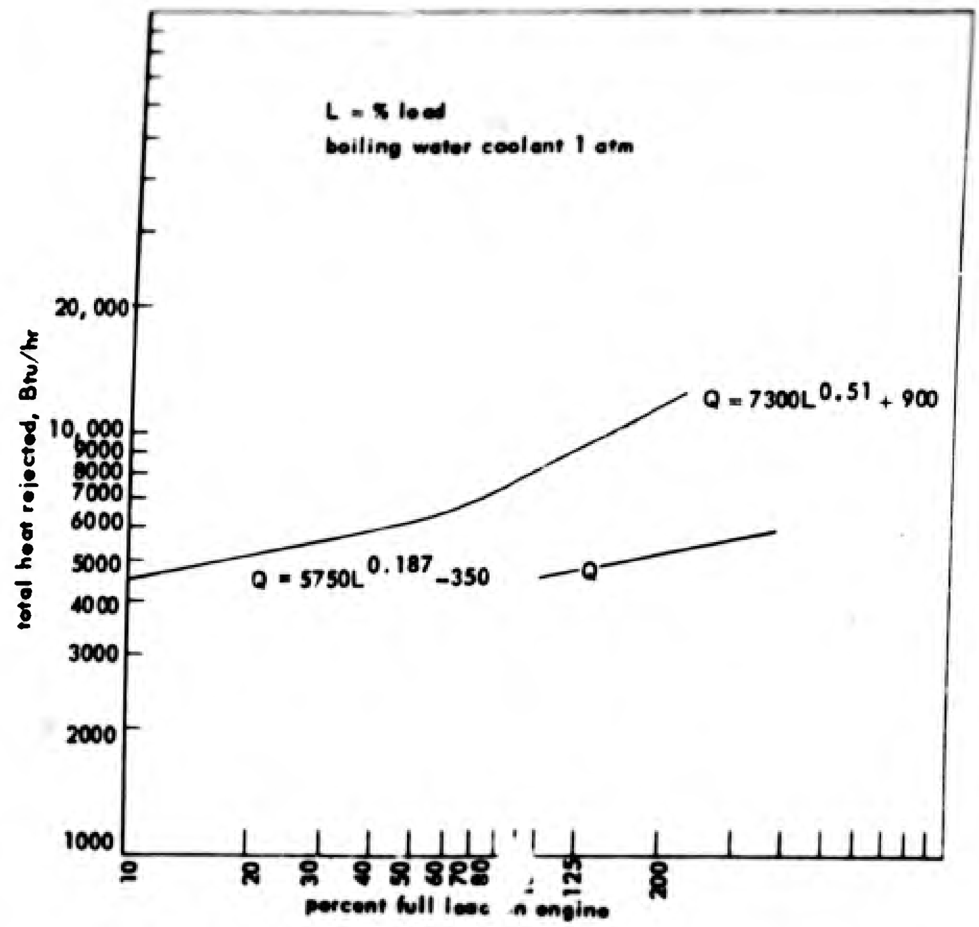


Figure 30. Graph of calculated total heat transfer within an engine from experimental data.

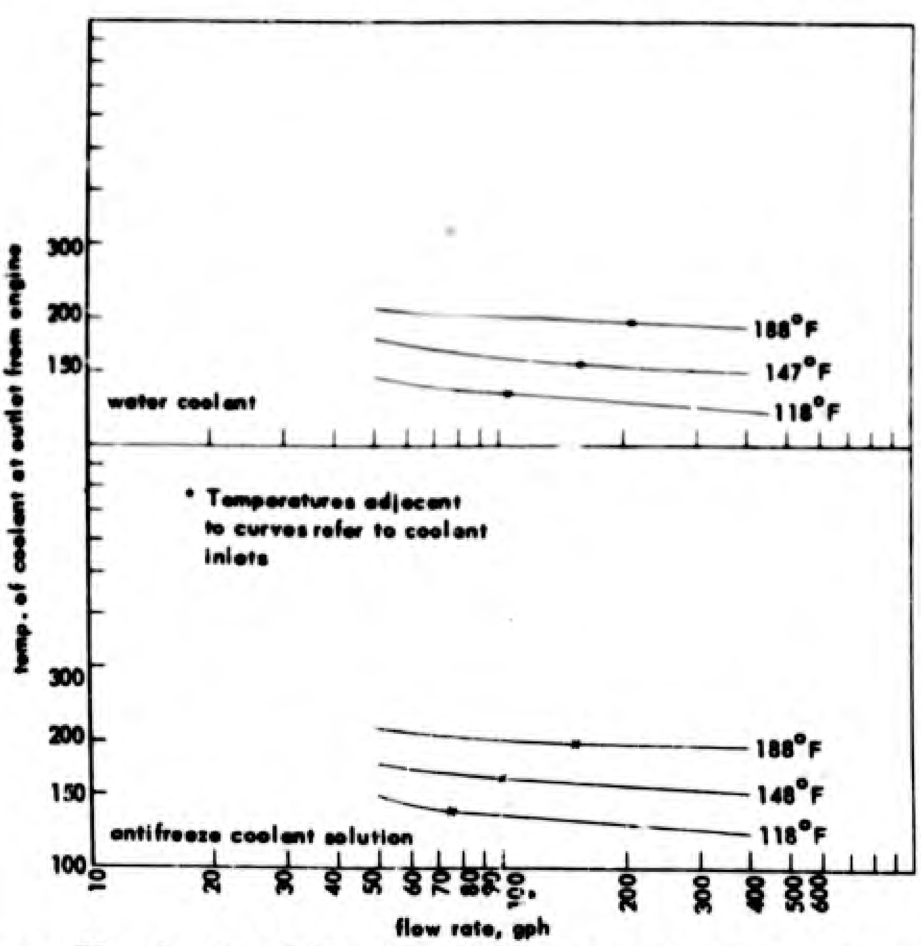


Figure 31. Graphs of data from Figure 25 replotted to log-log scale with approximate points of discontinuities.

The basic commercial application, generally applied to stationary equipment, is schematically shown in Figures 2 and 2A. For discussion purposes, the mean height of the water separation line "A", the size and slope of connecting lines, and their insulation, are basic criteria in determining the characteristics of the cooling circuit. Those portions to the right of the dashed line, on the other hand, relate to disposal of the heat not lost as vagrant heat. The heat will be contained as heat of evaporation of steam, and rejection will therefore be accomplished by condensation and return of the condensate to the system, preferably to the separator where subcooling can be counteracted by heating to the boiling point. Under these conditions, the engine will not be affected by the heat disposition, and the reject heat will be available for process, heating, fuel preheating, etc. In the systems discussed below the specific application will be discussed from both viewpoints separately, i.e., engine cooling and heat rejection.

In the simplest case, for stand-by engines with a plentiful and dependable supply of relatively pure fresh water, the simplest system conceivable will consist of an elevated tank with a minimum separation device, e.g., a hole or series of holes placed to provide exit for steam but not convenient for the entrained water, Figure 8. The old fashioned counterpart of the system is found in the open, horizontal piston "hopper" cooled engine (see Figure 1). At one time this system was used for engines driving small concrete mixers, and the operator occasionally filled the hopper with a hose or bucket from the same source as was used for furnishing concrete mixing water. Fitted with a minimum condenser over the hopper, the system is found on many oil well pumping engines today.

In the "vapor phase" system of Figures 2 and 2A, a circulation pump was usually furnished until a few years ago. In a favorable installation, the circulation and condensate return pump both can be eliminated, and the configuration becomes that of the hopper-condenser cooled engine, with components suitably separated by connecting lines, separating tanks, etc. In fact, all configurations below will vary in the location of the components to facilitate rejection of the heat at a pressure and in a manner convenient for its further use or disposition.

None of the above systems is limited at above-freezing temperatures. Below-freezing, the use of a solute with a low freezing point does not insure safety, since a solute of water with a solid or high boiling point liquid will necessarily produce a vapor of essentially pure distilled water. This can be conveniently and easily counteracted in a practical condenser by having the steam introduced to the lower end of relatively large vertical or sloping condenser tubing, with the condensate return in counter flow in the same tube, in the manner of old single-pipe steam radiating. Unfortunately, this produces the new problem of attempting to vent noncondensing gases heavier than saturated steam from the upper part of the condenser. In the "Laboratory design" of a boiling-condensing system this has been successfully counteracted in prototypes by having the steam first progress upward through part of the tubes, then what steam remains condenses on the downward pass, from which it is vented upward through a selected tube receiving the coldest air. Another arrangement (see Figure 13D) provided for condensation in the radiator (of the vehicle) with upward flow of the steam and continual enrichment of the condensate by drain off from the separator, and major drain down of the separator to the condensate sump on engine shutdown. A similar unit, (see Figure 2B) without drain down on a 30 kw generator set failed by freezing of the pump in early cold room tests. The "Jeep" installation performed successfully at sub-zero and high temperatures, at altitudes over 8000 feet.

A recent commercial application, shown in Figure 32, incorporates the pump, remounted on a commercial line of diesel engines, an elevated separator-surface condenser, and two coolant circuits.

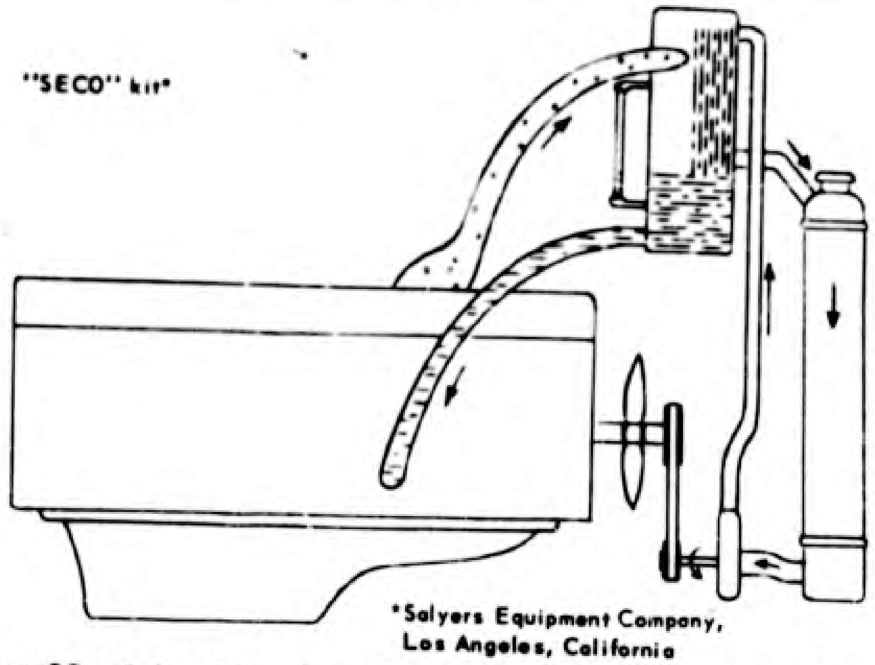


Figure 32. Schematic of commercial boiling application of liquid-coolant system to a commercial line of diesel engines.

