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FATIGUE OF INTERNAL COMBUSTION ENGINES.

By P. Dumanois.

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FATIGUE OF INTERNAL COMBUSTION ENGINES.*

By P. Dumanois.

Internal combustion engines comprise two categories, those which follow the cycle of combustion at constant volume, commonly called the explosion cycle, or the cycle of combustion at constant pressure, derived from the Diesel cycle.

From the kinetic viewpoint, both classes work on the same principle as the steam engine, namely, the transformation of the alternating motion of a piston into a circular motion by means of a crank and connecting-rod.

The analogy with the steam engine ceases here, however. In a steam engine, the temperature of the steam rarely exceeds 200° C. (392° F.), which is too low to affect the structure of the metals in contact with it. When the parts have been calculated in conformity with the customary rules regarding the strength of the materials, with suitable safety factors, the problem has been solved and we have an engine whose functioning is assured without other time limit than that resulting from the normal wear of the moving parts.

With a well-designed lubrication system (which presents no difficulty at the functioning temperatures attained) and

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suitable coefficients of wear, it may be considered that the entire fatigue of a steam engine depends solely on the mechanical phenomena of its functioning.

Such, however, is not the case with interval combustion engines. Here the combustion temperature attains 1600° C. (2912° F.). The cylinder-head and wall, piston-head, valves and spark plugs of explosion engines are periodically in contact with very hot gases. In order to render the mechanical functioning possible, i.e., for the piston to be able to slide in the cylinder without jamming, it is obviously necessary to keep the temperature of the inner wall below that which would cause decomposition of the lubricants. Whence the necessity of vigorous exterior cooling. It follows that the inner and outer walls of the cooled parts have very different temperatures. Thus internal stresses are produced, which are added to those resulting from the mechanical fatigue produced by the combustion pressure, the latter being more than double that produced in the cylinders of steam engines. From another viewpoint and in spite of energetic cooling, the temperature of the walls in contact with the hot gases oscillates at each piston stroke about the mean temperature produced by the cooling. Certain parts, such as spark plugs and exhaust valves, which are difficult to cool or remain long in contact with the hot gases, may even reach critical temperatures for the mechanical properties of the materials composing them, thus causing rupture or defor-

mation and stopping the engine. Moreover, the periodic action of the hot gases on the walls produces, with time, a molecular transformation which modifies their mechanical characteristics and gradually weakens them.

From the above considerations, it follows that, contrary to what takes place in steam engines, the general fatigue of an internal combustion engine is not determined simply by the knowledge of the mechanical data of its functioning. The fatigue depends both on the temperature reached and on the period for which it is maintained.

We can therefore appreciate the importance of this fact for high-powered engines, in which it is sought to obtain, with the minimum material, the perfect combustion of the maximum of fuel in the minimum of air, i.e., for light engines with high temperatures. This is the case of submarine Diesel engines and airplane explosion engines, as a result of the conditions to which they are restricted.

Either a submarine or airplane engine can hardly be gotten into the best working order (which sometimes requires several years) before it becomes obsolete. The designing of new engines handicaps the future and entails large expenses. We can, therefore, understand how important, in such study, is the possibility of referring to a coefficient of fatigue which shall take account of the above considerations.

Conditions which must be satisfied by any criterion of general fatigue.

The usual coefficients of mechanical similitude employed in steam engines not only do not take account of thermal phenomena, but even give results contrary to those sought.

On the other hand, the practical utilization of a general coefficient of fatigue necessitates the preliminary determination of numerical values for satisfactory existing engines, for the sake of comparison.

It is evident that the absolute value of such a coefficient varies with the type of engine considered and the conditions of its employment. An engine which would function only 100 hours without a hitch would be of extraordinary endurance for an aviation engine, but of little use in an industrial establishment. It is also evident that, for a given list of engines, the choice of the engine type, which determines the numerical value of the coefficient, rests on a question of valuation, since it is a question of representing a quality, estimated chiefly in a subjective manner, by a concrete number. It must therefore be based on the mean of a large number of observations. Lastly, as regards the algebraic expression of the coefficient, it is evidently a function of the temperature of the inner wall. We have seen that it is the increase in the temperature of certain parts in contact with the hot gases, which causes the deterioration of those parts and the consequent injury to the engine.

To put it in other words, let us imagine a high-powered engine running continuously. At the end of a certain period, the above-mentioned causes of fatigue will produce phenomena, such as cracks in the cylinder head, warping of the valves, ruptures of spark plugs, etc. Let us now suppose that we cause the engine to produce more power. The mean temperature of the inner walls will increase and the above phenomena will be produced more rapidly. The general fatigue of the engine has therefore increased with the mean temperature of the walls. Since, on the other hand, heavy high-powered engines are made for functioning with as high mean temperatures as possible and just strong enough to obtain the desired endurance, a slight increase in temperature will cause an excessive general fatigue. A direct relation between the general fatigue and the temperature may therefore be assumed. On the other hand, the coefficient must be adapted to geometrically similar engines, whose mechanical dimensions must therefore be independent of the unit of length.

Effect varying the average pressure.

We will now consider the effect on the fatigue of the engine exerted by each of the different quantities entering into the value of the power. The effective power of the engine, which we will designate by π_e , is defined by the formula $\pi_e = A n D^2 C N p$, in which A is a function of the organic

efficiency, D represents the bore, C the stroke, N the R.P.M., n the number of cylinders and P the mean ordinate of the diagram.

