5. FLOW IN THE CYLINDER
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INTRODUCTION

The gas flow in the cylinder has a profound influence on the performance of the engine, 
whether intended or not. In the early days of the automobile, it was seldom intended, 
because relatively little was understood about turbulence, the primary player in this 
drama. Engines were designed largely by empiricism; from time to time a particularly 
successful design emerged, and its characteristics (to the extent that their relevance was 
recognized) were preserved in subsequent engines.

There are a couple of notable exceptions to this: In the first, between 1903 and 1907 H. 
R. Ricardo [45] (the father of the octane number), working at Cambridge University with 
Professor Bertram Hopkinson, did pioneering work on the effect of turbulence on 
combustion and heat transfer in the IC engine, particularly on the effect of the increased 
effective flame speed on knock, and on the possibility of stratified charge, among other 
things. This led, during the First World War, to great improvements in the design of tank 
engines (giving short flame travel and high turbulence levels, permitting higher 
compression ratios without knock), and after the war led to design modifications of the 
flat-head, or side-valve engine, which resulted in the same performance as the overhead 
valve engine, and which were generally adopted, and resulted in patents.

The other notable exception involved measurements made of the swirl and tumble 
produced by various inlet configurations, and the effect of the swirl and tumble on 
combustion [63], [86]. The flow measurements were made using high speed photography 
of goose down (!) in a glass cylinder. The combustion measurements used schlieren 
techniques. The authors determined the major mean flow and turbulent characteristics of 
swirl and tumble, and their effects on combustion (we will detail this later).

The work of Ricardo (above) and this pre-war work were largely forgotten, and had to 
be rediscovered during the 1980s and 1990s.

An example of an early engine designed largely on an empirical basis is the DOHC 
penta-head engine, which was developed for racing during the teens of the century, and 
was afterward extensively used in aircraft. It was designed principally to maximize the 
valve area, which keeps the Mach index as low as possible, although the Mach index was 
not understood at the time. It was known to be particularly successful, but it was not 
understood until recently (except for the work of Ricardo [45] and the NACA work [63], 
[86]) that the orientation of the inlet valves induces tumble, which is then broken up as the 
piston approaches TC, resulting in high turbulence levels, and high effective flame speeds. 
The central location of the spark plug also gives relatively short travel distances for the 
flame front, which Ricardo [45] understood. The combination of short distances and high 
effective speeds results in short burn times, which means that the compression ratio can be 
increased at the same octave number without knock, one of Ricardo’s basic findings [45].

In modern engines, short burn times can be taken advantage of in other ways. For 
example, higher exhaust gas recirculation rates (EGR) could be tolerated, resulting in 
higher efficiency and lower NOx production. This would be preferable to increasing the 
compression ratio in a modern engine, since an increased compression ratio could raise 
the unburned hydrocarbon emissions, because the crevice volume would become a larger 
fraction of the total volume. We will discuss all these possibilities in due course.

Now, engines are designed in an attempt to consciously bring about some of these 
effects. Unfortunately, our ability to apply existing limited understanding of fluid
mechanics and turbulence to the complicated situation in the cylinder is still fairly rudimentary, and while the process is more rational and less empirical than it used to be, it still has some way to go.

Also, engine designers are trying to do rather ambitious things. These come under the general heading of flow management, with the goal of reducing brake-specific fuel consumption and emissions. There is a complex interaction among the flow management, the characteristics of the catalyst, the emissions, and the fuel consumption.

For example, the lean burn engine burns a homogeneous mixture of perhaps 24:1. Such a mixture is relatively difficult to burn, and requires a high turbulence level. Exhaust gas recirculation (EGR) is another technique to meet the same goal. EGR keeps approximately the same proportions, but replaces some of the excess air with recirculated exhaust gas, which has the advantage that a 3-way catalyst remains effective (current catalysts are not effective when the gases contain an excess of oxygen). Again, high turbulence levels are required to obtain reliable combustion.

