# A THEORETICAL INVESTIGATION OF THE POSSIBILITIES OF INTERNAL COOLING OF AIRCRAFT ENGINES BY WATER INJECTION TO THE CYLINDER.

By

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Introduction

The purpose of this paper is to investigate, theoretically, the several possibilities of internal cooling of engines by water injection to the cylinder, and the necessary condensor system for obtaining the required amount of wat er from the exhaust gases.

The radial air cooled engine is inherently more efficient in regards to weight than the inline type either air or liquid cooled. In both cases it has the advantage of a considerably lighter crank shaft , the heaviest individual component of a power plant, plus the radiator system in the case of the liquid cooled. With the development of the larger twin rows, it appears that but little more can be done with the type due to cooling difficulties. In service conditions the cooling of this type is barely satisfactory, necessitating in general the use of a considerably richer mixture than is dictated by either maximum power or economy in order that the incoming charge may absorb a portion of the excess heat. Thus the increase in power per unit for the radial engine depends on auxiliary cooling to allow an increase in rows of cylinders or an increase in compression ratio, with, of course, the common possibility of all engines of increased R.P.M. The applications discussed in this paper are considered only in connection with the radial type, although in general applicable to the inline types.

The three possibilities of water injection are:

(1)~~For~~ Toward the end of the power stroke thus decreasing the gas temperature during the exhaust stroke.

(2)Go to a six stroke cycle, two of which are made on a mixture of air and water particles.

(3)Injection of water with the charge of fuel.

The first two methods will be discussed as to resulting cooling, and the third in conjunction with the required condensing system.

The writer hastens to advise that his experience with airplane power plants is limited to use under service conditions and the study of books and reports on the subject. In this connection the two volumns of D.R. Pye are strongly recommended to those interested as being unique among technical books in regards to organization and presentation.

The Units Employed

Standard Cubic Foot - SCF - the quantity of gas which occupies 1 cubic foot at 1 atmospheric pressure and $0^{\circ}$ C.

Pound Molecule - MOL - is a weight in pounds of a gas equal to its molecular weight and occupying 359 Cu.ft. at standard conditions.

Specific heats will be expressed in foot pounds per SCF or CHU per MOL.

CHU - heat required to raise 1 lb. of water from $14^{\circ}$C to $15^{\circ}$C.

Fuel-Air Relation Employed in Calculations

Since the term gasoline covers a mixture of hydrocarbons varying greatly in number and relative proportion, for the purpose of calculations the fuel in this paper is considered to be composed of 50% each by volumn of Benzene ($C_6H_6$) and Heptane ($C_7H_{16}$). Their complete combustion by air is as follows.

$$2C_6H_6 + 15O_2 + 56.4N_2 = 12CO_2 + 6H_2O + 56.2N_2 + 1.52 \times 10^6 \text{ CHU}$$

| | | | |
|---|---|---|---|
| weights | 156 | 2060 | 2216 |
| volumns | 2 | 71.4 | 74.4 |

$$2C_7H_{16} + 22O_2 + 82.8N_2 = 14CO_2 + 16H_2O + 82.8N_2 + 2.155 \times 10^6 \text{ CHU}$$

| | | | |
|---|---|---|---|
| weights | 200 | 3020 | 3220 |
| volumns | 2 | 104.8 | 112.8 |

Total weight of fuel is 356, water formed = 22 x 18 = 396.

Thus the ratio of water formed to fuel burned is about 1.20. Average aviation gasoline runs about 1.35 lbs. of water to lb. of fuel.

## Actual Cycle Temperatures

To compute either the maximum temperature of the combustion stroke or the temperature of the gas at the time the exhaust valve opens for an actual case is almost impossible. The total heat energy liberated with a given fuel air ratio is well enough known but the effects of dissociation, i.e. the fact that at high temperatures carbon, carbon monoxide, hydrogen and oxygen may exist in certain proportions in equilibrium, the effects of heat transfer to the walls and the values of the specific heats at high temperatures all combine to prevent an analytical solution. By measurement, however, the maximum temperatures of the working substances in a modern engine vary between 2500°C. and 3000°C. and at the time the exhaust valve opens, normally 50° to 60° before the end of the combustion stroke, the gas is between 1200°C and 1400°C. When this has been expanded to about 15 lbs. absolute pressure its temperature varies from 600°C to 800°C. If a lean mixture is used the temperatures are lower during the combustion and higher during the exhaust stroke since the rate of combustion is reduced.

I. Injection of Water Towards End of Combustion Stroke.

Assuming the gas is at a temperature of 1400°C when water is injected, we can calculate with sufficient accuracy the temperature resulting for various amounts injected. Since the specific heats vary continously, strict accuracy would only result from a series of successive approximations. In this case mean values as shown are taken. Referring to the combustion equation:

| Gas | No. of MOL | | $C_p$(CHU/MOL) |
|---|---|---|---|
| $CO_2$ | 26 | x | 14.08 = 366 |
| $N_2$ | 139.2 | x | 7.35 = 1024 |
| $H_2O$ | 22 | x | 9.88 = 217 |
| Heat to be absorbed per degree drop in temperature | | | 1607 CHU |

Assume the water added is at $100^{\circ}C$. but has lost none of its latent heat, also that the water is converted to steam at 15 lbs.Abs., a conservative assumption since the heat energy absorbed increases with pressure. The latent heat in CHU per MOL is 9712 at 15 lbs. Abs., and for the range involved for the injected water the specific heat is taken as 9.05 CHU/MOL.

Let Tr be resulting temperature and W = no. of MOL of water added.