Let us first examine the effect of varying p . Since it is proportional to the mean couple, it may be made to represent the mean fatigue resulting from the mechanical functioning, but it is necessary to determine the effect of its variation on the thermal fatigue. The problem differs for the Diesel engine and the explosion engine. In the former, the pressure, at the end of the compression, is already so high that it should not be increased above the point necessary for the automatic ignition of the fuel. On the other hand, we cannot increase indefinitely the quantity of fuel consumed per cylinder charge, since the method of fuel injection employed (by introducing it into the air without previous mixing) renders it impossible to utilize more than half of the oxygen in the charge.

Hence all Diesel engines, using the same fuel, will have practically the same pressure at the end of the compression. This is the combustion pressure which remains practically constant during one period of the cycle. Hence the value of the combustion pressure and of the mean ordinate are practically the same for these engines at their maximum power. It is quite different, however, for explosion engines. For the latter, it suffices theoretically that the pressure p_1 , at the end of the compression, be less than the pressure which would cause auto-

matic ignition. Since the volumetric compression η in existing engines is far below this point, there is evidently considerable margin for variations of the maximum pressure and consequently of the mean ordinate, thus necessitating the study of the possible resulting thermal conditions. We find that the thermal efficiency $\rho = 1 - \frac{1}{\eta^{k-1}}$, increases with the compression, but that, for a like increase in the coefficient of volumetric compression, the increase in thermal efficiency is inversely proportional to the given increase in volumetric compression. Thus, in passing from the compression 4 to the compression 5, the efficiency increase is 0.049, while it is only 0.024 (i.e. half as much), in passing from compression 7 to compression 8. From another viewpoint, when the volumetric coefficient increases, the pressure of combustion and the pressure during the whole expansion increase alike. The same is true of the stresses on the bearings and consequently of the effects of friction. Consequently, the organic efficiency has a tendency to diminish. Under present conditions, it may be considered that, beyond a coefficient of volumetric compression equal to 10, the gain in total efficiency would be quite small. It remains no less true that there is a considerable scale of variations between the values 5 and 10, the former value corresponding practically to that of existing airplane engines. Whatever the given compression pressure may be, if we designate by T_1 the absolute temperature at the end of the compression on starting from the same

exterior temperature, the temperature increase t_1 , resulting from the combustion, is practically constant (within specific heat variations which may be considered negligible), since it corresponds to the combustion of the same quantity of the fuel mixture. The following table gives the principal combustion characteristics, when the coefficient of volumetric compression varies from 4 to 8, on assuming the absolute exterior temperature to be 283°C .

η	P_1	T_1	$T_2 = T_1 + t_1$	P_2	T_3	ρ
4	6.957	493	2393	33.83	1375	0.426
5	9.52	538.8	2438.8	43.09	1281	0.475
6	12.3	579.5	2479.5	52.57	1211	0.512
7	15.26	616.4	2516.4	62.25	1156	0.541
8	18.38	650.2	2550.2	72.1	1110	0.565

The above table shows that the absolute temperature T_2 at the end of the combustion, which is equal to the absolute temperature T_1 at the end of the compression increased by 1900°C , increases less rapidly than the temperature T_3 (at the end of the expansion) decreases. Thus, in passing from compression 5 to compression 8, everything else being equal, T_2 increases 111.4°C while T_3 decreases 171°C . The mean temperature of the cycle tends to decrease. The decrease of T_3 causes a decrease in the temperature of the exhaust valves, so that it may be considered that the thermal fatigue, save perhaps for the

spark plug,* does not vary appreciably with the compression.

It is quite different with the mechanical fatigue. In fact, for the same change in compression (from 5 to 8), the maximum pressure P_2 passes from 43 to 73 kg, an increase of 67%, which greatly affects the conditions of the mechanical work of the engine.

Because of the importance of the increase in the compression pressure for aviation engines, we are going to study this case more closely.

Let P_0 represent the intake pressure, i.e., in the cylinder at the end of the intake.

Let P_1 and T_1 respectively represent the absolute pressure and temperature at the end of the compression.

Let P_2 and T_2 represent the same at the end of the combustion.

Let T_0 represent the exterior temperature.

Let t represent the rise in temperature produced by the combustion.

Let K represent the ratio $\frac{C}{c}$ of the specific heat C at constant pressure of the gases in the cylinder to the specific heat c at constant volume.

Let η represent the ratio of volumetric compression $\frac{v+V}{V}$, v being the volume of the dead space or clearance and large V

* In reality, the trouble with the spark-plugs seems to be chiefly the mechanical difficulty of withstanding the high pressure, rather than any thermal difficulty.

the stroke-volume.

In combustion at constant volume, we have

$$\frac{P_1}{P_0} = \eta^k \quad (1)$$

$$\frac{P_1 v}{T_1} = P_0 \frac{(V + v)}{T_0} \quad (2)$$

$$T_2 = T_1 + t \quad (3)$$

$$\frac{P_2}{P_1} = \frac{T_2}{T_1} \quad (4)$$

whence we deduce:

$$\begin{aligned} P_2 &= P_1 \left(1 + \frac{t}{T_1}\right) = P_0 \eta^k \left(1 + \frac{t}{T_0 \eta^{k-1}}\right) \\ &= \left(\eta^k + \frac{t}{T_0} \eta\right) P_0 = P_0 d \end{aligned}$$

on designating by d the quantity

$$\eta^k \left(1 + \frac{t}{T_0 \eta^{k-1}}\right) = \eta^k + \frac{t}{T_0} \eta$$

On the other hand, if Q denotes the quantity of heat liberated by the combustion and m the weight of the combustible mixture, we have $t = \frac{Q}{m}$ and, for a given cycle and a given fuel, $\frac{Q}{m}$ is a constant. The same is true of t .