The stratified charge engine attempts to segregate the incoming fuel vapor so that it does not mix with all the air in the cylinder, so that the engine will run at an effective air/fuel ratio of perhaps 50:1, while the fuel is confined to a small region of the cylinder volume where the air/fuel ratio is only 15:1, which will ignite and burn. Stratified charge engines now are direct injection engines; that is, they inject the fuel spray at high pressure directly into the cylinder, using various strategies to keep the fuel spray from mixing with the entire contents of the cylinder. In the early development (say, twenty years ago) of the stratified charge engine, various techniques (other than direct injection) were used to segregate the charge, and they were not very effective. We will discuss some of these techniques later. Such engines did not achieve air/fuel ratios much above those of the lean burn engines, and are now thought to have been probably about as homogeneous as a poorly managed homogeneous charge engine. Terms like "nearly homogeneous," or "weakly stratified" might better be used to describe these engines. The suggestion here is, that 'homogeneous' charge engines, unless considerable care be taken to induce homogeneity, are not particularly homogeneous.

Lean burn results in a lower combustion temperature overall, producing fewer oxides of nitrogen. Because the mixture is approximately stoichiometric or slightly rich in the fuel cloud in the stratified charge engine, the oxides of nitrogen are no lower than those of the lean burn engine, although the air/fuel ratio may be much higher. However, combustion quality is poorer than in stoichiometric homogeneous engines (due to slower burn and lower temperature), and as a result the emitted hydrocarbons are higher. CO is lower for the lean burn engine, but rises again for the stratified charge engine, due to the poor combustion. Moreover, lower levels of NO out of the engine do not necessarily result in lower levels of NO out of the tailpipe, because the conversion efficiency of the catalyst is low in the excess air of a lean-burn or stratified charge system. By comparison, EGR offers fuel economy and engine-out NO benefits comparable to a mildly lean combustion system, but allows the use of a three-way catalyst for oxidation of hydrocarbons and CO, and reduction of NO. This is also a cost-effective approach. Note that, for $150 Ryobi offers a 4-stroke-cycle gasoline weed-whacker with EGR - Los Angeles County take note! (Los Angeles County recently passed an ordinance outlawing weed-whackers because of their high emissions level.)

For at least the past fifteen years, the motivation for lean stratified charge engines in the automotive industry has been improved fuel economy. Not only are HC and CO emissions higher, but one has to start worrying about particulates, as in a diesel, because of the poor combustion. The few lean-burn engines that currently meet Federal emissions standards operate lean over only a very small portion of their speed-load map. None of them meets U.S. 50-state standards.
Management of flow in the engine cylinder to bring about any of these effects is very tricky to do, as we will see, and the fluid mechanical details of how it works are poorly understood.

**PHASES OF THE FLOW**

The flow in the cylinder can be divided into several distinct phases. The flow into the cylinder through the inlet valve or valves (forming a jet) does two things: first, the geometrical configuration of the inlet ports and the valves, and their opening schedule creates organized motions in the cylinder, known as swirl (about the cylinder axis) and tumble (orthogonal to the cylinder axis); second, the jet itself is turbulent, and in addition much of the directed (non-turbulent) energy in the jet is converted to turbulence, resulting in a very high turbulence level during the inlet stroke.

During the second half of the inlet stroke much of this turbulence decays, which is to say that the intensity decreases markedly, both because the source (the jet) is coming to an end, and due to the effects of viscosity. The organized motions are carrying fuel vapor and droplets, residual gases, and all the contents of the cylinder, down the sides and across the bottom and up the other side, and round and round (depending on how much swirl and tumble there is). In addition, the turbulence is spreading itself, organized momentum, fuel, residual gases, in fact anything that can be transported, throughout the cylinder, trying to make everything as uniform as possible. This is what turbulence does best (transporting things), and it does it many orders of magnitude better than molecular transport.

During the compression stroke, the increase of density and the changes in length scales (due to the change in geometry of the charge as it is compressed) have the effect of amplifying the turbulence which remained from the inlet jet, although the viscous decay and turbulent transport continue. In addition, the swirl and tumble are affected by the same phenomenon.

If the piston and combustion chamber have been designed to produce squish (that is: if the piston crown approaches very close to some part of the combustion chamber roof), then this will have two effects: first, the fluid squeezed out of the squish clearance volume will produce organized motions, most of which will break up into turbulence; and second, the change in geometry due to the squish will have dynamical effects on the organized tumble and swirl.