The cooling circuit incorporating the engine is similar to that to the left of the dashed line shown in Figure 33. The pump is used to circulate the radiator coolant through the surface condenser in the upper portion of the separator tank, and handles only undiluted radiator coolant. The system was successfully operated at -65 degrees Fahrenheit in the cold chamber of the Laboratory by the manufacturer. Its main feature from a development point is that the few conversion parts are assembled largely from commercial components. A "kit" for field conversion consists of a few lengths of formed piping and the condenser-separator section. The "kit" is applicable only to the line of engines for which it was designed, but could be applied to other engines with modifications.

Of special interest in large stationary installations are the arrangements using steam turbines at low pressure for operating air circulation fans (shown schematically in Figure 34). This application is fully protected by patents, including variations with air-driven oil pumps, etc. This requires, in most installations, the equivalent of a boiler feed pump, "A", which can be eliminated if an equivalent water head "H" can be achieved by elevating the condenser. A head of 30 feet, for example, would furnish a steam pressure of approximately 12-1/2 lb per sq inch, well within the pressure limits for untended steam generators, 15 psi gage. At constant load, the steam condensed would just equal the steam necessary to maintain the liquid level; since the specific volumes of the steam and condensate are in the approximate ratio of 1585 to 1, formation of steam from the coolant would evaporate very little water in filling the steam system. The turbine application is especially attractive in that it is obviously self-regulating in addition to saving blower power. Recent information from the patent holder indicates that the price, size, and complexity of the turbine installation are currently materially reduced over that of first installations, which utilized commercial unit heaters and not turbines specifically designed for the application. The application is not limited to large installations with unlimited overhead, but is applicable to portable power units if sufficiently small turbines are available. As schematically shown, no "boiler" feed pump would be needed. A low displacement pump capable of handling the relatively small volume of condensate at engine pressure would allow relocation of the condenser and surge tank for those applications in which the head is not available in portable power units, and where the reject heat is desired for space heating in low silhouette buildings. A small positive displacement pump with a capacity two times the maximum steam production, placed as shown dotted at "A" would give the necessary head. To prevent the pump

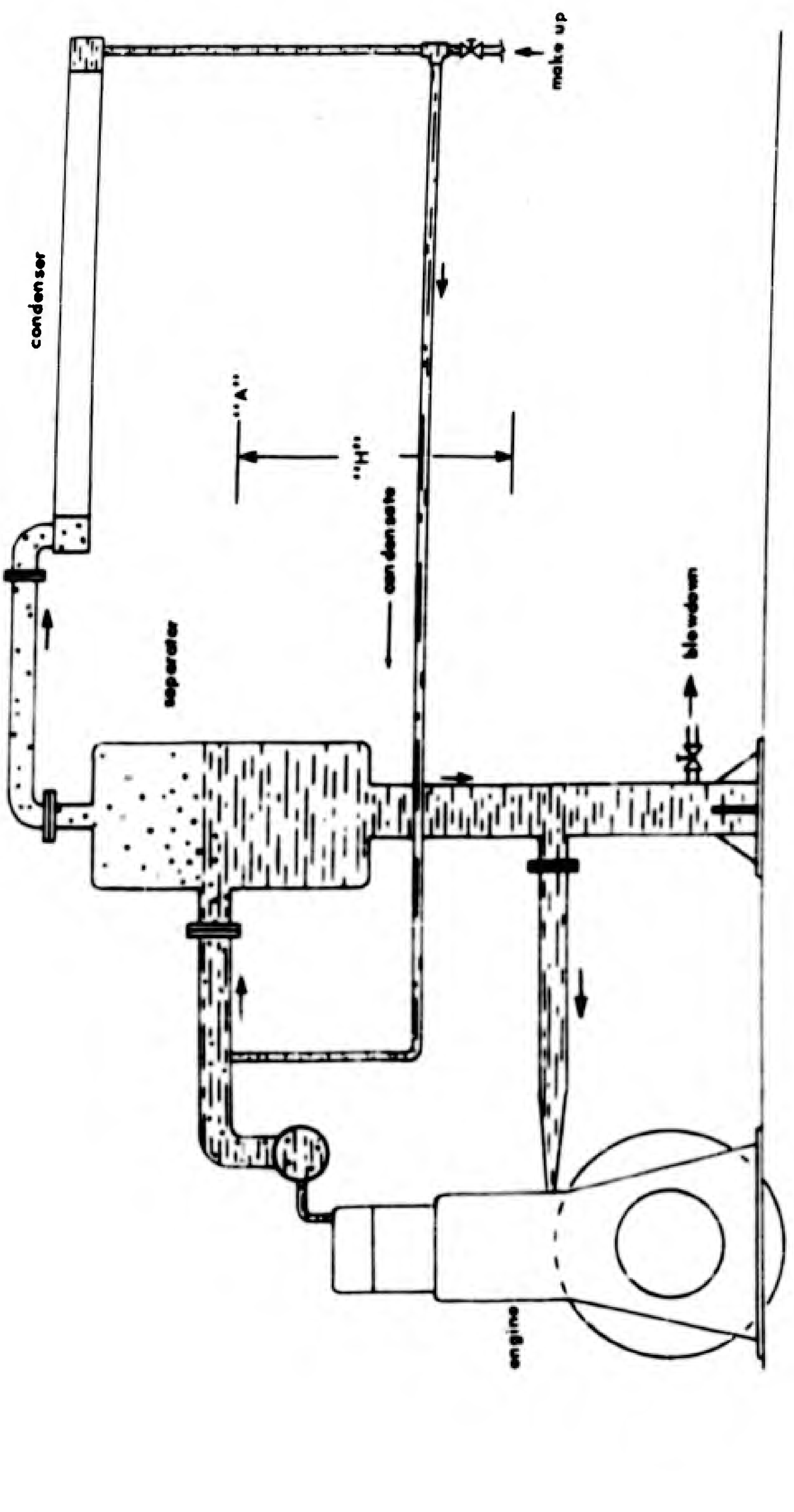


Figure 33. Schematic of complete boiling-condensing cooling system used to cool a large stationary engine.

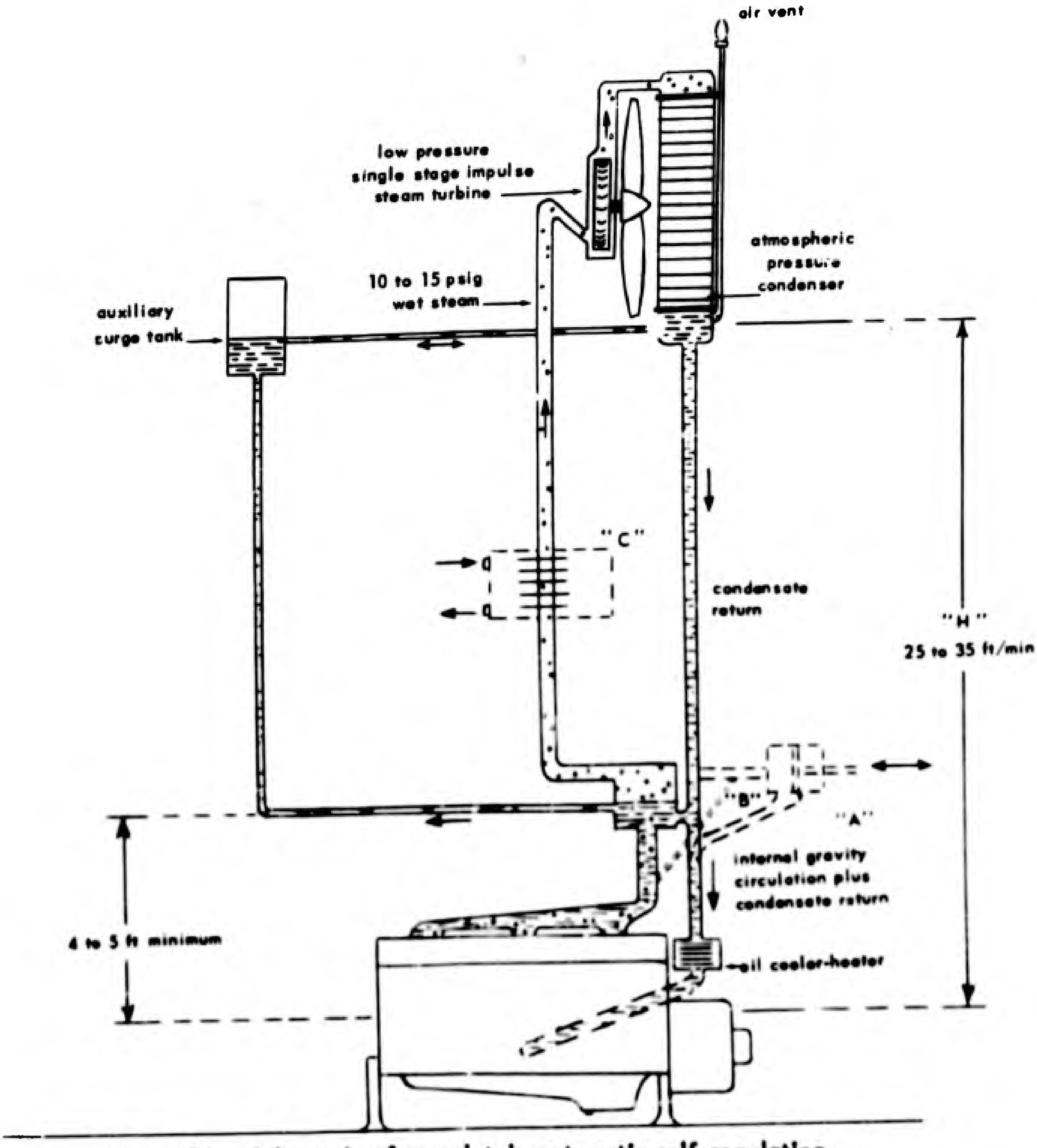


Figure 34. Schematic of completely automatic self-regulating boiling-condensing installation with steam turbine suitable for stationary installations.