In the case under consideration, we can take 6.71 as the mean numerical value of the ratio $\frac{t}{T_0}$.

We will designate by the same symbols, with the index ' , the same quantities relating to the same engine, but with a different degree of compression, on supposing η' to be greater than η .

The compression, intake temperature and revolution number remaining constant, the indicated power π_1 and, consequently, the couple Γ are practically proportional to the thermal output. Thus we have

$$\Gamma = b P_0 \rho, \quad (6)$$

b being a constant.

For the same intake pressure P_0 , we will have

$$\frac{P'_a}{P_2} = \frac{d'}{d} \quad (7) \quad \frac{\Gamma'}{\Gamma} = \frac{\rho'}{\rho} \quad (8)$$

If we wish a high-compression engine to work under the same conditions of maximum mechanical fatigue as an ordinary engine, i.e., at the same explosion pressure, it will be necessary, by reason of formula (5), to reduce the intake pressure to a value P'_0 , so that

$$P'_0 = P_0 \frac{d}{d'} \quad (9)$$

Under these conditions, the corresponding couple Γ' of a high-compression engine and that of an ordinary engine are connected by the ratio

$$\frac{\Gamma'}{\Gamma} = \frac{\rho'}{\rho} \frac{d}{d'} \quad (10)$$

If we assume that we wish to obtain the same value of the couple for both engines, it will then be necessary to give a value P_0'' to the intake of the high-compression engine, so that

$$\Gamma = b P_0'' \rho' = b P_0 \rho$$

whence

$$P_0'' = P_0 \frac{\rho}{\rho'} \quad (11)$$

and the corresponding combustion pressure will be

$$P_2'' = P_0'' d' = P_0 d' \frac{\rho}{\rho'} = P_2 \frac{d' \rho}{d \rho'} \quad (12)$$

In the particular case of functioning on an airplane, if it is desired to obtain, with a high-compression engine, an engine which has, on the ground, the same couple or the same explosion pressure as an ordinary engine, it suffices to throttle, on the ground, the intake of the carburetor, so as to obtain, according to the case sought, the intake pressure P'_0 or P''_0 . In climbing, it is possible, by gradually opening the throttle, to keep either the couple or the explosion pressure constant up to an altitude defined by the condition that μ_z^* be equal to $\frac{\rho}{\rho'}$, in the case of a constant couple, and to $\frac{d}{d'}$ in the case of a constant explosion pressure.

It is interesting to note, when the engine is provided with a device enabling it to maintain a constant couple up to the altitude of wide-open throttle, how the resistance of the propeller diminishes with the density and how the revolution speed and, consequently, the engine power increase with the altitude up to the moment when the throttle is wide open.

We can likewise envisage the take-off, not with a couple, but with a power equal to that of an ordinary engine. In this case, the opening of the throttle near the ground is greater than when functioning with a constant couple and the power re-

* μ_z is the ratio of the density of the air, at the altitude z , to its density near the ground corresponding to P_0 .

mains constant until the carburetor is wide open, i.e., at the same altitude as in the case of functioning with a constant couple.

Hence the power at this altitude can be obtained either by maintaining, after leaving the ground, a power equal to that of an ordinary engine, or by maintaining a couple equal to the couple of an ordinary engine.

It is, moreover, important to note that, in both cases, the explosion pressure near the ground is considerably higher than with an ordinary engine and that it is greater in the case of constant power.

The following table sums up the comparative results we may expect according to these different hypotheses, by taking, as the basis of the comparison, an ordinary engine giving, on the ground, 300 HP at 1800 R.P.M. with $\eta = 5$, when the pressure is carried to a value defined by $\eta = 7$.

	Intake pres- sure	Combus- tion pressure atm.	R.P.M. on ground	HP	Altitude of full intake m
Ordinary engine	1	42.85	1800	300	0
High-compression engine.					
Equality of R.P.M. on ground.	1	62.25	1800	342	0
Equality of power	0.9166	57.06	1723	300	1380
Equality of couple	0.8776	54.63	1686	263.3	1380
Equality of explo- sion pressure	0.6884	42.85	1493	195.3	3700

Thus far we have assumed that it is possible to increase the coefficient of volumetric compression, by reinforcing the crankcase and other parts of the engine, so as to enable them to withstand, with sufficient margin of safety, the corresponding explosion pressure.

In practice, it is not possible to increase arbitrarily the pressure at the end of the compression for a reason, which is, moreover, essentially a function of the fuel used. Experience has, in fact, shown that, for a given fuel, if the compression pressure is increased, there comes a moment when the combustion takes place in a form which causes a shock phenomenon producing the impression of an explosion. In reality, this phenomenon, which the English call "pinking," is the result of undulatory pressures in the cylinder, of which the following is the commonly accepted explanation.

When the charge is ignited at any point, by means of an electric spark, for example, there is produced a local pressure increase in the vicinity of this point, which compresses the rest of the charge at the same time that the combustion is gradually spreading. If this compression is propagated with sufficient rapidity to be adiabatic and if, at the same time, the combustion spreads slowly enough, there comes a moment, when the unburned portion of the charge acquires the temperature of spontaneous ignition and burns instantly, with a local increase in pressure and temperature proportional to the adiabatic

compression necessary to cause this spontaneous ignition. This suddenly increased local pressure is greater than that of the burned mixture, since it occurs at a higher initial compression, and therefore causes, in its turn, the compression of the portion of the charge already burned. This gives rise to a particularly destructive dynamic phenomenon causing a metallic shock similar to that which would be produced by a blow on the combustion chamber.

Experiments, now become classic, performed by Ricardo on his variable-compression engine with different fuels, showed clearly that the phenomenon of the production of the explosive wave is essentially a function of the compression pressure. This result may be readily explained by the following consideration.