Then

$$1607(1400-Tr) = 9700W + W(Tr-100)9.05$$

$$\text{or } 177.5Tr + TrW = 248{,}500 - 972W$$

| | | | | | |
|---|---|---|---|---|---|
| W | 1 | 4 | 10 | 20 | 40 |
| Tr | 1385 | 1347 | 1273 | 1158 | 967 |
| Wt. of water / Wt. of fuel | .051 | .202 | .505 | 1.01 | 2.02 |

Effect of this Type of Injection on Component Cylinder Parts

Since by definition any successful engine will be designed up to the limits allowed by the cooling aspects, it is a corollary that any cooling will increase performance. However, for an airplane,performance is judged only on a weight basis. In an attempt to evaluate the cooling resulting from such an injection, reasonable assumptions could not be found from the general collection of test data. Thus, the following is the only method the writer could visualize.

If we assume that a specific temperature is known, which would give the same resulting wall temperatures under conditions of a steady state as are actually realized, the problem becomes one of merely equating heat transfers. This "potential" gas temperature will be a continuous variable throughout the length of the cylinder. The critical points are at the top of the cylinder wall, the head, the exhaust valve and the center of the piston. A consideration of the cycle indicates that excluding the exhaust valve, these parts would reflect approximately the same "steady state" temperature of the working substance.

Let $T_p$ be potential gas temperature for the top portion of the cylinder wall.

$T_{si}$ inside cylinder wall temperature at the top.

$T_{so}$ corresponding outer wall temperature.

$T_m$ temperature of the cooling air stream.

$Q$ the rate of heat transfer per unit area, assuming that all heat from the inner wall goes directly to the outer, i.e. no longitudinal transfer.

For transfer from gas to inner wall

$Q = C \rho V_m C_p (T_p - T_{si})$ where $C$ is a function of the Reynolds number, and $V_m$ the mean velocity.

From inner wall to outer wall

$Q = \frac{K}{t} (T_{si} - T_{so})$ where $K$ is a conductivity coefficient and $t$ the wall thickness.

From outer wall to air stream

$Q = C \rho V_m C_p (T_{so} - T_m)$ with constant defined as above.

Thus, the three equations result for any given physical setup in linear equations for the temperature differences multiplied by some constant. These constants in the cases of transfer between solids and gases, above, are to all purposes inpossible of theoretical determination.

Thus we have

$$Q = K_1 (T_p - T_{si})$$

$$Q = K_2 (T_{si} - T_{so})$$

$$Q = K_3 (T_{so} - T_m)$$

Thus $$Q = \frac{(K_1 + K_2 + K_3)}{3} \times (T_p - T_m)$$

$$\text{and } T_{si} = \frac{Q}{2}\left[\frac{-1}{K_1} + \frac{1}{K_2} + \frac{1}{K_3}\right] + \frac{T_p + T_m}{2}$$

$$\therefore T_{si} = \frac{T_p - T_m}{2}\left[\frac{K_1 + K_2 + K_3}{3}\right]\left[-\frac{1}{K_1} + \frac{1}{K_2} + \frac{1}{K_3}\right] + \frac{T_p + T_m}{2}$$

Calling the resultant constant K

$$T_{si} = \frac{T_p}{2}(K+1) + \frac{T_m}{2}(1-K)$$

With this relation if Tp can be determined from other consideration, the other temperatures being known, an idea as to the value of K may be obtained.

Solution for Tp.

As stated the heat transfer between the gas and the wall is given by the relation:

$Q = C\rho V_m C_p(T_p - T_{si})$ where C is a function of the Reynolds number, $\frac{V_m l}{\nu}$. If the flow is considered as being completely turbulent C will be proportional to $\frac{1}{R^{.15}}$. Neglecting the variations of ρ and Vm to .15 power we have

$$Q \backsim \rho V_m \Delta T$$

which checks well with experiment.

Figure 1 shows a plot of $\frac{\Delta T}{\Delta T_{max}}$ for the four stroke cycle in red. The assumptions for $\Delta T$ are that the maximum temperature is 2800° C., the exhaust stroke 1400°C. at the top of the wall or head temperature is running being 300°C. Vm is assumed to be that of the piston, and $\frac{V}{V_m}$ is plotted in black. At the time the exhaust valve opens, V is assumed to increase rapidly to Vmax and then fall off gradually to zero by the time the piston is again at top dead center and is shown in dashed black. The ratio of $\frac{\rho}{\rho_{max}}$ is shown in blue. Thus as stated above the product of these three variables should be proportional to the rate of heat transfer from the gas to the wall. In this case only the effects of the exhaust and combustion strokes were considered, the other two being negligible in comparison. The green curves give this product of the three variables, and the areas A and B will thus give the relative magnitude of the heat transfer from gas to wall for the two strokes. It is interesting to note that A:B::2:3. This indicates that the total heat transfer during the two strokes varies **approximately as** the mean temperature differences between the gas and the wall **during** the strokes, since these mean temperatures are themselves approximately of the ratio 3:2. In other words, the variations of density and velocity are such as to compensate each other. We can thus assume our steady state conditions such that Tp will be the arithmetical mean of the temperatures of the working substance throughout the cycle. If the effects of the intake and compression stroke were considered they would cause a reduction in Tp since, except in the latter portion of the compression stroke, there is a flow of heat from the wall to the working substance. Neglecting this effect we may consider Tp as defined as

$Tp = \left[\frac{Tmax + T_2}{2} + Tex\right] 1/4$ where $T_2$ is the exhaust temperature with no water added. We may thus say that a one degree drop in the exhaust gas due to cooling should give a 1/4 of a degree drop in Tp.

Determination of K.

Taking as assumed Tmax as 2800°C. and 1400°C. as the exhaust gas temperature. Since both of the final equations are expressed on temperature differences, it is immaterial whether temperatures are expressed on the absolute scale or not.

$$Tp = \left[\frac{2800 + 1400}{2} + 1400\right] 1/4$$

$$= 875^{\circ}C.$$

And, assuming we have a hot cylinder head of 300° or actually at the top of the cylinder wall with an outside air temperature of 15°C., we obtain for K

$$300 = \frac{875}{2}(K+1) + \frac{15}{2}(1-K)$$

$$K = -.337$$

$$\text{or } Tsi = Tp\ .33 + Tm\ .67$$

Since K is only a function of the physical setup and independent of the temperatures, it will remain constant with any internal cooling done.