from operating dry when steam production is off, a calibrated, restricted bypass from the steam line to its suction is indicated at "B". With suitable components arranged to prevent freezing of condensate, this exact arrangement would be directly applicable to mobile equipment. As stated elsewhere, this feature of using a turbine to power the blower is one of only two principal features of any of the described system fully protected by patents.[7] The other feature is that of using the gravity returned condensate in automatic oil temperature conditioning. Patent rights have not yet been fully established.[8] In any of the above applications, the steam can readily be used for space or process heating, or can be condensed with other than an air condenser, and a great variety of applications of otherwise waste heat are possible, i.e., utility water heating, snow melting for camp water, fuel oil preheating, etc. In the latter connection there is considerable interest currently being shown in the utilization of heavy residual fuels in diesel engines. In most successful applications, fuel preheating to about 220 degrees Fahrenheit, that of the usual steam in a low pressure system, is desirable and necessary. Preferential use of the steam for any such purpose is indicated at "C" in a dotted (see Figure 34) heat transfer device. Because of the excellent heat transfer characteristics of condensing steam,[1] the schematic representation is approximately the configuration desirable, except that the fins on the oil side should be vertical to promote gravity circulation. In all cases, the inclusion of drips, pressure relief devices, etc., typical of low-pressure steam practice, is assumed, and excluded from this discussion.

Before deserting the subject of pressure-operated steam devices, it should be noted that the pressures and temperatures associated with low-pressure steam do not necessarily represent the optimum for either the engine or heat utilization. Where higher pressure steam is desired and the extra expense to insure safety can be justified, there may be considerable advantage in operating the engine at higher jacket pressures. Safety of the engine jacket as a boiler should be a matter of direct approach as if it were a boiler, and the selection of valves, feed water treatment, piping, etc., approached from a conventional standpoint. There is ample experimental evidence[1,3,4] to indicate that steam pressures of, say, 50 psig (saturation temperature 298 degrees Fahrenheit) will give excellent temperatures within the engine, well below critical values established for air-cooled engines. Further, the temperature differential, steam to metal, on the cylinder wall will be reduced for a given heat flux[1,3] and the engine will not be affected by the coolant temperature rise of 86 degrees Fahrenheit but only a portion of it.

The above examples are cited only to demonstrate the potentialities of utilizing boiling-condensing arrangements in cooling internal combustion engines, and schematically show what has or might be accomplished with "conversions", i.e., engines designed to be cooled by forced convection. Obviously, this has some of the aspects of improvision. Optimum results will not accrue in attempting to adapt any and every engine without changes other than removing the circulating pump and thermostat. Good judgement indicates rather that those engines which by their construction appear to be "well cooled" should be selected for pilot applications. Care must be taken to remove all obstructions such as water distributing tubes and to collect the steam in the head with a minimum of horizontal travel, etc.

MAJOR POTENTIAL BENEFITS

Claimed for commercial versions of a boiling-condensing cooling system are the following:

1. Reduction of the temperature variation in the cylinder wall.

2. Lowering of "hot spot" temperatures (in the cylinder area).

3. As a result of 2, above, increase in engine overload capacity.

4. Maintenance of cylinder wall temperature above the dew point, reducing condensation, which leads to corrosion by acid formation and rapid wear.

5. Elimination of pumps and drives.

6. Elimination of pump seal leaks.

7. Reduction of piping costs.

8. Constant temperatures at all loads and ambient temperatures, i.e., using the boiling point of the coolant as the "thermostat" rather than mechanical devices.

9. Reduction in capital investment in cooling equipment.

10. Proper engine circulation self-regulated by the engine with changing load, according to two reports.

It is safe gross statement to say that all or most of the above have been demonstrated to be true to a greater or lesser degree. It is not believed, for instance, that the report on costs gives a true representation of the cost benefits to be realized with the current relatively noncompetitive state of the art. Radiator construction of aluminum, for instance, has been retarded by the requirements for corrosion and small water passages, etc., necessary for handling water successfully. Fully developed, relatively larger passages of aluminum to condense essentially pure water appear plausible. The increased heat transfer from the condensing steam to the tube wall should materially simplify and reduce costs, since an economical heat transfer will have few relatively larger tubes with fewer joints. Smaller fans with less fan power loss should result. With the relocation of the condenser for all but a few installations in which the visibility from the cab prevents it, better utilization of the ducted output of efficient fans should be realized. The potential savings in going to low pressure steam turbines are obvious. The typically quoted value of 6 percent of power output for accessory pump and fan drive could be entirely eliminated.

There is considerable evidence that diesel engines, particularly, benefit from the slightly higher cylinder metal temperatures which would occur in a boiling engine compared with one cooled by conventional forced convection at light loads. Very high cylinder temperatures, on the other hand, while perhaps of considerable benefit to spark ignition engines in regard to corrosion prevention, deposit formation, etc., will require the premium fuels typical of aircraft engines. There is no evidence that slight increases in temperature associated with boiling-condensing operating at atmospheric or slightly higher pressures will affect the fuel requirement. If a cylinder capable of withstanding coolant pressures of several hundred pounds per square inch were to be made, diesel engines might benefit, while it is certain that spark ignition engines would require premium fuels.

From an extensive literature study, it is concluded that some limiting acceptable temperatures for various engine components have been established in practice as being in the operable range. The tabulation below reflects what various investigators and manufacturers have found tenable, and does not give clearly acceptable values for every situation. For instance, while 250 degrees Fahrenheit might be a desirable and acceptable value of cylinder wall temperature for most engines, a specific engine might require much lower cylinder wall temperature to facilitate piston cooling through the rings and piston wall contact area.

Component	Lower limiting °F	Desirable range °F	Upper limiting °F
Crankcase lubricant bulk temperature	160	180-220	280
Fuel where heating is necessary	Cloud point	180-200	220
Cylinder wall	190	200-350	500
Cylinder head	200	250-400	500
Valve tulip	--	1200	1400
Valve guide	--	200-300	795
Valve seat	--	200	515

Part IV. SUMMARY

Chapter 4

CONCLUSIONS AND RECOMMENDATIONS

Conclusions

It is concluded that:

1. Cooling of an internal combustion engine by allowing the coolant to boil, thus transferring heat from the engine to the reject device condenser as latent heat of evaporation, will prove beneficial in many ways. The benefits are listed in approximate order of value as follows:

(a) Raising the cylinder wall temperatures above the dew point of the combustion gases, thus eliminating condensation conducive to formation of acids by combination of water with combustion products.

(b) Eliminating corrosive wear, allowing the use of sulfur-containing "non-premium" fuels, possibly at very low ambient temperatures.

(c) Eliminating overcooling of the front of the cylinder block by cold air at intermediate ambient temperatures, and eliminating excessive general cooling of the block at very low temperatures.

(d) Through using a suitable heat exchanger, provide the basis for automatic oil temperature control tending toward stable crankcase temperatures of the order of 200 to 220 degrees Fahrenheit, heating to those temperatures during warm up and cooling at high outputs and ambient.

(e) Reducing crankcase dilution at low ambient temperatures by raising the temperature of the oil charge, thus promoting evaporation of the diluents.

(f) Simplifying the manufacture, service, and installation of cooling systems, eliminating major sources of trouble, the water pump, and the thermostat.

(g) Increasing the thermal efficiency of the power plant by decreasing the power consumed by the pump and the fan.

(h) Promoting the fastest possible warmup of the cold started engine.

(i) Yielding more uniform cylinder wall temperatures, top to bottom, eliminating or reducing taper and [illegible] cooling around the cylinder (the cause of [illegible] distortion).

(j) Allowing relocation of the reject device (condenser) in more, elevated locations without excessive pumping requirements.

(k) Reducing water makeup in systems from steam loss at high ambient temperatures.

(l) Making available highly usable reject heat in the form of low pressure steam without danger of starving the engine for heat, i.e., overcooling it.

Recommendations

It is recommended that:

1. Application of boiling-condensing cooling be made to all stationary and semistationary systems, notably pumping plants, power plants, etc.

2. An appropriate application of a boiling-condensing cooling system be considered for cooling of all future engines purchased except those assured of medium to high continual loading at ambient temperatures usually above 70 degrees Fahrenheit, e.g., heavy duty truck and tractor engines in tropical use.

3. That all applications be preceded by a pilot application furnished with the original equipment. That is, conventional engine cooling systems should not be modified by field forces for proof testing.

ACKNOWLEDGEMENTS

The author wishes to express his appreciation to Mr. K. N. Tinklepaugh, NAVCERELAB, an ardent co-worker during the developmental phases of the study and original project engineer; to Mr. James L. McCarthy, NAVCERELAB, for his interest, efforts, and advice in fabricating and testing the various working models; to Mr. Lloyd Harbert of Engineering Controls, Inc., Los Angeles, California; to Mr. O. B. Freeman of Tidewater Associated Oil Company, Ventura, California, and to Mr. A. Tachaella of Salyers Equipment Company, Los Angeles, California, respectively, for their interest, advice, and help in securing technical literature, and sharing their experiences and test data. He is particularly grateful for the untiring efforts of Mr. Charles S. Bates, NAVCERELAB, in the organization and editing of the final manuscript.

Appendix A. DISCUSSION OF THE LITERATURE

APPENDIX A

SURVEY OF THE FIELD

The literature offers much test data relating engine wear, parts life, lubricants, engine capacities, etc., with some cylinder or coolant temperature, or both. Quotations are given[9] in full because they sum up the data fairly well as follows:

"It can therefore be concluded (from these graphs) that excessive dilution is only likely to produce high cylinder wear when the oil is contaminated with abrasive matter," p 31.