When, in an engine with a given compression ratio, a fuel is used, which, with the throttle wide open, produces the phenomenon of undulatory combustion, it is found that the combustion can be made normal by partially closing the throttle. This does not affect the temperature at the end of the compression, if the intake temperature remains constant, since the former depends simply on the ratio of volumetric compression. On the contrary, the pressure at the end of the compression is reduced, since it is related to the intake pressure by Poisson's law and since the intake pressure is reduced by the throttling.

This phenomenon can be analyzed still further. Let us imagine an explosion chamber in which a spark is produced and let us consider what takes place with different combustible mixtures

having increasing combustion speeds. In mixtures with a low combustion speed, the compression produced by the initial combustion increases so slowly that it ceases to be adiabatic and consequently the temperature of spontaneous ignition is not reached. This is the case with very heavy fuels, whose spontaneous ignition temperature is very high.

When the combustion speed increases, the increase in the initial pressure is greater and the compression of the rest of the charge is more rapid, so that there comes a moment when the combustion speed is too low to ignite all the charge before the adiabatic compression is produced, causing the phenomenon of undulatory combustion. If the combustion speed increases still further, it reaches a point where it is greater than the propagation speed of the adiabatic compression and the phenomenon of undulatory combustion disappears. In the case limited by infinite combustion speed, we enter the theoretical cycle of combustion at constant volume, without undulatory phenomena.

From another viewpoint, account is taken of the effect of the absolute values of the compression pressure and combustion pressure on the nature of the phenomenon. The higher the compression pressure, the smaller the space occupied by a given mass of gas and, consequently, the smaller the portion of the walls of the combustion chamber in contact with it. The possibility of losing heat is consequently diminished and the probability of realizing adiabatic compression of the unburned portion of the

charge is increased. Generally everything which opposes the temperature elevation resulting from the adiabatic compression, increases the rate of compression. This is especially the case, when the fuel has a high latent heat of vaporization, like alcohol, for example, the vaporization of which prevents the temperature from going high enough to produce spontaneous ignition. For a similar reason, the injection of water or an excess of fuel gives the same result. Also for a similar reason, the critical rate of compression, for a given fuel, increases when the bore is diminished. In fact, the mass of the fuel mixture varies as the cube of the linear dimensions and the surface area of the combustion chamber varies as their square. Hence, if we pass from a certain bore to one half as large, the ratio of the radiating surfaces to that of the number of calories liberated is doubled, the compression is less adiabatic and a higher pressure is required for spontaneous ignition. Thus the maximum rate of compression for gasoline, which is 5 for a 140 mm bore, may reach 7 for a 70 mm bore. Another probable phenomenon is the resistance to penetration offered by the unburned portion of the charge to the products of the initial combustion.

The phenomenon may be explained, moreover, by the following analogy. If we consider a Diesel engine with a two-stroke cycle, we find that, as the scavenge pressure is increased, the expulsion of the burned gases improves, passes through a maximum at a certain pressure, beyond which it diminishes and remains at practically the same value regardless of the scavenge pressure

employed. This phenomenon is explained as follows:

When the scavenge pressure is sufficiently small, the scavenge air drives before it the burned gases without penetrating them, by playing a role analogous to that of a piston. If, on the contrary, the pressure is high enough, the jet of scavenge air penetrates the burned gases after the manner of a projectile, in order to take the shortest path from the scavenge ports to the exhaust ports, thus replacing by fresh air only that portion of the exhaust gases thus traversed. It is conceivable that an analogous phenomenon may take place in the case we are considering.

If the instantaneous pressure resulting from the local ignition is sufficiently high, i.e., if the combustion speed is great enough, the burned gas, instead of compressing the rest of the charge, penetrates and ignites it, thus rendering the phenomenon of "pinking" no longer possible. If, on the contrary, the local pressure increase is too small, this penetration will cease and give place to adiabatic compression, susceptible of producing undulatory combustion.

There are other causes due to the inherent peculiarities of the engine and which cause, for a given fuel, the critical coefficient of compression to vary for different engines.

We have indicated that the production of undulatory combustion depended on the pressure at the end of the compression. This is determined, however, not only by the coefficient of volumetric compression resulting from the geometrical dimensions

of the engine, but also by the effective filling of the engine cylinders. The latter depends on the lift of the intake valves, the shape of the pipes and the revolution speed of the engine. Thus, a fuel, which, with the throttle wide open, normally burns rapidly, may produce undulatory combustion, if the revolution speed decreases, as a result of the increase in the resisting couple, because the filling improves. This sometimes occurs in automobile engines while climbing a hill.

It is likewise clear that the location of the ignition point, with reference to the combustion chamber, and the shape of the combustion chamber are of considerable importance and that a central ignition in a hemispherical chamber, by diminishing the maximum distance to be traversed by the combustion, is preferable to a lateral ignition. For the same reason, simultaneous ignition at two opposite points is better. It would be advantageous to increase the number of ignition points even further, since, at the limit, the simultaneous ignition would produce the theoretical cycle without undulatory combustion. It might be possible to approach this ideal by igniting the charge not by a spark but by a jet of burning gas at high speed as we have had occasion to propose.

Lastly, it is not improbable that the molecular constitution of the fuel may have a certain effect and that phenomena analogous to cracking may take place as we suggested in 1916, in connection with Diesel engines, in an article which was awarded

a prize by the Academy of Sciences. This effect must be slight, however, in a combustion whose total duration is only about $1/300$ of a second.