Near TC, some of the organized motions may find they have insufficient room to maintain their form, and they will break up into turbulence, increasing the turbulence level. By this time conditions in the cylinder have become crudely homogeneous, due to the transporting effect of the turbulence and the organized motions, unless a concerted effort has been made by the designer to avoid this. Bear in mind that the time available for transport (measured by the time scale of the turbulence itself) is not large, so the uniformity achieved can only be crude. During combustion the turbulence level rises somewhat. Then, during the power stroke, the geometrical changes result in a strong attenuation of the turbulence, and any organized motion that has survived. This, combined with the viscous decay, results in the turbulence being sharply suppressed, so that by the time the exhaust valve opens, there is virtually nothing left. Very little turbulence is generated during the exhaust stroke.

We will examine these various phases one by one.

**AVERAGING**

We need to talk first about averaging, since turbulence is defined by fluctuations about an average velocity.
If we were dealing with a process that was not changing statistically from instant to instant (even though the instantaneous values were chaotic) such as the wind speed at midday on a day without severe changes in the weather, then we could use a time average, integrating over a time long compared to a typical time over which the values fluctuate. Such a situation is called stationary. Unfortunately, the situation in an engine cylinder is not stationary. Successive cycles are similar, but within each cycle the situation is statistically quite different from instant to instant.

In such a situation, it is useful to use a phase average. Imagine measuring, say, the circumferential velocity $U$ at a particular location in the combustion chamber. This will be a function of the crank angle $\theta$ and of the particular cycle in which we measured it, say

$$U(\theta, i)$$

where $i$ indicates the number of the cycle in which it was measured. Then the phase averaged value is defined as

$$\overline{U}(\theta) = \frac{1}{N} \sum_{i=1}^{N} U(\theta, i)$$

(5.2)

where $N$ is the number of measurements available. There are a number of interesting questions which we could address, such as, how many individual values do we need to include in Equation 5.2 to arrive at a stable value? This and many other questions are covered in [96], and need not concern us here.

Now that we have a convenient definition of an average value, which is a function only of crank angle, we can define the turbulent fluctuating velocity as the difference between the instantaneous value and the average value:

$$u(\theta, i) = U(\theta, i) - \overline{U}(\theta)$$

(5.3)

Statistics of $U(\theta, i)$ can now be defined as (for example)

$$\overline{u^2}(\theta) = \frac{1}{N} \sum_{i=1}^{N} u^2(\theta, i)$$

(5.4)

the mean square value.

Note that the mean values defined in this way are functions only of the crank angle. There is another way to define an average in this situation. Consider the time scale of the turbulence. For example, we will find below that the turbulence during the middle third of the intake stroke has an intensity (the root-mean-square value, from Equation 5.4) of roughly $10 \overline{V_p}$, where $\overline{V_p}$ is the average piston speed, and a length scale of roughly $b/6$ [98], where $b$ is the bore. This means it has a time scale of roughly $\frac{b}{60\overline{V_p}}$. This time scale is $1/60$ the time it takes the piston to complete the intake stroke (presuming that the engine is nearly square, $b = S$). Now, what can we do with this? It tells us that there is a time, say $S/\sqrt{60\overline{V_p}}$, that is about $\sqrt{60} = 8$ times larger than the times typical of the turbulent fluctuations, and yet shorter than (about $1/\sqrt{60} = 1/8$) the time of the intake stroke.

$$\frac{S}{60\overline{V_p}} \ll \sqrt{60} \frac{S}{60\overline{V_p}} = \sqrt{\frac{S}{60\overline{V_p}}} \ll \frac{S}{\overline{V_p}}$$

(5.5)

where $\sqrt{60} = 8$. We can use what is called a moving average:

$$\langle U(\theta, i) \rangle = \frac{1}{\theta} \int_{-\theta/2}^{+\theta/2} U(\theta + \psi) \, d\psi$$

(5.6)

where $\theta = S/8 \overline{V_p}$. We can call this the individual cycle mean. Figure 5.1 shows a cartoon of such an instantaneous velocity, the phase average, and the individual cycle mean.
This question of whether there is an intermediate time scale, larger than the small scale of the turbulence, but shorter than the longer scale of the intake stroke, is the same question that arises in meteorology, whether there is an intermediate scale between turbulence and weather, that would permit a distinction between them. It also arises in viewing pictures in a newspaper. The distance between the Benday dots that make up the picture is the fine scale, and the scale of the detail in the picture is the large scale. For the picture to be interpretable by our eye, a scale must exist that is large compared to the Benday dot scale and small compared to the scale of the detail. The situation is only unambiguous when there is a wide separation between the scales. There are many interesting statistical questions that can be asked about this situation - some of them were discussed in [66].