"It can, therefore, be concluded from these tests that, cylinder-wall temperatures of 125° to 265° C. (257° to 509° F.) and under steady running conditions, deficiency of oil is unlikely to be a factor of practical importance in regard to cylinder wear," p 34.

"At temperatures above 265° C. (509° F.) trouble was experienced through scoring of the piston skirt, this giving rise to large amounts of detritus which accelerated ring and cylinder wear," p 34.

"There is no doubt that the accelerated wear which occurs below 90° C. (194° F.), associated with the condensation on the cylinder wall of the water-cooled engines from the products of combustion...," p 34.

"...experiments described in the earlier part of this report indicated that the bulk of cylinder wear occurs when warming-up," p 47.

"The foregoing results therefore indicate that the sulphur content of high-grade petrols has not an important influence in relation to cylinder wear at low operating temperatures, but that sulphur contents of 0.1 percent or higher have a marked effect in accelerating cylinder wear," p 57.

"It will be observed that the cylinder wear under these conditions (low cylinder wall temperatures) was 80 percent greater, and the top piston ring wear 100 percent greater, with the alcohol blend than with the ordinary petrol."

"It will be seen (by referring to the test data) that the piston ring wear at high operating temperatures was the same on the two fuels, a result which would be expected if corrosion no longer plays a part at such temperatures," p 58.

"Consideration of the foregoing would appear to indicate that the field for upper cylinder lubrication, if any, is in relation to starting from cold, particularly if arrangements can be made to add temporarily a high percentage of oil to the intake, provided this can be done without interfering with carburetion or ignition," p 64.

"The above table indicates that, under conditions of these tests (110° C., 230° F. cylinder wall temperature), the rate of wear with pure medicinal paraffin was very low and, in fact, was approximately the same as that of the commercial mineral lubricant, the figures for which are given in the same table. It is interesting to note that the engine was run for more than 400 hours at normal temperatures when using pure medicinal paraffin, without any trouble due to wear, and the engine was always remarkably clean, there being no carbon formation. It will be recalled that, under conditions of corrosion, the wear was very high with medicinal paraffin," p 76.

"For example, it is clear that rapid warming-up is of greatest importance and that, from the standpoint of cylinder wear, it is possible to treat an engine too gently when starting from cold. Any devices, such as thermostats, evaporative-cooling, etc., which reduce the warming-up period to a minimum will obviously be of assistance," p 43.

"Instances of unequal wear of different cylinders of a block have been attributed to distortion resulting from non-uniform thickness of the cylinder walls, from unequal flow of the cooling water, and from the cooling effect of the fan on the front part of the block," p 7.

"There are various methods of minimizing the harmful effect of dilution when starting. Thus, on one car, oil is automatically added to the petrol when starting from cold and it is interesting to note that this particular engine shows the very low cylinder wear of 0.001 in. per 17,000 miles and that it is the only engine on record which does not show maximum wear at the top of the cylinder bore," p 17.

"It may be remarked, in parenthesis, that it does not necessarily follow that a low load is preferable to a high load, when warming-up an engine; in fact, the reverse is true, since the increased cylinder wear resulting from a heavier load is more than offset by the beneficial effect of the shortened warming-up period," p 55.

"It is probable that this increase in corrosive wear with load (at low cylinder wall temperatures) is due to the more vigorous scraping action of the piston rings as the pressure behind them is increased, this action tending to remove detritus and any protective film of oxide or lubricant." p 71.

This particular work is significant in that so many variables were examined under identical conditions, allowing the most valid of conclusions. Published in 1940, the variety of attacks to determine cylinder wear is no less than amazing and covers such diverse items as chromium plating, nitriding, dilution, cold starting, air cleaners, ethyl fluid, use of hydrogen fuels, compounding of lubricants, etc.. for a total of over 50 independent variables. Briefly, the remainder of the bibliography corroborates the findings of the work.

Figures 35, 35A, 35B, and 36 are reproduced.[7] Figure 35 shows calculated dew point temperatures of cylinder gases with increasing pressure. While this curve does not and can not take into account the variation in partial pressures with burning of combustion products, it does indicate the severity of the situation in which at, say, the point of expansion curve of 200 psi is reached. If at that time the cylinder wall on the part exposed to gases is not at or above the dew point (condensation temperature) of about 240 degrees Fahrenheit, condensation on the "fire wall" will occur. The importance of maintaining cylinder wall temperatures above the dew point is illustrated in Figures 35A and 35B. From these and similar data it has been deduced and widely corroborated that a minimum safe cylinder wall temperature of 194 degrees Fahrenheit should be maintained. Similar conclusions can be drawn from Figure 36, which shows two curves from Williams[7] relating cylinder wall wear to engine loading. At low wall temperatures, wear rate is highly sensitive to mean effective pressures, but at 248 degrees Fahrenheit is virtually dependent and relatively much lower.

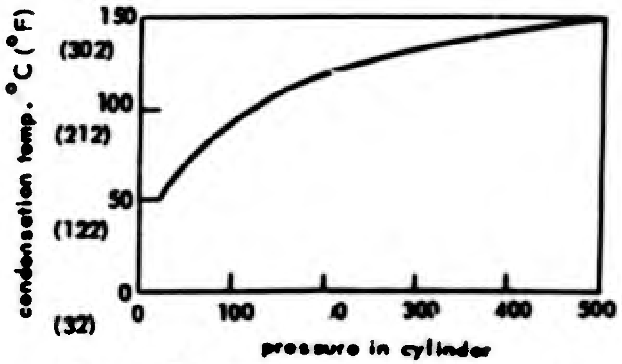


Figure 35. Graph* of condensation temperature rise.

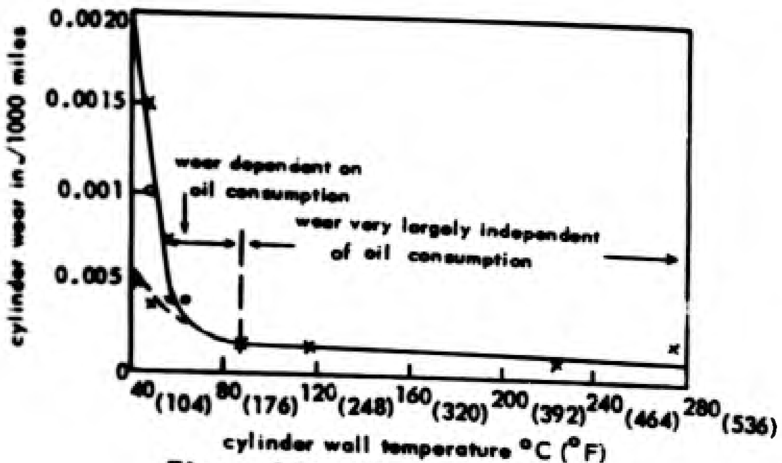


Figure 35A. Graph* of cylinder wear.

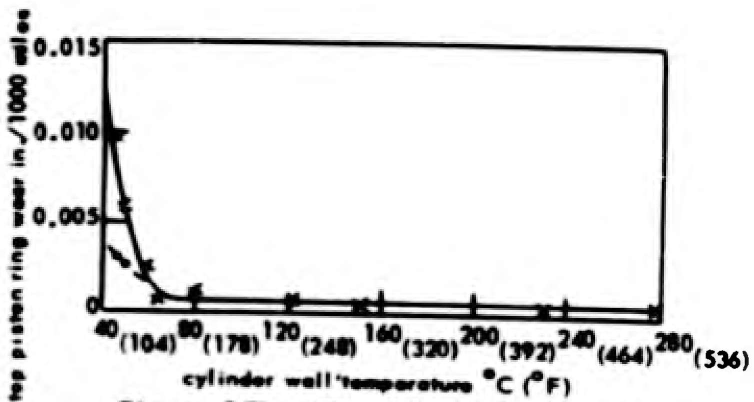
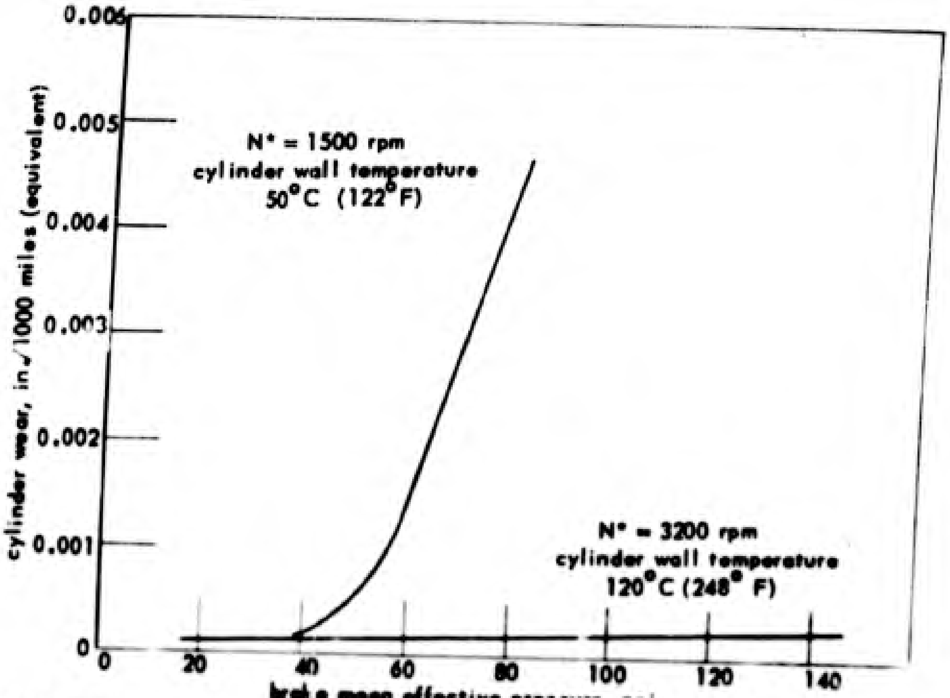


Figure 35B. Graph* of piston wear rates similar to those of cylinder walls.