On analyzing the above considerations, we arrive at the following conclusions, confirmed by experience, for petroleum fuels. For very light and homogeneous fuels, the admissible coefficient of volumetric compression, in order to obtain normal combustion with the present bores of aviation engines, is about 5. It is lowered to 4.5 by the employment of heavy fuels and below 4 if mixed with kerosene. If very heavy fuels are employed, the phenomenon ceases. In a phenomenon of the same order, we likewise find the explanation of the back-fires or rather of the local explosions which occur in some engines and which come from the lack of homogeneity of the fuel.

In this case, the non-homogeneous portion takes fire by spontaneous ignition resulting from the adiabatic compression produced by the more rapid combustion of the homogeneous portion. The final temperature of this localized spontaneous combustion is higher than the combustion temperature of the rest of the charge. If, for example, the normal volumetric compression is 5 and the compression of spontaneous ignition is 12, the temperature difference at the end of the combustion is 26°C (502°F) with a local explosion pressure of over 150 kg, instead of 42, capable of producing destructive local effects. Thus undulatory combustion produces more serious consequences with less volatile

fuels or, more exactly, when the temperature of spontaneous ignition is higher, but ceases altogether with very heavy fuels.

Since the temperature increase produced by the combustion is practically constant, the final temperature of combustion is, in fact, as much higher as the initial temperature is higher, i.e. as the fuel is less inflammable and the compression required for spontaneous ignition is correspondingly greater.

In order to avoid such disadvantages, a homogeneous fuel must be used in high-compression aviation engines. Thus, if we remove by distillation both the lightest and heaviest constituents from a given fuel, the product has about the same density as the mixture of these lightest and heaviest portions, but is not so volatile and hence not so liable to produce the above-mentioned phenomena. If the coefficient of compression for spontaneous ignition is so high that the corresponding pressure cannot be attained by the initial combustion of the charge, the phenomena will cease to be produced.

In contrast with gasoline, certain other fuels admit of very high compression. This is the case with benzol and alcohol, which stood, in tests recently made by the Washington Bureau of Standards on a Liberty cylinder, a compression of 14 without disadvantage. It is reasonable, however, on the basis of the above considerations, to suppose that when these fuels reach the critical pressure, they will produce particularly destructive phenomena.

The maximum limit of the compression pressure is not generally imposed by spontaneous ignition, but rather by undulatory combustion. These two limits may be quite different, according to the fuel. Thus, for some kerosenes, the second is less than 4, while the first is above 10. While the first depends solely on the ratio of volumetric compression and a well-determined physical characteristic of the fuel, its temperature of spontaneous ignition, the second depends simultaneously on the absolute value of the compression pressure, the shape of the combustion chamber, the ignition, the heat of vaporization of the fuel and especially on the combustion speed.

The increase in the compression pressure is very important. To say, in fact, that the thermal efficiency increases with the compression, is equivalent to saying that, for the same number of calories, more heat is converted into work, i.e., that the couple is greater, or that, with the same couple, the fuel consumption is less. The question is one of thermal efficiency, as expressed by the mechanical efficiency of the engine itself. The mechanical efficiency is diminished, however, by friction. When the compression is increased, the pressure on the bearings and, consequently, the friction increases, thereby diminishing the mechanical efficiency. There remains, however, at the compressions now attainable, an appreciable gain in the net efficiency of the engine.

The comparative tests made with alcohol on a 180 HP Hispano-

Suiza engine gave, in passing from the compression 4.7 to 7.4, an increase of 11% in the couple and a decrease of about 20% in fuel consumption. This increase in the value of the couple confirms the results previously obtained with gasoline on a Clerget rotary engine with variable compression. Mr. Clerget was, in fact, the first to solve the problem of varying the compression during flight by varying the distance of the common crank-pin from the axis of rotation. Thus both the length of the stroke and the compression were increased at the same time.

During the flight tests made with this engine mounted on a Liore-Olivier airplane, October 30, 1918, in passing, at an altitude of 3000 m (about 10000ft.), from the compression 5.2 to 7.33, thereby increasing the stroke from 170 mm (6.7 in.) to 190 mm (7.48 in.), the R.P.M. increased from 1210 to 1300. The ratio of the powers is practically $(1300/1210)^3$ or 1.24. After deducting from this quantity the part due to the variation in the stroke, or $190/170 = 1.118$ there remains 1.122 due to the variation in the compression. This increase, by itself, would give an R.P.M. of

$$1210 \sqrt[3]{1.12} = 1259$$

corresponding to a ratio of the couples of

$$1.12 (1210/1259) = 1.07$$

or an increase of 7% in passing from the compression 5.2 to the compression 7.33, a result which agrees perfectly with the one previously indicated.

Thus, as we have already seen, the increase in compression, which increases the explosive pressure, has no appreciable effect on the mean temperature of the cycle, but greatly increases the mechanical stresses undergone by certain parts. It is therefore necessary to strengthen certain parts of the engine, thus entailing an increase in weight. Let us investigate the importance of this increase, by comparing two like engines in which the ratio of the explosive compressions is λ . In passing from an ordinary to a high-compression engine, we find that the weight of some parts does not vary with the explosive pressure. Such are the pumps, valves, carburetors, camshaft and generally the cylinders and pistons, since their thickness, for existing bores, is determined not by the mechanical conditions of resistance to the explosive pressure, but by local strength considerations, which conduce to a greater thickness than that resulting from the preceding consideration. Other parts vary in proportion to the coefficient λ . They are the ones subjected to traction and compression stresses. This is the case of the crankcase attachments. Still others vary in proportion to the cube root of λ . These are the ones which are subjected to the stresses of torsion or flexure, such as the crankshaft or a portion of the crankcase. Lastly, it may be considered that the percentage weight of the high-compression engine is given by the formula

$$65 + 10\lambda + 25\sqrt[3]{\lambda}$$

which, in passing from the compression 5 to the compression 7, corresponds to an increase of about 8.4% in weight. The gain in the couple or power is at least as much for the same R.P.M. and the weight per HP will therefore tend to decrease. The fuel consumption is also diminished, thereby saving both weight and expense. The latter point is important, since it is in the diminution of the fuel consumption, rather than in the extreme lightening of the engine, that any considerable saving in weight can be made without compromising safety. A saving of 30 grams per HP amounts to 36 kg for a 300 HP engine or 12% of the weight of the engine, for a radius of only four hours.