Resultant Cooling of Head and Exhaust Valves.

It may thus be assumed that a 1° drop in the exhaust temperature will give .33 x .25 = .0825° drop in the temperature of the top of the cylinder wall or to what amounts to the same thing, the head.

Exhaust valve temperatures run much higher than any other portions of the cylinder, being on the order of 800°C., unless elaborate cooling devices are used. Since the temperature of the valve above that of adjacent portions is due solely to the exhaust gases, it is reasonable to assume that its temperature will decrease linearly with the exhaust temperature.

On the basis of the previously determined exhaust temperatures, the following table gives the resultant cooling.

| Wt of Water / Wt of fuel | Exhaust temp. | Head temp. | Exhaust Valve temp. |
|---|---|---|---|
| 0 | 1400°C. | 300°C. | 800°C. |
| .051 | 1385 | 298.8 | 792 |
| .202 | 1347 | 295.6 | 770 |
| .505 | 1273 | 289.5 | 728 |
| 1.01 | 1158 | 280.0 | 661 |
| 2.01 | 967 | 264.2 | 553 |

The drop in other portions of the cylinder would be proportionately a little greater than at the head from the same analysis. Also, it is believed that the drop in head temperature would be somewhat greater than the above since the presence of the very hot exhaust valve has an appreciable effect on the rest of the cylinder due both to conduction and radiation

As to what effect this cooling would have on power and fuel consumption no analytical analysis will apply since, if the materials would stand the temperature and the fuel not detonate, the hotter the better. As to test results, along this line, the following is quoted in substance from Pye. (Ref.1)

"In some experiments by Gibson the singleexhaust valve was artifically cooled by including a fluid within the hollow stem and designing a tiny radiator at the end of it, with the following results.

| Cylinder and valve size | RPM | Compression Ratio | Max BMEP | Petrol pint per BHP hour | Exhaust valve Temp | Valve |
|---|---|---|---|---|---|---|
| 5 1/2" x 6 1/2" | 1650 | 5 | 117 | .60 | 400°C. | Cooled |
| Valve 2.3"in dia. | | | 113.4 | .62 | 700°C | Uncooled |

Thus the maximum power was increased about 3% with a simultaneous reduction in fuel consumption of 3.3%. Furthermore, the maximum power obtainable uncooled could be maintained with a mixture so much weaker as to reduce the fuel consumption by 8%. The mean temperature of the cylinder head was lowered 26°C. and the hottest point on the piston by 25°C."

Conclusions Relative to This Type of Injection.

An analysis of obtaining water from exhaust gas condensation will be undertaken later in the paper, but even without this it would seem that this method would not be applicable to aircraft engines. Even though the cooling effects are desirable it seems that the required amount of water would make this method of obtaining cooling uneconomical. If the water content could be separated from the rest of the exhaust products and so condensed the weight of the required installation would be greatly reduced.

Separation of the Water Content from the Rest of the Exhaust Products.

The possibility of such a separation exists in the high velocity centrifuge. If the mixture could be rotated at velocities of the order of the mean molecular velocities which are themselves of the order of the speed of sound, an appreciable separation could be obtained. The situation is ideal for this application since the molecular weight of water is 18 whereas the next lightest molecule is nitrogen whose weight is 28. In other fields separations are obtained in this manner, and the rotor of the present turbo-super charger reaches velocities of the required order. It would seem that the development of such a process may be accomplished if the value to be obtained warrants the effort.

II. Cooling By a Six Stroke Cycle.

By following up the exhaust stroke with an air intake to which a finely divided water spray was added would be a most effective method of absorbing the heat from the cylinder wall, especially if the air intake were so located that the incoming air were directed perpendicular to the cylinder axis, thus giving a high swirl.

The same intake under the above conditions would be satisfactory for the fuel-air mixture. The reason that such would be undesirable in present type engines is that it is desired to have the charge pick up as little of the wall heat as possible. Aside from the reducing of the wall temperatures this method would have another important advantage in that the temperature of the residual gases left to mix with the fresh charge very much lower giving a much lower pre-compression temperature as some later calculations will show. Aside from the effect on the detonation, this would mean that the whole temperature cycle is reduced which gives two distinct advantages. First, the Carnot* cycle efficiency is automatically increased by the drop in $T_1$ if the numerator remains constant. Secondly, due to the very rapid increase of both dissociation and the specific heats at the higher temperatures this shift of the cycle tends to increase the spread between the maximum and minimum temperatures. The above gain in efficiency is borne out in the "hit and miss" type of engine which gives the highest thermal efficiencies of any type. In this method of speed control a governor closes the gas inlet valve for a stroke or two, the cylinder working on air only.

Being unable to find any test data applicable upon which to base calculations it is possible to make only some general observations as to the value of such a scheme. In general you would be dissipating the same amount of heat as the conventional water jacket and radiator system. Best efficiency would be obtained if the steam air mixture were recaptured without mixing with exhaust products. This could be done by keeping the exhaust valve closed forcing the mixture out through the same valve it entered and on to a condensor. If it were to be ejected through the exhaust valve a "Y" valve system would be used. Either method would add to the complication of the cam and valve mechanism, but does not present any greater technical difficulties than many other features.

The advantages of this would be that: (1) the same total amount of heat dissipation would result in a much lower cylinder temperature. (2) The power plant could be completely cowled, reducing considerably the total drag while, by proper design, the condensor drag could be made negligible. (3) All of

*Carnot eff. = $\frac{T_1 - T_2}{T_1}$

the auxiliary engine components such as magnetos, generators, pumps, etc. would operate without present temperature difficulties. (4) A considerable increase in economy plus an increase of the maximum power per stroke. (5) This method if practical would allow an indefinite increase in rows of cylinders.

The obvious disadvantages would be a decrease in maximum power of somewhat less than one third on the basis of present types, and the condensor system. It is possible that other factors such as displacement and RPM could be increased to improve the power output. A test of this principal would not be difficult and might prove fruitful.