* C. G. Williams



*C. G. Williams (upper curve, Figure 5, p. 72), (lower curve, Figure 4, p. 70).

Figure 36. Graph* of wear rates at varying engine loads and cylinder wall temperatures.

The investigator[7] found it possible to reduce materially cylinder and ring wear, the major cause of engine retirement, by practically every technique studied, i.e., changing oil composition, plating, changing dimensions, using upper cylinder lubricants, etc., but the necessity for the refinements was in every case eliminated or largely lessened by raising the cylinder wall temperature (located as was thermocouple 5 of the single cylinder engine tests reported here) above 194 degrees Fahrenheit. It was found, both in controlling laboratory testing and in comprehensive fleet analysis, that any condition which tended to reduce cylinder temperatures below the criticals, e.g., such as extended idling, light loads, poor design, faulty thermostats, etc., promoted sludging and corrosive wear. It is probably not an oversimplification to say that most of the severe operating problems, other than abrasives in the air intake and oil, stem from condensation, either in the cylinder area or crankcase, or both. Wherever waters occur, there are the solvents necessary to form corrosive acids in the products of combustion, the most common and frequently cited being sulfur. In this light, sulfur can be considered to be a good fuel when its products of combustion are exhausted without being dissolved in water.

The investigator[7] and others cite cylinder wall temperatures of a maximum of the order of 450 to 500 degrees Fahrenheit to be acceptable. In determining the cause for scoring in a free-piston gasoline operated air-cooled drilling device, measurements of cylinder temperatures were made on two units. In the unit giving trouble, temperatures shown in the upper curve of Figure 37 were found to cause seizing at the end of the period indicated, with severe scoring of the cylinder walls by the rings. The driller did not have well established coolant (air) flow over the cooling fins, but depended largely on gravity circulation. Another device of approximately the same capacity but with forced fan cooling, lower curve, operated satisfactorily without evidence of overheating. This is in agreement with the findings[7] and aircraft practice.

Without dwelling on the subject, it should be noted that a common cause of distress in practice has been the mechanical failure of thermostats. This is a particularly insidious problem since its detection would be possible only with the best of engine instruments and highly trained operators, trained to mistrust low operating temperatures.

The type of operation for which minimum temperature control will be most important is that described[10] under Internal Combustion Engine Service Classifications, designated MS (Gasoline Engines), DS, (Diesel Engines). The general conditions of service are quoted: "Service MS represents the most severe service encountered in the operation of gasoline and other spark ignition engines. It includes two different types of severe or adverse operating conditions which are as follows:

"Start and stop operation: This causes the steam formed from fuel combustion to condense as water on cylinder walls and in crankcase. Other combustion by-products, mixed with this water, form acids which caused corrosive wear on cylinders, pistons and rings. This type of operation promotes oil ring and oil screen clogging, varnish deposits, especially on hydraulic valve lifters, and the formation of crankcase sludge containing water. It also causes dilution of the oil by unburned fuel. In passenger cars and other units the severity of this operating condition increases in wintertime as atmospheric

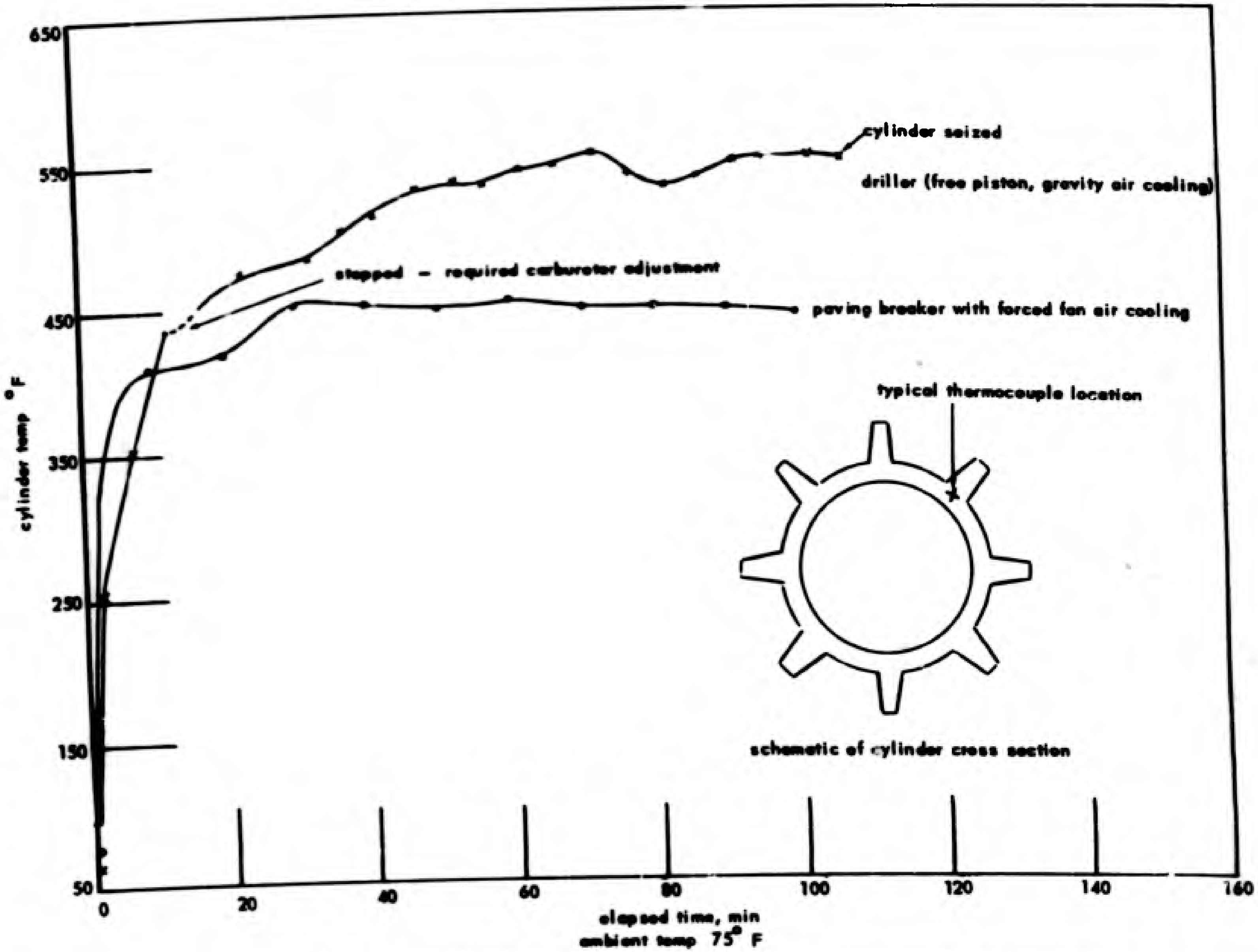


Figure 37. Graph of operating temperatures in two air-cooled engines with dissimilar conditions at air flow.

temperatures drop, although it is often a year-round problem in taxicabs, delivery trucks, and engines of other units used intermittently or subject to prolonged idling. The design of the cooling system and the effectiveness of crankcase ventilation can increase or decrease the severity of these troubles while the nature of the fuel can also influence them." pp 376-377.

"The service requirements in this classification are the most severe encountered in the operation of diesel engines. High load operation at high temperatures, design factors, especially supercharging or engine installation details causing unusually high temperatures within the engine, constitute severe service, as does intermittent operation at low temperatures since both promote wear and deposit formation. Cooling system and crankcase ventilating system design, also exhaust line arrangement, can aggravate or minimize the severity in either case. The use of high sulfur content fuels increases service severity with respect to wear and deposits in varying degree, depending upon design, maintenance and operating conditions, especially low temperatures. Hence, frequently their use is considered to constitute severe service," p 377.

In terms of the Society of Automotive Engineers' classifications, operation at temperatures down to zero-degree Fahrenheit are considered low temperature, -10 to -15 degrees Fahrenheit, extreme. It will be recognized that what might be intermediate duty for either spark or compression ignition engines at mild temperatures might be severe at sub-zero or even near-zero temperatures.

In terms of arctic operation, it has been observed by the author during 5 years of inspection of operating engines in vehicles, idling, and under load, that seldom does the engine reach a temperature sufficiently high to record any temperature rise on the dash temperature indicator, usually calibrated from 100 degrees Fahrenheit. Coolant temperatures of the order of 50 degrees Fahrenheit have been observed on idling engines. Unfortunately, idling is the rule rather than the exception. "Idling-stopped" operating records of vehicles and earth-moving equipment used during the winter 1951-1952 at the arctic test station showed that for every hour of useful operation typical equipment was idled for 2 or 3 hours. The difficulty or inconvenience of starting cold engines is a decided deterrent to their being shut down.

References 11 through 26, inclusive, discuss various phases of boiling heat transfer, both in engines and experimental laboratory equipment. Reference 27 relates the use of a high boiling point coolant with poor heat transfer characteristics, "Orsil Coolant", and advocates much higher cylinder area temperatures than those usually found in liquid-cooled engines. References 28 through 39, inclusive, discuss engine sludging and deposits, and various cures. The general conclusion is that sludging occurs with low engine temperatures at low ambient temperatures or light, intermittent loads. Conclusions are drawn that both crankcase sludge, wear, and engine cylinder deposits can be reduced by raising temperatures in appropriate points.

References 40 through 44, inclusive, discuss the use of heavy fuels in diesel engines, with special problems requiring fuel preheating and in one case, show a definite reduction in cylinder wear with an increase in coolant temperature from 165 to 200 degrees Fahrenheit. This particular area of burning cheaper, heavy fuels is currently attracting considerable attention and is the basis for much research. Excess engine wear is the current major problem, and this is demonstrated to be somewhat aided by raising cylinder temperatures.

References 45 through 47, inclusive, point out possible current design criteria available for preparation of optimum condensers.

Reference 48 gives comparative costs and power losses for current commercial boiling-cooling systems in stationary engines. In general, it presents a probable advantage of the boiling systems in the larger sizes of engines. No cost data on small, portable engines or production made systems for large units are available.

References 49 through 80, inclusive, present a wide spectrum on various engine temperatures as they affect fuel, parts life, wear, etc. From these and the data collected in the investigations reported here, the tentative optimum temperatures for various points presented earlier were compiled.