Unfortunately gasoline, which is the most powerful fuel (with a heat content of nearly 11000 calories per kilogram), hardly admits of a compression of over 5 at sea level. Benzol, which has a calorific content of 9000, can stand greater compression, but its consumption is a little larger and it begins to yield crystals at -4°C (25°F). As for alcohol, which stands high compressions and remains liquid at the lowest temperatures, it contains only 6900 calories per kilogram and accordingly necessitates the consumption of a considerably greater weight of fuel per HP. Even with a compression of 7, it is necessary to count on a consumption of 350g per HP, or over 100g more than the weight of gasoline consumed in an ordinary engine.

Special dispositions must accordingly be made in order to enable the use of high-compression engines. Let us first note that, whatever be the degree of compression, there is an altitude

at which, because of the decrease in density, the pressure at the end of the compression is no greater than in an ordinary engine at sea level. Hence gasoline may be used after attaining this altitude. This solution, recommended by some, is very simple.

A second solution consists in a special assemblage of the engine parts, enabling the variation of the compression during flight. This method has been employed on the Clerget engine, of which we have already spoken.

A third solution, which has likewise been tried, consists in the employment of an under-fed, high-compression engine. The throttle is only partially opened on the ground, so as not to exceed the critical pressure at the end of the compression. Unfortunately, this method has the disadvantage of limiting the possible increase in compression. On referring to the results already given, we find that a 300 HP engine, using gasoline, with a compression of 7, is thus limited, on the ground, to less than 200 HP, a decrease of over 30%, which may make it very difficult to take off.

In our opinion, however, these three methods all have one disadvantage, namely, the necessity for the pilot to execute a maneuver at a certain altitude. Even if this is admissible on commercial airplanes, where the pilot has nothing to do but follow his course, such is not the case on military airplanes, where the pilot must fight at a constantly varying altitude.

His task then requires his entire attention and it is inopportune to require of him a supplementary maneuver, the neglect of which may have particularly serious consequences. After a descent at low engine speed, to pick up near the ground would entail the destruction of the engine in a very brief space of time.

We can, indeed, imagine this maneuver to be executed by an automatic device actuated by the changes in pressure caused by changes in altitude. But if such a device can give satisfactory results for maneuvers requiring a very slight effort, by acting, for example, on the carburetor or throttle, as in the third method, the case is quite different when a greater effort is required, which is the case for a maneuver of variable compression or of adjusting the fuel cocks. In any case, moreover, there would be danger of serious damage to the engine, if the automatic device should fail to function.

We think there is a simpler way to seek the immediate amelioration of the available power of an aviation engine, namely, by increasing the compression, not by changing the engine (except to reinforce it locally), but by modifying the fuel, which is really at the bottom of the difficulty. Ricardo's experiments have shown, moreover, that it is possible to prevent undulatory combustion by adding certain substances to the gasoline. To be sure, this method has not yet been practically applied in aviation and the substances suggested have a low calorific content or form rather unstable mixtures with gasoline.

This is particularly true of ethyl alcohol which is an anti-detonant of the first order. It should be noted, however, that in replacing gasoline by a mixture of gasoline and alcohol, the consumption per HP does not vary in proportion to the ratio of the calorific contents. The power exerted on the crankshaft depends, in fact, not only on the thermal and mechanical efficiency, but also on the efficiency of the cycle actually obtained, as compared with the theoretical cycle.

This efficiency is increased by the improvement in the combustion speed due to the presence of the alcohol and it is thought that, for certain proportions of alcohol, the improvement thus effected would more than offset the loss in calorific content. We have recently demonstrated the correctness of this supposition on an engine which gave, with gasoline, 532 HP at 1530 R.P.M. A simple manipulation of the fuel cocks replaced the gasoline, having a calorific content of 10800 calories, with a mixture of 90% alcohol and 10% absolute alcohol having only 10300 calories, thus increasing the R.P.M. for the same consumption, to 1535 and the HP to 540. In order that false conclusions may not be drawn from this experiment, it must be added that the employment of this mixture is not practicable in aviation, because a very small proportion of water or a temperature only a few degrees below freezing (0°C or 32°F) suffices to separate the constituents. The alcohol would then sink to the bottom of the tank and alone enter the carburetor adjusted for the mixture and thus cause back-firing. This is the reason which led the "Comite'

Scientifique du Carburant National" to recommend to the French Government, for the application of the law of February 28, 1923, mixtures containing 40 to 50% of absolute alcohol, whose stability is satisfactory. The presence of such a large proportion of alcohol has the further advantage, by reason of its anti-detonating properties, of enabling the use of heavy and non-homogeneous oils having just the right volatility for starting a cold automobile engine without risk of undulatory combustion.* With the above proportions, while improving the efficiency of the cycle, the consumption on the bench is, in weight only, 8-10% greater than that of pure gasoline, although a simple comparison of their calorific contents would indicate an increase of 16-18% in the fuel consumption. An increase of even 8% is not admissible, however, for aviation engines. On the contrary, this increase in fuel consumption, as determined on the bench, is not so great in the ordinary use of an automobile. Since the engine is more constant with the alcohol mixture and does not knock, it is possible to avoid speed adjustments which increase the fuel consumption and which are necessary with gasoline.