Heywood [47] says there is considerable debate in the literature over whether the cycle-to-cycle variation in the individual cycle mean is a physically distinct phenomenon from the turbulence, which we have defined as the departure of the instantaneous velocity from the phase average, but which could also have been defined as the departure of the instantaneous velocity from the individual cycle mean. The way we have defined it, it includes both this other definition of turbulence, and the cycle-to-cycle variation.

Cycle-to-cycle variability can be related to the turbulence in the inlet manifold. To begin with, we have evidence that the flow in the cylinder at TC (and after) is very sensitive to the velocity field in the cylinder at the moment the inlet valve closes, which is determined by the flow in the inlet manifold while the cylinder was filling. Exponential dependence on small differences in initial conditions is a characteristic of turbulence [49]. In the case of flow in the cylinder, the dependence on the initial conditions is known both from computational evidence and from experimental evidence. In [80] the authors examine the flow in a single cylinder engine with a pancake-shaped combustion chamber, with an inlet and exhaust manifold. The inlet valve is shrouded, to produce high swirl; such a configuration also has relatively little cyclic variability of the large scales. The authors calculated the mean flow structure, modeling the small scales of the turbulence. They found that the swirl ratio, and the phasing of the swirl structure 90 CAD ATC was very sensitive to the initial velocity field at inlet valve closing.

In [79], the author made measurements of the flow in the same engine, this time without the shroud on the inlet valve, so that the flow is undirected. The measurements were made by introducing very small particles that follow the flow, and photographing them at closely spaced times, obtaining velocity vectors from the differences in position. This is known as Particle Image Velocimetry, or PIV. In Figure 5.2 I show this flow. From Figure 5.2 it is clear that there is substantial variation in the large scale structure at TC. This variation in large scale structure will affect the flame propagation and heat release, and must consequently be responsible for the cycle-to-cycle variability.
In [43] the author carried out preliminary calculations of the unsteady flow in this same engine, including the inlet and exhaust manifolds, resolving only the largest scales (modeling the smaller scales). Again, he found large variability of the large scale structures, completely consistent with the PIV measurements and the mean flow calculations.

The flow is deterministic, and the flow at TC is entirely determined from (and very sensitive to) the flow at the inlet valve closing. Since this is entirely determined by the flow in the inlet manifold during filling, it must be this flow that is responsible for the cycle-to-cycle variability. The flow in the inlet manifold is very compressible as we have seen, and large amplitude sound waves are present in the manifold. It might at first be thought that these were responsible for the fluctuations in manifold flow. Measurements of the pressure fluctuations in the manifold of a 2.2 L Renault J7T (four cylinders, with two valves per cylinder) at several engine speeds [14], however, show that the pressure fluctuations (and hence the sound waves) are quite periodic; the very small differences from cycle to cycle are below the resolution of the pressure transducer.

We have to conclude that the differences in the initial velocity field in the cylinder when the inlet valve closes are attributable to the turbulent velocity fluctuations in the inlet manifold. These may have an indirect influence also, since the velocity fluctuations affect the pressure drop across the inlet valve during filling, and may have a small effect on the sound waves in the manifold. The relatively small differences at inlet valve closing attributable to the manifold turbulence are enough to cause considerable variability at TC, due to the extreme sensitivity of the turbulence to initial conditions.

Note that the various kinds of average (the phase average, and the moving average, which could also be applied in space within the engine cylinder) are not statistically equivalent. It is necessary in any discussion to be clear about which average is being discussed. Since the cycle-to-cycle variation and the small scale turbulence in the cylinder appear to be physically the same, differing only in scale, it probably makes sense to lump
them together, because the cycle-to-cycle variations are also responsible for transport in the cylinder. In the rest of this book, I will use the definition of turbulence as the departure from the phase average.

This does not mean, however, that the cycle-to-cycle variation should be ignored; it is responsible for unevenness in the running of the engine, and is generally undesirable for that reason. In addition, if there is cycle-to-cycle variation, some of the cycles will be of lower efficiency, and this will affect the average efficiency. I would rather describe the situation this way, however: that it is desirable to suppress as much as possible the largest scale variability of the flow entering and in the cylinder (which will be responsible for perceptible unevenness and fluctuation in efficiency, the longer, the more perceptible). This says the same thing, but the emphasis is a little different.