The probability of distortion with temperature variation is well described in references 81, 82, and 83. Taylor[81] arrives at the startling conclusion that for high cylinder loads, the desirable coolant temperatures are high. To quote, "Thus, with a given mean gas temperature, a high coolant temperature is desirable from the thermal-stress point of view." To expand the reasoning, thermal-stress is a function of the temperature drop across the metal wall. Therefore, to increase the outside (coolant side) wall temperature is to reduce the heat flow, reducing stress. Alcock[83] expands this to place a limiting upper temperature of about 480 to 530 degrees Fahrenheit because of lubricant oxidation. He states, "In large engines, the liners of which are thick for strength, there is often serious thermal-stress due to the temperature difference between the inside and outside." The same paper gives excellent theories and data on various cylinder and engine area temperatures difficult to measure.

An often-cited work[14] points out the increasing difficulty in cooling engines adequately with increasing size, culminating in the frequent need for water cooling of cylinders in very large sizes. Since cooling by boiling has in general most frequently been applied to these larger diesel engines, we may logically proceed to the hypothesis that application to smaller bore engines, 12 in. and below, will be more satisfactory than the highly successful stationary applications to very large engines.

Appendix B. THE MECHANISM

AD 163134

Armed Services Technical Information Agency

ARLINGTON HALL STATION
ARLINGTON 12 VIRGINIA

FOR
MICRO-CARD
CONTROL ONLY

3 OF 3

UNCLASSIFIED

APPENDIX B

HEAT TRANSFER TO BOILING LIQUID

The idea of promoting heat transfer from a solid surface to a fluid, be it a liquid or vapor, by increase in velocity, thereby increasing the turbulence and literally scouring the boundary layer to an infinitesimal thickness, is so general in our everyday life that it is difficult to consider that under some conditions this may not be the physical action occurring. The literature mathematically describes boiling in terms of wave theory which gives theoretical credence to the thought that this "high velocity" technique does not control. Figure 4 gave pictorial representation of possible pumping energies if this turbulent heat transfer induced by velocity were the only mechanism available. Once it is seen that the disturbance necessary to accomplish high heat transfer rates comes from "under" or "within" the stagnant layer, the possibility that high induced velocities may inhibit uniform cooling is suggested. Evidence in this direction is seen in the various data collected on forced cooling in the small single cylinder engine reported earlier, Figures 15 through 18. This is corroborated in the results of studies[3] reproduced as Figure 38. The maximum flux density is of the same order at varying velocities, but the flux density at which the curve becomes very steep, as predicted by boiling theory and experimental evidence, increases with increased velocity. In a device in which it is desired to obtain uniform metal temperatures and the maximum flux is well within the critical range (below that at which the surface becomes steam bound because of high steam production, poor removal, or both), it is, of course, desirable to operate as nearly as possible to a vertical line on a plot such as shown in Figure 6 or 38. On a vertical line, were it possible to operate on such, the surface temperature would be independent of heat flux and metal temperatures would be uniform except as they must vary to conduct the heat within the solid.

An extension of the above thinking indicates that in a "perfect" boiling-condensing arrangement from a design standpoint, velocities would be as low as practicable, perhaps with little or no water carry-over. The condensate and carry-over return would further be introduced

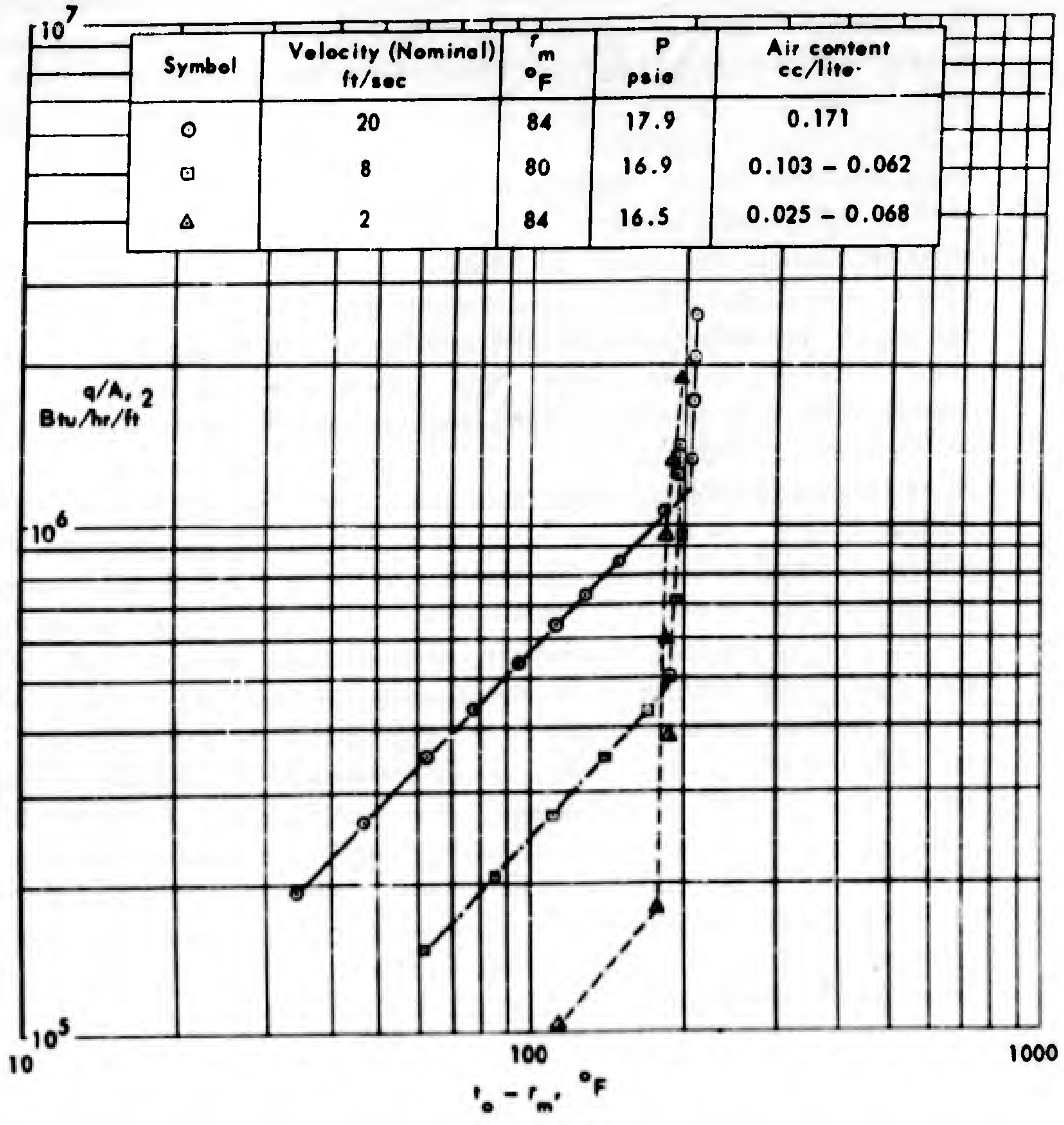


Figure 38. Heat flux density, q/A versus ($t_o - \tau_m$) for the nickel tube (from reference 3).

as high in the cylinder as feasible, insuring boiling in the lower portion and perhaps bringing the metal temperatures in the low heat flux areas not in contact with gaseous products nearer to those necessary to accomplish high heat transfer in the high-pressure combustion area. Circulation of the type necessary has been observed in large gas engines driving compressors in oil fields fitted with viewing windows on the cylinder walls (shown in Figure 39). This view is of the active upper cylinder area and was taken at a pumping installation of the Tidewater Associated Oil Field, Ventura, California. In this particular application, the coolant temperatures measured in the lower cylinder area were frequently observed to be higher than those in the active boiling area of the upper cylinder. The steam-water mixture discharged from the engine is shown in Figure 39A and the engine in Figure 39B. In terms of the above discussion, there appears to be merit in promoting poor heat transfer with attendant superheating above the saturation temperature in the lower cylinder areas. In terms of the type of plot shown in Figure 6, in an engine the heat transfer rates are so high that whether or not boiling is obvious, the upper cylinder area temperatures and heat flow will be controlled by some relationship of the sort shown in the upper portions of the curve, and cooling, to be uniform, must be promoted by the boiling mechanism over the entire cylinder length. This is the principal thesis leading to the conclusion that cooling by boiling must be the superior system in cooling the combustion-cylinder area per se. The ultimate in refinement would apparently be a system in which every effort is made to allow free, unrestricted boiling in the critical upper areas, meanwhile stifling the flow and, if possible, retarding the heat transfer in the lower cylinder areas. Point of entrance of the make-up return condensate and carry-over, holding it to a minimum, and restriction of the passage width in the noncritical areas appear to be evident methods of achieving optimum cooling.

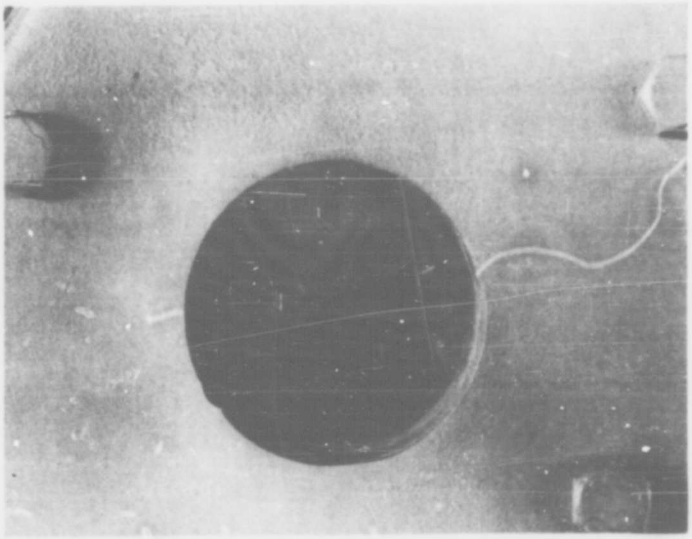


Figure 39. Boiling on upper cylinder wall of a large gas engine.