We accordingly believe the solution of the problem should be sought in the study of anti-detonants. We have thus been able to use a 300 HP Hispano-Suiza engine compressed to 6.5, with a mixed fuel containing about 10000 calories per kilogram, which

* We have pushed the problem even further and are now using an automobile which consumes a mixture of alcohol and kerosene (the worst detonating fuel) and which detonates less than with pure gasoline.

gave no trouble above -20°C (-4°F). It should be noted, moreover, that with such a method, the only danger is that of obtaining too high powers at ground level. It may, therefore, be desirable, simply for diminishing the fatigue of both material and personnel, to employ some device for regulating the throttle up to a given altitude. It should be remembered that, from the physiological viewpoint, a swift ascent which removes the pressure from the body, is more dangerous than the descent and also that it is of prime importance to improve the performances at the altitude of utilization.

In an explosion engine, the variation of the compression pressure, provided it does not produce undulatory combustion, only very slightly affects the thermal fatigue of the engine. On the other hand, it causes variations in certain local stresses and consequently in the work of the corresponding parts.

Furthermore, both in explosion engines and in Diesel engines, the general fatigue of the engine and its endurance depend not only on the maximum pressure, but also on the length of time this maximum pressure is exerted or, in other words, on the couple furnished by the engine.

We may, therefore, as in the case of the steam engine, consider the value of P as expressing the mechanical fatigue of

similar engines running at the same speed with different couples.

Variation of the thermal fatigue in terms
of the revolution speed.

The amount of heat absorbed by the wall during a cycle, varies as the length of the cycle itself, i.e. inversely as N . This loss of heat diminishes the temperature of the combustion products or, in other words, creates a certain difference $\Delta\theta$, varying as N , between the actual mean temperature of the gases and their theoretical temperature.

If the revolution speed of the engine increases, $\Delta\theta$ decreases or in other words, the mean temperature of the gases increases by a $\Delta\theta'$ equal to the variation of $\Delta\theta$ with the opposite sign, on approaching the temperature obtained in the theoretical cycle. This increase in the mean effective temperature of the gases causes a correlative increase in the temperature of the inner walls of the combustion chamber and in the corresponding tensions.

For a given engine of fixed mode of construction and running with wide-open throttle, we consider that the thermal fatigue, previously defined, is diminished in proportion to the reduction of the temperature of the inner walls of the combustion chamber or, in other words, in proportion as the difference increases between the mean temperature of the inner wall (or, what amounts to the same thing, between the mean temperature of the gases) and

the theoretical temperature with an athermanous surface.

We may consider that the endurance will be just as much greater as the difference $\Delta\theta$ between the actual mean temperature of the gases and the mean theoretical temperature, with an athermanous surface, is greater. We can accordingly determine a criterion of thermal fatigue by the condition $\Delta\theta$ greater than a given value, or (what amounts to the same) with N smaller than a given value.

Since we have the fundamental formula

$$\pi_i = A n N D^2 C p$$

n being the number of cylinders, the above inequality becomes $\pi_i/A n D^2 C p$ smaller than a given limit or $\pi_e/a n D^2 C p$ smaller than a given limit, on designating by π_e the power exerted on the shaft and on neglecting the variations in the organic efficiency.

The original hypothesis is that the couple and, consequently, the mean ordinate p are constants, regardless of N . The quantity $A p$ is accordingly a constant B . Under these conditions, the above inequality may be written

$$\frac{\pi_e}{n \frac{\pi}{4} D^2 C} < \frac{A B}{\frac{\pi}{4}}$$

The denominator of the first member is the total volume of the cylinders and the quotient thus obtained consequently represents the power per liter of the stroke-volume, which is the standard generally employed for automobiles. It is of value only

for a given engine of constant couple with variations in the revolution speed alone.

Case of geometrically similar engines.

If we compare two geometrically similar engines, whose pistons have the same linear speed, and if μ represents the ratio of the linear dimensions of the two cylinders, the power and the areas will vary as μ^2 and the thickness will vary as μ , which satisfies the law of geometric similitude and the law of equality of mechanical resistance, although, in order to maintain the same conditions of calorific exchange, the thickness would have to remain the same.

In fact, the quantity of heat imparted by the gases to the inner wall of the combustion chamber varies as the area of contact S , i.e. as μ^2 . The same is true of the quantity of heat imparted by the outer wall to the cooling water. The two quantities must be equal and must also equal the quantity of heat which passes through the wall of thickness e . The latter quantity varies as S/e , i.e. as μ .

In order that the equations of calorific exchange or (which amounts to the same thing) the temperatures of the walls remain constant, the thickness e of the wall of the combustion chamber would have to be independent of μ . There is therefore a contradiction between the mechanical similitude and the maintenance of cooling conditions. The condition of mechanical resistance interferes with the flow of heat and increases the thermal fatigue of the walls.

Determination of a criterion of general fatigue.