It has been suggested that either this method or the previous might result in the formation of cracks in the walls and pistons. While no test data is available it does not seem that this would be likely, since the temperature variations would be no greater than those which are experienced in starting. This is because under throttled operation only the maximum pressure not the maximum temperature is appreciably reduced.

III. Water Injection for Detonation Suppression at Higher Compression Ratios.

The thermal efficiency of an internal combustion engine is a direct function of the compression ratio, or actually the expansion ratio at which it may be operated. Thus for a gasoline engine this is definitely limited by the fact that even the best grades of fuel with an optimum amount of tetra-ethyl lead, detonate at CRs between 6 and 7. This results in practical engines of indicated thermal efficiencies of from 31% to 35%. A curve of indicated thermal efficiencies against CR is shown in Fig. 2, both on the basis of the so called air standard efficiencies which consider the working substance to be a perfect gas with no heat loss and also for the best actual test engines. The latter is extropolated by the generally accepted experimentally obtained relation, efficiency $= 1-\left(\frac{1}{CR}\right)^{.256}$ as given by Pye (Ref. 1) for volatile fuel engines. It will be noted that the theoretical and experimental curves are of the same type differing by a constant amount. There are also shown experimental points for two test diesels operating at two different load conditions.

At this point it is believed desirable to explain the fundamental differences in the problem between the volatile fuel engine and the diesel type. While with the latter, indicated thermal efficiencies well above the experimental curve K-M may be obtained at high air to fuel ratios and low RPM, it has been found impossible to burn more than 75% of the air in a high speed diesel. This is because a more complete mixing of the fuel and air cannot be obtained in the time available. Any further injection of fuel in an attempt to burn more air results only in a production of black smoke with no appreciable increase in power. This limitation would obviously not apply to the volatile fuel type which experiences no particular difficulty on that score. Another limitation which would apply equally to both types of engines at high CRs is that of maximum pressure. At a CR of 12:1 the diesel on a basis of 75% of the air burned at close to constant volumn would give maximum pressures of about 1200-1300 lbs. per sq. inch, while the volatile fuel engine would develop pressures of the order of 1500 to 1600 lbs. per sq. inch. The diesel operating at high CRs can avoid the high pressures with a consequent reduction in efficiency by delaying the fuel injection until later in the stroke. The volatile fuel engine would be afforded no such relief.

Theory of Detonation

As indicated the major problem in increasing the thermal efficiency of a gasoline engine by increase of CR is the phenomena of detonation. Since this first became a problem a vast amount of experimental work has given rise to several theories as to the actual cause. Basically these are the same but they vary widely in attempting to explain associated phenomena in connection with "dopes". The theory which appears to the writer as offering the best physical explanation in the light of the various tests is that offered by Egerton and reviewed by Pye in Vol.1(Ref.1) and of which the following is a brief summary.

When no detonation takes place the flame started by the spark spreads out steadily until the fuel-air mixture is consumed with a speed which is a function of turbulence, the fuel itself and the pre-combustion temperature. When the pre-combustion temperature is such that only a slight increase in temperature is required to reach the spontaneous ignition temperature of the mixture, the increase in pressure of that portion near the spark plug may be sufficient to cause the rest of the mixture to ignite spontaneously giving very high local pressures. Pistons have actually been broken from that cause. It was soon discovered that various substances would delay the onset of detonation when mixed with the fuel in very small proportions, and that there were definite optimum mixtures in each case. By far the most effective is what is commonly called "tetra-ethyl lead", which is a mixture of lead ethicle, Pb $(C_2H_5)_4$, and ethylene dibromide, $C_2H_4Br_2$, the latter being added for the purpose of preventing excessive lead deposits.

In the nuclear drop theory the value of this dope through lack of more complete knowledge is referred to as an "inhibitor of chemical reaction". By various tests it has been determined that at temperatures considerably below that of ignition most fuel-air mixtures undergo low order oxidations with the formation of highly unstable peroxides. These peroxides have a lower ignition temperature and when reached give a very rapid release of energy. The formation of these peroxides is considered to be the result of a reactive chain set up in the higher energy molecules of fuel and oxygen, each peroxide molecule formed inducing the formation of more such molecules. Without going through the chemical line of reasoning the presence of the lead ethide completely suppresses the formation of these low temperature partial oxidations by breaking up the reactive chains. Thus it can be seen in theory why leaded fuel can prevent detonation only up to a certain point, a fact every one who has driven a hot motor realized in practice.

We thus arrive at the point, that to increase indefinitely the onset of detonation the temperature, at the time of the spark passage of the mixture must be reduced to the point that the temperature of spontaneous ignition will not be reached before the flame front has covered the major portion of the combustion chamber. The most practical method of doing this is by the addition of water to the fuel-air mixture, its very high latent heat of vaporization will appreciably decrease the temperature rise during compression. Thus the water to be really effective must not be added as vapor but as finely divided water particles.

In a water injection test which is not available at present to the writer, the Army Air Corps Engineering Section, operated a test engine at BMEP of 579 lbs. per sq. inch, an almost inconceivable figure. Aircraft engines at present operate at a maximum of the order of 140 lbs. BMEP.

Required Amounts of Water to Permit Operation at Various Compression Ratios.

On the basis just discussed is now presented a series of computations to determine the amount of water necessary to prevent detonation to successively increased CRs.

The first phase of this has to do with a determination of the maximum temperature of compression, taking into account the residual exhaust gas left present when the fresh charge is drawn in. A conservative note is added by taking the temperature of the residual gas at 800°C. each time, since this temperature would be continuously lower.

Assumptions:

1. A cylinder with a swept volumn of 1 cubic foot.

2. "N" as recommended by Pye as 1.33.

3. Kv (volumetric heat of specific of a given volumn)

   Kv = 23.5 ft. lbs. per SCF for products of combustion for the range up to 800°C.

   Kv = 20.5 ft. lbs. per SCF for fuel-air mixture up to 100°C.