**A WORD ABOUT TURBULENCE**

The reader will need to understand just a little about turbulence for this discussion to make sense. It will not be necessary to consult a text on the subject, such as [96].

All fluid flow is either laminar or turbulent. The flow of pancake syrup is an example of laminar flow - smooth, steady and uncomplicated. When the kitchen faucet is turned on full and the flow leaves the faucet in an unsteady, rough, chaotic manner, filled with small eddies, that is turbulent flow. Most of the flow in the universe is turbulent.

Whether a flow is laminar or turbulent is usually (and certainly in our case) determined by a Reynolds number, which compares inertial to viscous forces. If \( U \) (m/s) is a typical velocity in the flow, \( L \) (m) a typical length, and \( \nu \) (m²/s) the kinematic viscosity, a Reynolds number is defined as \( UL/\nu \). It is dimensionless. In any given flow, there are usually several Reynolds numbers that can be defined, depending on what velocity and length scales are used, but they are all related. The flow of pancake syrup is dominated by viscous forces, while the water flow is not. The Reynolds number of the pancake syrup is low, while the Reynolds number of the water flow is high. If the Reynolds number is large, the flow will probably be turbulent.

As the Reynolds number increases, a laminar flow becomes unstable. That means, that small disturbances, which would have been suppressed by viscosity when the Reynolds number was smaller, will instead be amplified, and the flow will become chaotic and unsteady. In this way, energy in the mean flow is transferred to the turbulence.

Turbulent flows contain a wide range of scales - that is, the velocity field contains motions of all sizes, from eddies that are essentially large enough to fill the space available, in our case the engine cylinder, down to eddies often substantially below a millimeter in size. The size of the largest eddy can be guessed by asking for the diameter of the largest sphere that will fit in the available space since turbulent eddies are approximately the same size in all directions. Hence, at TC the largest eddy will be roughly the clearance height, while at BC it will be roughly the cylinder bore. The kinetic energy of these eddies varies: the largest eddies are relatively weak; as the size drops, the energy rises rapidly to a peak, and then falls continually down to the smallest eddies. The most energetic eddy, at the peak, which is responsible for most of the transport, is about 1/6 the size of the largest eddy - thus, 1/6 of the bore, or 1/6 of the clearance height [98].

Turbulence consumes energy. Mean flow energy is converted to large scale turbulence and those large scale motions are themselves unstable, and break down into smaller scales, and those in their turn break up into still smaller scales, until the smallest scale is reached. It is a curious experimental fact that the amount of energy consumed in a turbulent flow has nothing to do with the viscosity, but is determined entirely by the large
turbulent scales. That is, the amount of energy taken from the mean flow is determined by
the process of creating the largest turbulent eddies. This amount of energy is passed on to
the next smaller eddies, and on to still smaller eddies, and is finally dissipated by viscosity
at the smallest scales. The amount of energy passed from the mean flow to the largest
eddies and on to eddies of progressively smaller size is called the dissipation, the energy
consumed per unit mass of the flow, and is designated by $\varepsilon$. If $u$ is a turbulent velocity
scale, the root mean square fluctuating velocity (typical of the most energetic eddies), and
$\ell$ is a length scale of the most energetic eddies, then it is an experimental fact that

$$\varepsilon = \frac{u^3}{\ell} \quad (5.7)$$

You can understand this by writing $u^2 = u^2(l/l)$. $u^2$ is proportional to the energy of
the turbulence, and $l/u$ is proportional to the natural time scale of the most energetic turbulent
eddies, which determines how fast the most energetic eddies lose energy to the next size
eddies. From this you can see that $u^2/l$ is like $du^2/dt$.

The dynamical behavior of the smallest eddies is determined entirely by the rate at
which they are fed energy, $\varepsilon$, and the size of the kinematic viscosity $\nu$. The small eddies
are where the mechanical energy is converted to heat, although the rate of conversion is
determined by the large scales. If the viscosity is raised, $\varepsilon$ does not change, but the scale at
which the dissipation takes place increases. The smallest scale is called the Kolmogorov
microscale, designated by $\eta$, and defined as

$$\eta = \left( \frac{\nu^3}{\varepsilon} \right)^{1/4} \quad (5.8)$$

Smaller scales cannot exist because the viscous transport smears them out. In most flows
the Kolmogorov microscale $\eta$ is a few tenths of a millimeter. We will find that in an engine
cylinder, it is of the order of 0.01 mm, or 10 $\mu$m.