The mechanical fatigue of an engine may be expressed by the mean value of the ordinate p . Its thermal fatigue follows from the above considerations which it will vary as N and e . As the coefficient of general fatigue we will therefore adopt

$$p N e = \frac{\pi_e e}{A n D^2 C}$$

The thickness e is related to the bore D by the condition of mechanical resistance

$$e = \frac{P D}{a r}$$

in which P denotes the maximum relative pressure of combustion, r the unitary fatigue of the metal and a , a constant, so that on neglecting the constants, we finally have

$$\Phi = \frac{\pi_e}{n D C} \frac{P}{r}$$

It may be objected that the reasoning which led to this coefficient is not strictly mathematical and that it would be possible to imagine other coefficients which would lead to different results. Aside from the fact that mathematical analysis cannot be strictly applied to the study of phenomena so little known as the temperature exchanges under consideration, without non-verified hypotheses, which render the results doubtful, the above coefficient, being the result of logical hypotheses on means in accord with experimental results, has the advantage of being particularly simple.

From the mechanical viewpoint, in order to be valid in comparing engines of different dimensions, it must be independent of the unit of length. Now the dimensions of

$$\Phi = p N e \text{ are } ML^{-1} T^{-2} T^{-1} L = MT^{-3},$$

which are practically independent of L .

Lastly, the application of this coefficient to existing engines, especially high-compression engines, leads to a method of classification in general conformity with that practically obtained from the endurance.

In the special case of engines of the same construction and following the same cycle with the same volumetric compression, P and r are the same. It suffices to take, as the criterion of fatigue, the value μ of the variable part of Φ :

$$\mu = \frac{\pi_e}{n D C}$$

This is, in particular, the criterion applicable to most of the Diesel engines, which all have practically the same volumetric compression and a similar structure.

Lastly, the general criterion of fatigue can, in this case, be represented by the simple condition π_e/nDC , smaller than a fixed value, and we have had occasion to find that it agrees with the reality. Its value does not exceed 4 for four-stroke industrial engines of current type, D and C being expressed in decimeters and π_e in HP. When it exceeds 5, the endurance is often less. Above 5.5, the engine would probably be too weak.

In the general case of comparing similar engines of the same construction and cycle, but of different compression and different mechanical fatigue, it is necessary to use the complete formula

$$\phi = \frac{\pi_e}{n D C} \frac{P}{r}$$

On remarking that $P = (P_2 - P_0)$, taking $P_0 = 1$ and referring to formula (5), we have

$$\phi = \frac{\pi_e}{n C D} \frac{1}{r} [\eta^k + 6.7 \eta - 1]$$

In the cycle of combustion at constant pressure, we have

$$P_2 = P_1 = P_0 \eta^k \text{ and}$$

$$\phi = \frac{\pi_e}{n C D} \frac{1}{r} [\eta^k - 1]$$

Conditions for employing the power per liter of stroke-volume as the criterion of general fatigue.

Up to this point, we have assumed that the thickness e could be replaced by the proportional quantity D . The relation which D bears to e , in terms of the maximum combustion pressure P and of the fatigue r of the metal is $r = P D / a e$.

The admissible fatigue for the metal used may happen to lead, by the application of this equation, to a thickness so small as to be practically impossible and expose the cylinder, at the least shock, to local deformations which, by compromising the functioning, without considering that the decrease in thickness,

due to the oxidation of the walls in contact with the cooling water, would cause very appreciable local increases in the fatigue.

Let us consider, for example, a steel cylinder of 120 mm bore. The pressure P being 40 kg per sq.cm., with an admissible fatigue r of 15 kg per sq.mm., the thickness deduced from the formula would be hardly a millimeter for the cylinder and still less for its head, which is insufficient for giving the cylinder a suitable rigidity and which would make the work of construction extremely difficult and expensive.

Let us then consider different engines of increasing bore, of the same structural type and cycle and with the same maximum pressure or (which amounts to the same thing, as we have seen) having the same volumetric compression. Let r represent the admissible fatigue for the metal of the cylinder and let e_1 be the requisite minimum thickness. Let D represent the bore, as defined by the equation

$$D_1 = \frac{a e_1 r}{P}$$

For all the engines of the type considered, with a bore less than D_1 , we will retain for the thickness the constant value e_1 , the unitary fatigue diminishing with the bore. The coefficient of the general fatigue, for such an engine, is

$$\phi = p N e_1 = \frac{\pi e e_1}{A n D^2 C}$$

We can therefore take, as the coefficient of general fatigue, the quantity

$$\frac{\pi_e}{n D^2 C} \text{ or } \frac{\pi_e}{\frac{\pi}{4} n D^2 C}$$

Now this is precisely the case of most automobile and aviation engines for which the structural requirements lead to cylinder thicknesses well above those required by the condition of mechanical strength.

The above conditions therefore justify the employment, as the criterion of general fatigue, of the coefficient of power per liter of stroke-volume, commonly employed for automobiles, it being understood that it will only give comparable results for similar engines of the same compression, materials and thickness of wall and of the same volumetric compression.

For engines of the type under consideration, with a bore exceeding D_1 , the thickness is determined by the condition of constancy of mechanical fatigue. Since the thickness is proportional to the bore, we return to the case of utilizing the coefficient ϕ already examined.

Conclusions.

The above conditions enable the employment of a criterion of general fatigue which simultaneously takes account of both mechanical and thermal conditions, for the sake of comparing any projected engine with engines of the same type already in use.

Whatever the case examined, the coefficient of general fatigue can be considered only as giving a relative indication. The numerical value determined by experiment is not absolute. It is a valid limit, at the instant considered, for the structural processes actually employed, but is susceptible of modification, as warranted by progress in metallurgy and methods of construction. This is true of most of the coefficients used in applied mechanics and here lies their fundamental importance from the viewpoint of the engineer. They put in concrete form, at any given epoch, the results of the knowledge acquired and the experiments made. In thus enabling a clear appreciation of the difficulty of the new problems presented, they prevent too bold leaps and conduce to the technical prudence indispensable in industrial progress.

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