4. 60°C. as fuel-air temperature upon entering the cylinder but before mixing with the residual exhaust gases.

| CR | Vol. of Exhaust gas at 800°C | Vol. of Exhaust gas at 0°C | Vol. fresh charge-0°C | Temp. before Compression | Max. Temp of Compression |
|---|---|---|---|---|---|
| 6 | .20 | .051 | .82 | 109 °C | 419 °C |
| 7 | .116 | .042 | .82 | 102 | 440 |
| 8 | .143 | .036 | .82 | 96 | 461 |
| 9 | .125 | .032 | .82 | 92 | 481 |
| 10 | .111 | .028 | .82 | 88 | 500 |
| 11 | .100 | .025 | .82 | 85 | 518 |
| 12 | .091 | .023 | .82 | 83.5 | 535 |

Now if we assume a given fuel operating at a CR of 6:1 without detonation, then by adding water to the fuel-air mixture at higher CRs sufficient to bring the maximum temperature of compression to that of 6:1 there should be no detonation. This assumes that the injected water is completely mixed and vaporized in the time allowable. Whether this will be true or not, theory cannot reasonably state. However, vaporization in other types of apparatus namely, the cloud chamber, would seem to indicate that the vaporization would be complete. However this should be one of the first experiments in case the theory advanced in this paper is given an experimental test.

It is necessary to make some simplifying assumptions for this calculation as an exact method would be quite tedious. The basis used is that the water is added after the compression is completed and that the water all changes state at a pressure equal to that at the end of the compression stroke. Any reasonable variation from the above will alter the results only by a very small amount.

The fuel-air mixture used is that given at the first of the paper. It being easier to work in MOL, the change is made, and the computation for a CR of 7:1 is as follows.

From steam tables it is found that one MOL of water will absorb 10157 CHU in going from water at 100°C. to steam under a pressure of 160 lbs. the temperature of the change of state being approximately 184°C. The volumetric heat of steam in that range is about 6.62 CHU per MOL. Thus

$$10157\ W + 6.62\ (420-184)\ W = 105$$

where W = number of MOL of water required.

$$W = .00895$$

The number of MOL of fuel in the fuel-air mixture used is

$$\frac{4}{180.2} = .0222.$$

The molecular weight of $C_6H_6$ = 78 and of $C_7H_{16}$ = 100, thus the mean molecular weight of the assumed fuel, 50% of each by volumn is 89 and the total weight of fuel in a MOL of the fuel-air mixture is 1.98 lbs. Also .00895 MOL of water is .161 lbs.

Thus the ratio of the weight of water required to the weight of fuel used is .0817. Following is the result for the various assumed CR's.

| CR | CR | Comp. Pressure | Max. temp. of Comp.— 420° | CHU/MOL to be absorbed | Wt. of water required to fuel used |
|---|---|---|---|---|---|
| 6 | 10.84 | 159 | 0° | 0 | |
| 7 | 13.3 | 196 | 20° | 105 | .0817 |
| 8 | 15.85 | 233 | 41° | 215.5 | .168 |
| 9 | 18.6 | 274 | 61° | 320.3 | .251 |
| 10 | 21.4 | 314 | 80° | 420 | .330 |
| 11 | 24.2 | 356 | 98° | 515 | .401 |
| 12 | 27.2 | 400 | 115° | 604 | .471 |

Cooling Problem with Increased CRs.

The increase in CR automatically solves the cooling problem since the mean temperatures of both the combustion and exhaust strokes are lower. This is exemplified by the following test quoted in substance from Pye (Ref. 1)

"As regards the effect of CR, Gibson's results show a quite regular drop in the fraction of Horse Power to cooling over rated BHP of from .83 to .69 for a series of CRs of 4.5, 5.0, 5.5 and 6 to 1, all tests being at 1600 RPM. The reduction of heat to cooling water at higher ratios will be chiefly in that part lost during the exhaust stroke. The higher the ratio of expansion, the more efficient will be the cycle, and therefore the cooler the working fluid when the exhaust valve opens."

The "potential temperature" analysis might be applied to this case, but the determination of the exhaust gas temperatures analytically is a difficult and dubious process. This temperature in the previous case was assumed from actual engine operation.

The real problem in this case is the weight of the condensor necessary, so we will now try to make some reasonable estimates on that score.

IV. The Condensor System.

As every one knows the later dirigibles and blimps used condensors to obtain the water content of the exhaust gases in order to obviate the valving of its expensive helium. For these craft the condensors were extremely heavy, those on the Akron weighing about 2000 lbs. per engine including attaching structure and connections. As compared to this the problem for the airplane can be solved with a small fraction of this weight. In the first place the boundary layer around the dirigible is very thick and it was found on the Akron that the condensors were operating in a field of greatly reduced velocity as compared to the craft's speed. Secondly, in the case of the dirigible it was necessary to act on the entire exhaust products if recovery of 100% on the basis of fuel consumed was to be expected. In any of the methods herein contemplated the water added to the working substance makes the recovery that much more economical for any given required amount, since the total amount of heat in the exhaust products is a constant for a given set up, though of course at a lower mean temperature differential. Thirdly, the attachment to the structure in the case of the airplane is much more simple.

Since any such system if applicable at all would be particularly advantageous in long range operation such as that carried out by planes of the patrol boat, trans-oceanic commercial boat or long ranged bombardment types, the ideal condensor location would be in the wing section. In the larger contemplated types there will be a depth at the region between the two motors of a four motored ship of around five feet at the maximum point, thus allowing ample room. The condensor structure could be designed to support the stiffeners and to transmit the torsional load. Thus the design problem from a structural view point while difficult should not be critically so.

In Fig. 3 is shown a schematic drawing of what the writer believes would be the most efficient type of condensor. The theory of the ducted radiator has received much attention of late and indicates that if the velocity through the radiator is sufficiently reduced below that of the free stream, the heat energy added may be made to equal or even exceed the energy removed from the stream thus reducing the drag of the system to a negligible figure and offering the theoretical possibility of a thrust. The basic theory is covered in Rand M 1683 and 1702. In experimental test the drag has been reduced to a negligible figure but to the writer's knowledge no thrust has yet been developed.