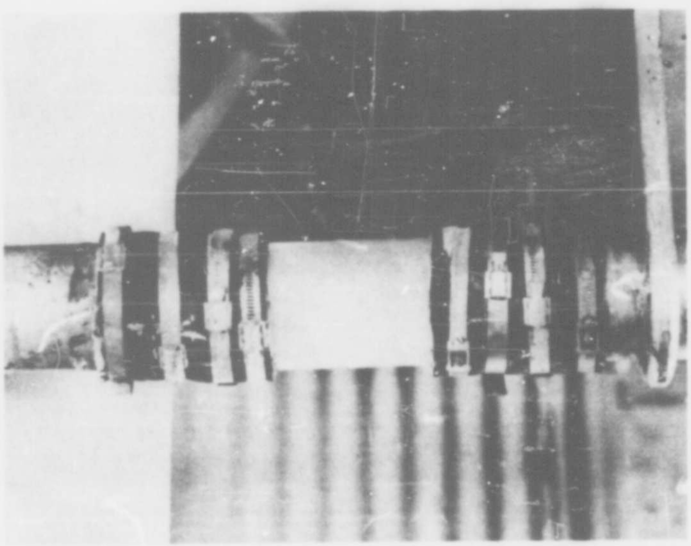


Figure 39A. Steam-water mixture in horizontal pipe (note separation).

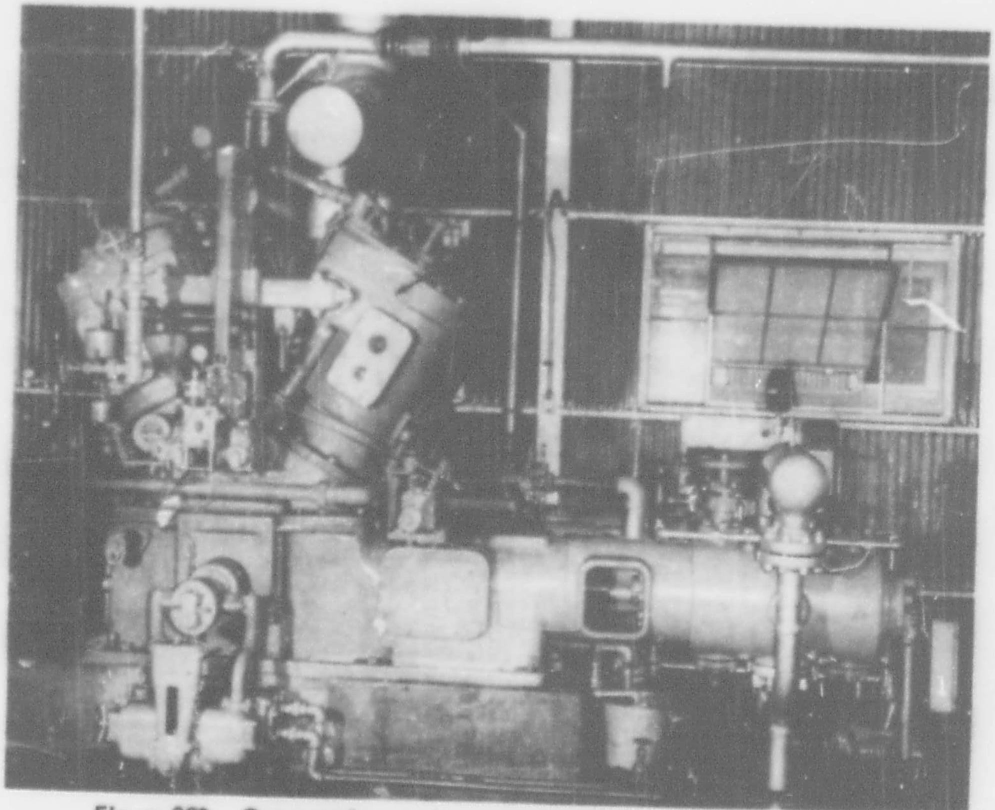


Figure 39B. Gas pumping engine cooled by boiling (note cylinder wall windows (A), and horizontal pipe insert (B).)

Appendix C. PECULIARITIES OF THE SYSTEM

APPENDIX C

VENTING

Unfortunately, liquid-cooled internal combustion engines with head gaskets tend to leak products of combustion into the coolant. These noncondensable vapors present a special problem in a condenser operating above atmospheric pressure, or even at atmospheric pressure if not properly arranged. In the simplest case of a conversion from a conventional engine using the original radiator and down flow of the steam, the radiator will be somewhat oversized for condensing because of the high heat transfer rates to the condensing wall and a higher temperature difference than originally existed. The trapped air and noncondensable gases will collect over the condensate, being heavier than steam and must be carefully vented. In a conversion using a radiator, now an oversized condenser, a simple tubular vent will suffice, see Figures 2A and 2B. In a more critical application, in which an attempt is made to reduce the size of the condenser, special ducting of the steam to collect the gases in a remote, low, part of the condenser will be necessary. They are then exhausted through a preferentially cooled section of the exchanger. In the boiling-condensing cooling system, see Figure 3, the steam has in some instances been baffled to travel upward through the front section of the heat exchanger; down through the rear tubes. The venting tube is connected to this rear section but placed in front where it receives the coldest air. This arrangement worked well at Yuma to 115 degrees Fahrenheit and at Fort Churchill to -35 degrees Fahrenheit. Another possible arrangement, in which alternate upward and downward passes in tubes closed at the top but vented into the condensate at each lower bend, is shown in Figure 40. If there is no fear of freezing, simpler arrangements with downward flow of the steam at all times are possible.

MINIMUM HEAD

In the usual gravity system wherein the change in density of one leg with heating causes a "thermosyphon" flow in the system, the greater the vertical legs, the greater the head and the faster the

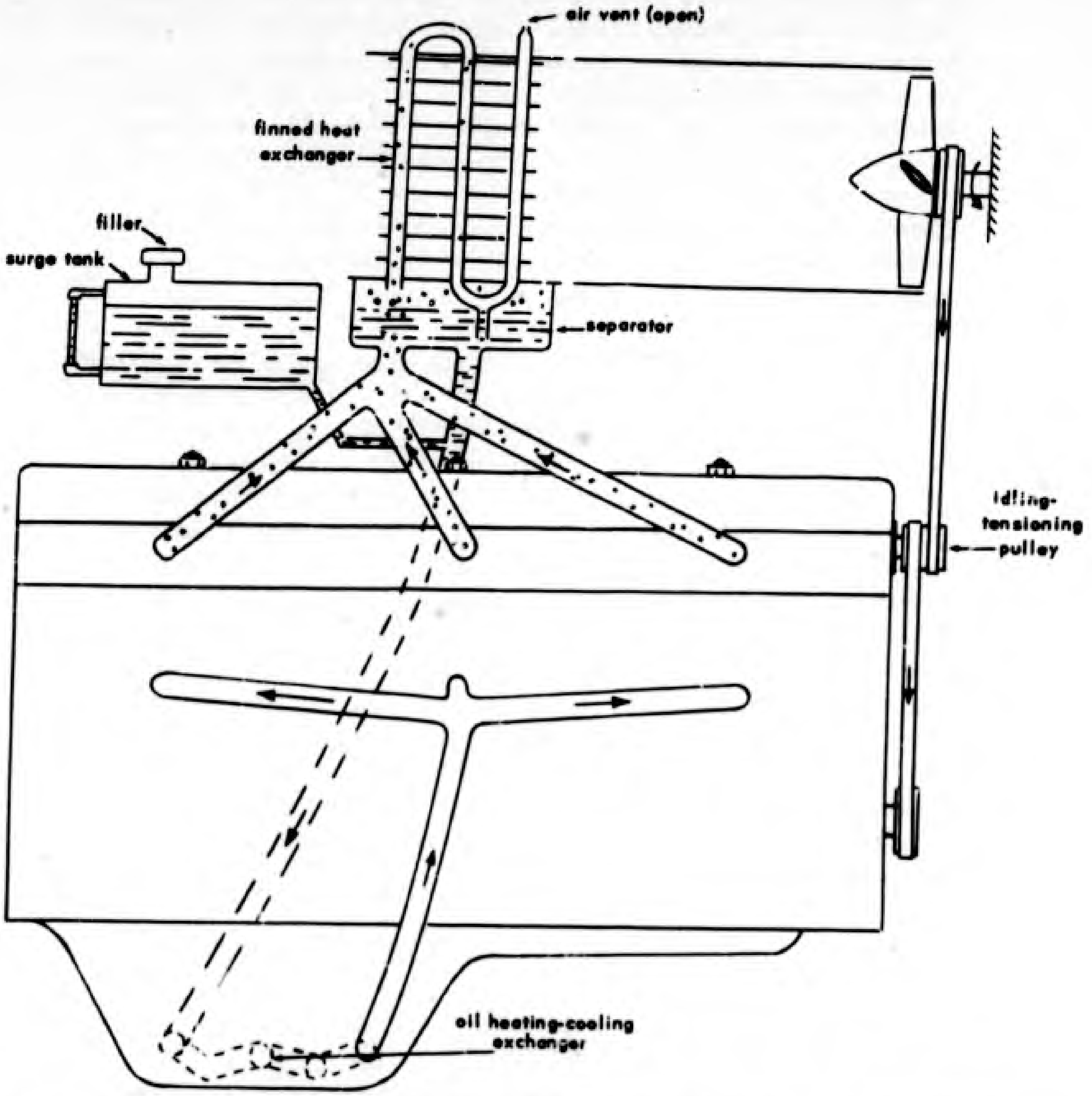


Figure 40. Schematic of idealized system for mobile equipment with limited vertical room. (Full automatic control without pumps, thermostats, etc.)

circulation. While it would appear that the same will hold true for a boiling-condensing system, and that the higher the separator, the more certain the flow, this has not been proven inevitably true. Extensive practice may show that the pressure drop in the steam-coolant leg is so great that raising the separator above a minimum of, say 12 inches, will be a minor, if any, benefit. A rational design based on pressure drops to achieve certain flows, for which the solution is quite difficult, appears to be of little importance or interest at this point, since it appears that little, if any, benefit accrues from higher than minimum coolant flow rates. In fact, it is basic to the heat transfer-by-boiling concept that no flow is necessary other than local displacement of steam bubbles by water as they are detached. The experiment of variable height did not indicate any better cooling with the additional static head, see Figure 21, over the minimum. In fact, temperature differential ΔT, along the cylinder wall was smaller with the low heads, presumably because the condensate and carry-over returned with less subcooling with short lines. This indicates that the optimum size of the condensate return line may be a compromise between a very large line to reduce the pressure drop, and a smaller line in which the time in transit and heat losses would be reduced. The possibility of allowing the steam mixture to traverse the outside annular space of a double pipe arrangement, with the returning condensate and carry-over preheated to full engine temperature (shown in Figure 41), should be considered for stationary installations or any installation where the condenser is remote from the engine. Some gain in performance should be noticed in such systems operating at slightly above atmospheric pressure in the engine but condensing at a lower pressure, since there would be a definite rise in temperature from the "flash" area to the relatively high temperature area of the engine, with a maximum temperature of the returning condensate.