In order to get some idea of just what an outlay a condensor system would be for various required recoveries the writer used as a basis the work reported in ACIC No. 44 of May 1, 1924, "Condensation of Water from Engine Exhaust for Airship Ballasting". This was used rather than some of the more recent types of installation since the velocity was known, thus no boundary layer effects of the airship had to be considered. This appears to have been the first systematic tests made in regards to this problem and extensive service tests were made. The Model I which is used as a basis of extropolation was nothing more than a series of 1" pipes 60 feet in length and arranged in three banks thus giving an over all length of 20 feet, with the necessary headers and separator. The total weight of the pipes alone was about 400lbs. and with headers and separator about 450lbs. The power plant was a Liberty engine which with the propellor used was calibrated at 307 HP at 1600RPM. The result of a 90 hour test is summarized as follows:

| | |
|---|---|
| Average air speed entering condensor | 48 MPH |
| " " temperature entering condensor | 15°C. |
| Total weight of fuel used | 15,075 lbs. |
| " " " water collected | 13,943 lbs. |
| Water collected on basis of fuel used | 92.5% |

It was further stated that due to various leaks considerable water was lost, the amount not being estimated. The condensor was suspended at a slight angle to the horizontal, the longer dimension being in the direction of flight. With that type of condensor there must have been a very considerable loss of cooling efficiency by the very geometry of the system, also the unsupported tube lengths were 10 feet, making the tube strength and consequently the weight considerably greater than would otherwise be necessary. From this and boundary layer considerations along the tubes it is believed a much more effective system could be worked out and a very conservative assumption would seem to be that weight of the conducting surfaces could be reduced from 1.3 lbs. per BHPcondensed to 1 lb. for identical performance.

Since long ranged aircraft operations is tending towards higher altitudes, the condensor system should be prepared to operate effectively at 20,000 feet at least. Thus we will attempt to extropolate these test results to that condition, knowing that if successful their satisfactory operation is automatically assured at lower altitudes.

Effect of Velocity Variation

In the matter of velocity through the condensor the following is assumed.

If the velocity is kept at from 1/2 to 1/3 of that of the free stream, theory indicates (Rand M 1683) that the drag will be negligible with flight velocities of the order of 200MPH. Thus the expanded velocity is taken as 72 MPH. Since the rate of heat dissipation for this type of flow varies experimentally with $(V)^{.75}$ approximately the increase of effectiveness due to this velocity assumption on a weight basis is $\left(\frac{48}{72}\right)^{.75} = .738$. If we had assumed a velocity of 100MPH as we probably well could with an almost negligible increase in drag this factor would be .622.

Effect of Temperature Variation.

The extropolation for temperature differance in the cooling airstream is a bit more difficult in this case. In the various tests of the apparatus the temperature variation along the length of the tubes was not measured, thus it is impossible to give a mean operating temperature ~~for operating~~ as a function of the total dissipating surface. If this mean temperature were given the correction for outside air temperature would be very simple. It seems quite apparent that this mean temperature would be very much smaller tham the arithmetical mean, since after the water has started to condense the amount of heat transfer will be greater than previously for a given temperature drop. In other words, a considerable linear proportion of the condensor will be dissipating heat at temperatures of less than 100°C. This fact makes the effect of the drop of the outside air temperature of 40°C. between the surface and 20,000 feet result in a very large increase in the effectiveness of the condensor. For example, that portion where the temperature is 100°C. would have its rate of dissipation increased by the following factor $\frac{373-248}{373-288} = \frac{135}{288} = 1.59$ and in increasing amounts at the lower temperatures. While it can thus be seen that the temperature decrease of the outside air would be very effective to the condensor operation, not knowing the linear variation another method must be used to obtain an analytical expression.

A study of the variation of the final exhaust temperature with outside air temperature in the report above mentioned indicates that the mean effect of a $1^{\circ}$ drop in outside air is to give a little greater than a $1^{\circ}$ drop in the final exhaust temperature. In the test cited the mean exhaust temperature was $35^{\circ}$C. and the cooling air $15^{\circ}$C., the drop furthermore seemed to be accentuated at the lower air temperatures. Thus if it is assumed that the $40^{\circ}$ drop in outside air will give a decrease in final exhaust temperature of $20^{\circ}$ we may consider this to also include the variation of mixing ratio with temperature.

The Effect of Velocity and Density Variation with Altitude.

If the velocity through the condensor is allowed to increase in proportion to the plane's velocity the effect on the drag of the condensor will be slightly negative. In a plane equipped with turbo super charger and constant speed propellors the variation of velocity with altitude may be expressed as $\frac{V_\sigma}{V_0} = \frac{1}{\sigma^{1/3}}$ up to the critical altitude. The effect of velocity in regard to heat dissipation is expressed by $\left(\frac{V_\sigma}{V_0}\right)^{.75}$ approximately. The heat transfer will vary inversely with the density, thus the factor for the combustion of these two will be $\sigma\left(\frac{V_\sigma}{V_0}\right)^{.75} = \sigma\left(\frac{1}{\sigma^{1/3}}\right)^{.75} = \sigma^{.75}$

Since $\sigma$ at 20,000 feet is .532 the factor of effectiveness will be .614.

Combining the various factors with the originally assumed weight of 1 lb. per BHP condensed $\frac{1 \times .738}{.614} = 1.2$ lbs. per BHP condensed at 20,000 feet with a final exhaust gas temperature of 15°C.

Thus at 15°C. with mixture ratio of 15, .149 lbs. of water per pound of fuel burned will be carried out with the saturated exhaust gas. This is a function of the partial pressure of water vapor, the calculations being shown in ACIC No.44. As previously assumed combustion of each pound of the average aviation type gasoline gives 1.35 lbs. of water, considering the air coming through the carburetor as dry, which makes the amount of water recovered about 1.20 lbs. plus any water injected per pound of fuel burned.

At this time it should be stated that the water collected will contain considerable carbon or soot and also small amounts of sulphuric and hydrobromic acids. This will necessitate the installation of mechanical and chemical filter but no particular problem should result from this score. Another practical difficulty would be the protecting of the water lines, etc. from freezing.

<u>Evaluation of the Complete System.</u>

From the previous analysis it is possible now to determine the weight added by the condensor system and balance this against the reduction in fuel consumption.

No attempt is made to estimate the maximum power which might be developed from an individual unit as this will be a function of too many other factors. In each case it is assumed that a 1000 HP engine is developed, the following assumptions to cover any increase in weight with the increase in Maximum pressure. It seems probable that with sufficient development this increase in weight would disappear or even go negative. While the increase in maximum pressure will require an increase in bearing sizes and wall strengths with constant power, the cubic dimensions will decrease. To be sure of remaining conservative on this score, however, .05 lbs. per BHP is added for each unit increase CR above 6 at which the specific weight is taken as 2 lbs. per BHP developed. The still born Packard Diesel is covered by this allowance since it weighed 2.27 lbs.per BHP and operated at a CR of 12. Also an allowance is made for the weight of the injectors, connections and separator, etc. separately as shown.

| CR | %of Exhaust to be cooled | total st. of cooling surface required | Increase in wt. of engine | Injectors Connections, etc. | Total increase |
|---|---|---|---|---|---|
| 7 | 6.3 | 76 | 50 | 100 | 226 |
| 8 | 12.3 | 148 | 100 | 120 | 368 |
| 9 | 17.3 | 208 | 150 | 140 | 498 |
| 10 | 21.3 | 256 | 200 | 160 | 616 |
| 11 | 25.0 | 300 | 250 | 180 | 730 |
| 12 | 28.2 | 338 | 300 | 200 | 838 |

From this the following table indicates the effect on range. The thermal efficiencies are based on an extropolated experimental curve previously discussed. The brake thermal efficiencies would all be somewhat lower but since the ratios should all be the same it makes no difference in the result as the comparisons are reduced to ratios with the specific fuel consumption for the 6:1 engine taken as .40 lbs. per BHP hour. The type of operation assumed is maximum patrol boat range which at present would give a fuel load of 6000 lbs. per 1000HP engine. Any other specific type of operation could be worked for given conditions.

| CR | Indicated thermal efficiency | %increase over that at CR=6:1 $\eta = 1-\left(\frac{1}{CR}\right)^{.236}$ | Consumption per BHP hour | Flight hrs to pay for installation at .75 pwr. | Total wt. of fuel carried per engine | Total flight hours .75 pwr. | % increase |
|---|---|---|---|---|---|---|---|
| 6 | .335 | | .40 | | 6000 | 20.00 | |
| 7 | .368 | 9.85 | .361 | 7.72 | 5774 | 21.20 | 6. |
| 8 | .389 | 16.1 | .336 | 7.66 | 5632 | 22.35 | 11.7 |
| 9 | .405 | 20.9 | .317 | 8.00 | 5502 | 23.20 | 16.0 |
| 10 | .420 | 25.4 | .299 | 8.15 | 5384 | 24.00 | 20.0 |
| 11 | .433 | 29.2 | .283 | 8.32 | 5270 | 24.85 | 24.2 |
| 12 | .445 | 32.8 | .269 | 8.54 | 5162 | 25.60 | 28.0 |

<u>Conclusion</u>.

As anyone who takes the time to follow through this analysis must very forcibly realize many somewhat dubious assumptions have had to be made. While in such cases an attempt has been made to keep well to the conservative side the writer has a very high respect for the laws of perversity and consequently feels rather pessimistic as to the ultimate value of the methods considered. It is believed, however, that a sufficient theoretical value exists to warrant the experimental check of the indicated possibilities. It is not anticipated that the required preliminary experimental work would be particularly costly. As regard to detonation suppression by water injection, it would seem that test engine measurements could be readily taken, if not already in existence. The preliminary condensor work could all be done in a wind tunnel with an easily constructed scale section, and the heat transfer determined by a step by step integration after the pressure drop and velocity distribution has been obtained.

Following is a brief summary of salient considerations.

1. Water injection for cooling of exhaust gas only does not seem to be feasible unless the water content could be separated from the rest of the exhaust products before condensation.

2. No evaluation can be made of the six stroke cycle without more experimental data.

3. Water injection for detonation suppression seems to offer reasonable possibilities of ultimate value.

4. In any condensation method at high altitude the gas to be operated on must be kept at approximately 15 lbs. absolute pressure as the exhaust gas will carry out too high a water content at reduced pressures in the saturated state. This means that the fraction condensed must by pass the turbo super charger.

5. The increase in operating compression ratios automatically solves the cooling problem of the engine itself.

6. The condensor system can be operated with a negligible increase in total drag.

VARIATION DURING CYCLE

OF

$\frac{\Delta T}{\Delta T_{max}}$ $\frac{\rho}{\rho_{max}}$ $\frac{V}{V_{max}}$

AND

PRODUCT OF THESE THREE

$\frac{\Delta T}{\Delta T_{max}}$

$\frac{\rho}{\rho_{max}}$

$\frac{V}{V_{max}}$ (piston speed)

$\frac{V}{V_{max}}$ (exhaust valve open)

$\frac{V}{V_{max}} \times \frac{\rho}{\rho_{max}} \times \frac{\Delta T}{\Delta T_{max}}$