PIPING SIZING

In considering the size of the steam-water piping, the possibility of considering the system as a steam pump with essentially the characteristics of an air lift pump is suggested. Efficiencies are listed[2] at relatively high heads of 300 ft of 50 percent and less with higher head. At the reduced heads involved, it will be conservative to assume 50 percent efficiency as a lift pump using the displacement work of the steam generated, that is, the pV portion of the increase in enthalphy. For the single cylinder engine discussed earlier producing 9-1/2 lb of wet

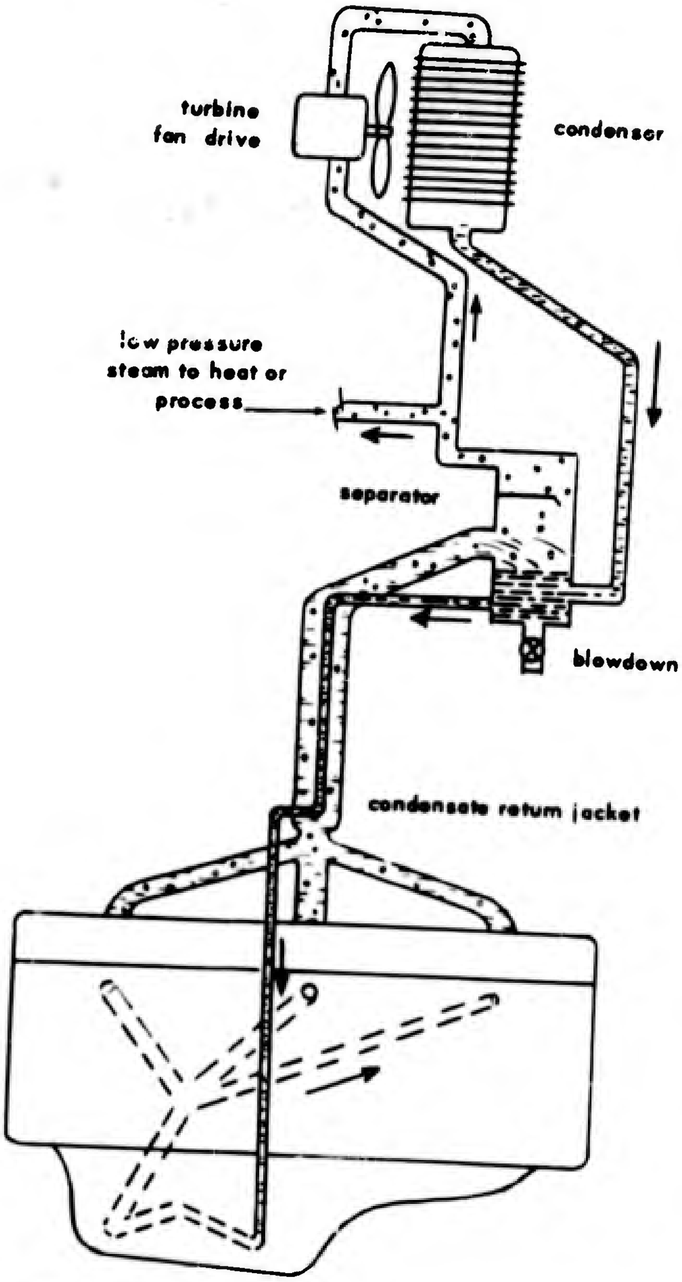


Figure 41. Schematic of idealized system for stationary or mobile applications with substantial vertical room.

steam per hour at an estimated 6.3 horsepower, or approximately 1-1/2 lb per hr per hour, the total volume of steam produced at atmospheric pressure is 254 cu ft per hour. The steam produced, 254 cu ft per hour, would be an ideally arranged situation, e.g., a "perfect percolator", be capable of displacing vertically or pumping a similar quantity of hot water weighing 59.8 lb per cu ft through 2 feet, doing 254 by 59.8 by 2, or 30,480 ft lb of work per hour. This would be equivalent to only 0.014 hp at 100 percent efficiency. Using a probable efficiency of 50 percent, according to Marks, 0.007 hp would be realized in a practical system (using the values for air lifts). The principal design criteria for a satisfactory air lift system is that the discharge line, in this case containing a mixture of steam and water, have a cross section area of:

$$\text{Discharge pipe cross section} = \frac{\text{GPM Liquid Pumped}}{12 \text{ to } 15} \text{ or} \qquad (11)$$

$$= \frac{254 \times 7.48}{2 \times 60 \times 15} = 1.06 \text{ sq in.}$$

This would be considerably larger than the internal cross section of the 5/8 in. inside diameter hose used in the single cylinder engine tests, so it may be concluded that the flow was somewhat restricted for the most efficient pumping system. Even so, there is no indication in the data of restriction leading to unsatisfactory cylinder metal temperatures, and no indication that maximum velocities should be sought. The extreme adaptability of the boiling-condensing process for the engine application is further illustrated. While beyond the scope of this paper to attempt to develop fully design techniques, an attractive possibility for determining the total coolant flow rate, condensate and carry-over, is suggested. In Figure 8, the manometer indicated (dotted) would measure directly the pressure drop in the return or "downcomer" to the engine, with suitable correction for differences in temperature. By running a short series of flow tests with measured volumes of water at 212 degrees Fahrenheit, the system can be directly calibrated. Measuring of the steam as discussed elsewhere would allow complete analysis of the system, and calculation of the pressure drop in the steam leg.

Appendix D. OPTIMUM DESIGNS

APPENDIX D

PROJECTED BOILING-CONDENSING SYSTEMS

While the "NAVCERELAB Design, Boiling-condensing System", utilizing gravity flow circulation and oil temperature conditioning toward the boiling point of the coolant is considered to be the ultimate in simplicity, practical variations having special features are suggested. For those applications in which overhead is limited, e.g., trucks, passenger cars, etc., and in which the vertical projection of the condenser, as in Figure 38, is not desirable, the unit suggested in Figure 42 is projected to have merit. This will be seen to be a

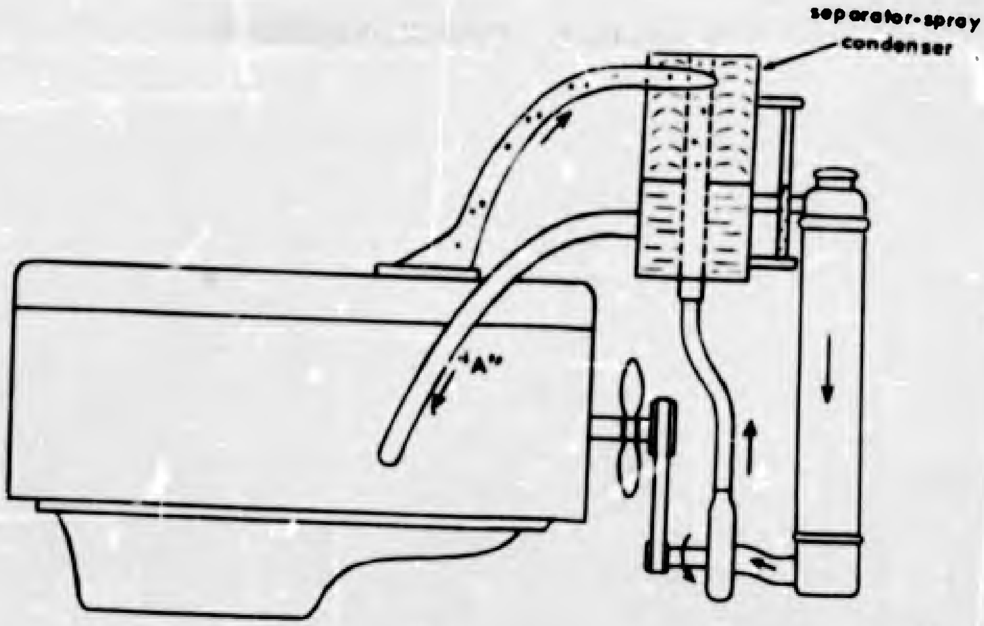


Figure 42. Schematic of limited overhead application of projected boiling-condensing systems.

simplification of the unit in Figure 35, which has a surface condenser and two coolant systems. By utilizing a spray condenser combined with the separator (if possible), the advantages of direct contact with the steam of the circulated coolant can be gained. A pump will, of course, be necessary. In this case, the coolant return to the engine, "A", will be subcooled, and the return line should be sized to achieve minimum cooling. Further, it should return as high as possible in the engine, in the manner shown in Figure 40, which shows the complete ideal system, with optimum fan efficiency, simple venting, separate surge tank for excess water, oil temperature conditioning, collection of steam from the points of generator, etc. The assumption is made there, though perhaps it is not justified, since the mechanical problems are not unsurmountable, that the incorporation of steam turbine, jacketed condensate return, take-offs for space heating, etc., are not justified in mobile equipment. Figure 41 shows a complete stationary system with these features as might be applied to a 6-cylinder engine, with coolant return and steam collection at the points of a maximum generation adjacent to the exhaust valve ports. This is the basis for the 3-part branch of the headers. In all of the above pressure release valves, make-up lines, drips, condensate return lines, etc., typical of conventional low-pressure steam practice are assumed and not shown for simplicity.

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