Fig 1

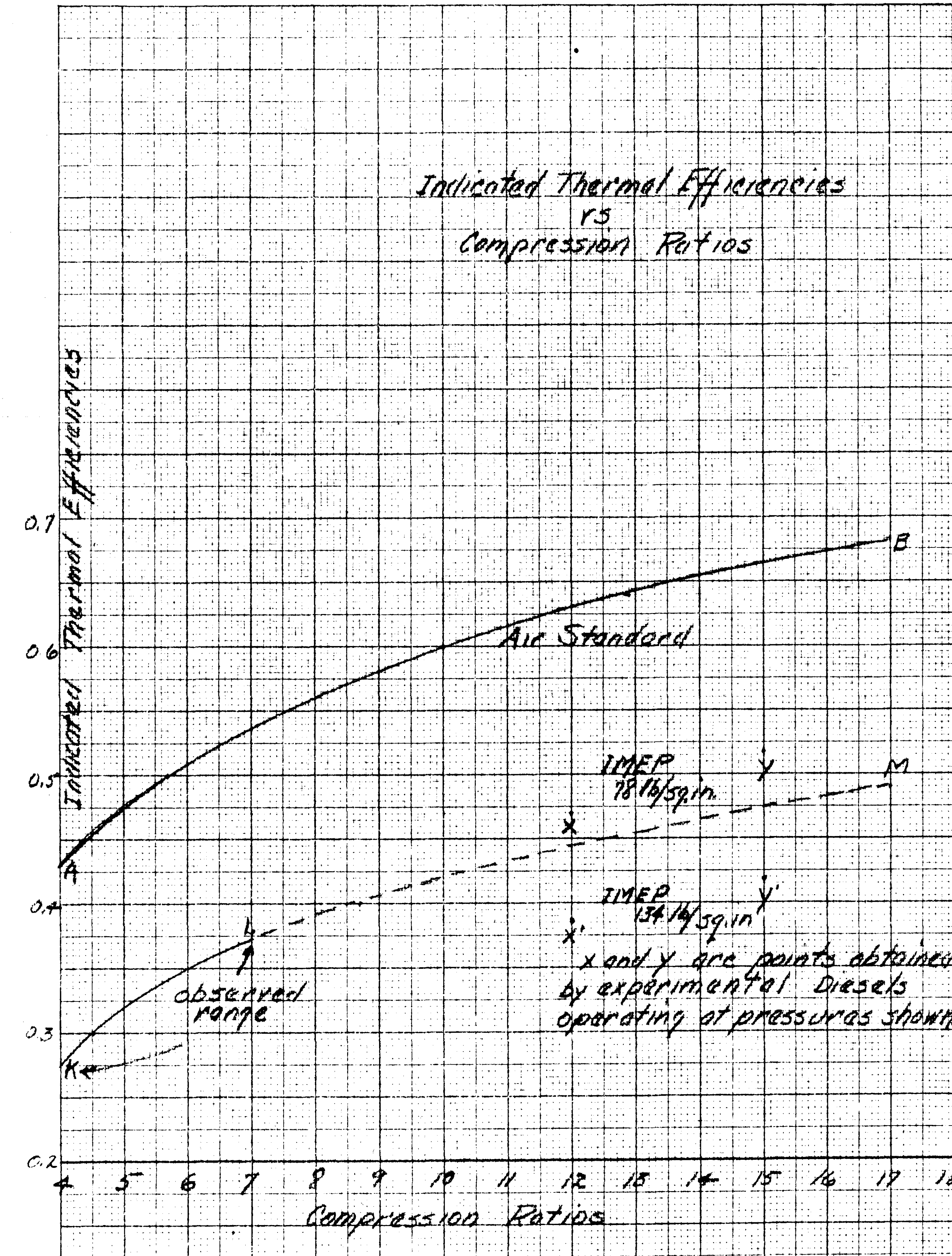
Indicated Thermal Efficiencies
vs
Compression Ratios
Indicated Thermal Efficiencies
Compression Ratios
0.7
0.6
0.5
0.4
0.3
0.2
4
5
6
7
8
9
10
11
12
13
14
15
16
17
18
Air Standard
A
B
IMEP
78 lb/sq.in.
IMEP
134 lb/sq.in
X
Y
M
X'
Y'
L
K
observed
range
x and y are points obtained
by experimental Diesels
operating at pressures shown

Fig. 2

Schematic Cross-section of Proposed Condensor.

The exhaust gas flow is normal to this section and from the top to the bottom, thus the velocity of the cooling air is decreased as the temperature differential increases. It will be noted that the structure presents no particular difficulties.

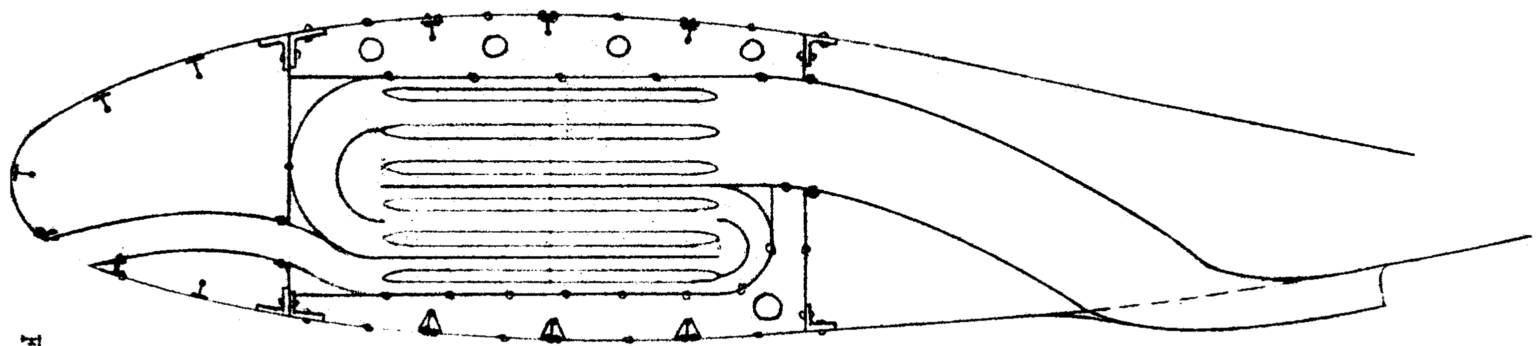

Fig. 3

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