As noted above, in an engine cylinder, we expect the most energetic scale to be of
the order of $h/6$ if the piston is near TC, or of the order of $b/6$ if the piston is halfway down
or more. Consider a cylinder halfway between TC and BC on the intake stroke, and
suppose the engine is running at a piston speed of 5 m/s. Assume an inlet density of $\rho =
0.5$ kg/m$^3$, and suppose $S=0$. We will find in Section 5.5 that the turbulent velocity when
the piston is in the middle of the intake stroke is about 10 times the mean piston speed, or
perhaps 50 m/s here. We expect that $\varepsilon = b/6$, and let us take $b = 82.6$ mm. This then gives us an
$\varepsilon = 9$ MW/kg, giving $\eta = 7.4\times10^4$ m = 7.4 $\mu$m. As turbulent flows go, in the environment
and the laboratory, this is extraordinarily small; $\eta$ in the atmosphere or the ocean is about
1 mm, and in laboratory experiments not much smaller than a fraction of a millimeter.
The value of $\varepsilon = 9$ MW/kg sounds enormous, but bear in mind that the mass in a cylinder
is less than a gram, that this level of dissipation does not last long: roughly $\frac{2\eta}{3}p = 5.5$ ms,
and it results in a temperature rise of less than 50 K. For higher piston speeds $\varepsilon$ rises very
fast, since $\varepsilon \propto \frac{V}{p}$. $\eta$ will fall slowly as $\eta \propto \frac{V}{p}^{3/2}$.

We can look at the cylinder at TC. Now $\varepsilon$ is 1/6 of the clearance height, $h$. If $r = 8$, the
clearance height is approximately $h = 11.8$ mm. Hence, $\varepsilon = \frac{h}{6} = 2$ mm. We have already
seen in Section 3.3.4 that at TC, in the absence of swirl, tumble and squish, $u = \frac{4V}{2p}$.

The kinematic viscosity in a gas is proportional approximately to $\frac{\tau}{p}$. With $r = 8$, the
isentropic temperature at the end of compression has increased by a factor of 2.3, while
the density has increased by a factor of 8, giving a value for the kinematic viscosity of $\nu =$
2.84 $\times 10^4$ m$^3$/s. With $V_p = 5$ m/s, this gives $\epsilon = 7.9$ kW/kg, and a value of $\eta = 7.34$ $\mu$m, still about the same size, and quite small by non-automotive standards.

Notice, if we use both our Equations 5.7 and 5.8, we can write where $Re = u \epsilon \nu$ is the Reynolds number of the turbulence, based on the turbulent fluctuating velocity and the length scale of the energy-containing turbulent eddies. Notice that this Reynolds number determines the ratio of the energy-containing and small scales

$$\frac{\eta}{\ell} = \left( \frac{v^3}{\epsilon \ell^4} \right)^{1/4} = \left( \frac{v^3}{u^2 \epsilon^3} \right)^{1/4} = Re^{-3/4}$$

(5.9)

in the turbulent flow; when $Re = 1$, the energy-containing and small scales are approximately the same, and at this point the turbulence can no longer maintain itself. To exist, turbulence needs to have a range of scales; when the small scale and the energy-containing scale are the same, this means that viscosity acts directly on the energy-containing scales, and that makes the turbulence too dissipative, and it dies. At our two states, halfway down on the intake stroke and at TC, we have scale ratios of $\eta \ell = 1.9 \times 10^3$ and $1.6 \times 10^2$, for turbulent Reynolds numbers respectively of $Re = 2.3 \times 10^4$ and $Re = 8.7 \times 10^3$.

In a gas, $v$ measures the molecular transport of momentum, $v = c_\lambda$, where $c$ is the root-mean-square molecular thermal velocity, and $\lambda$ is the mean free path. That is, $v$ is proportional to the product of the velocity scale and the length scale of the process responsible for the property transport, the molecular motion. In a turbulent fluid, the turbulence transports properties, by simply carrying the fluid containing them to a new location. This transport process is nothing like (physically or mathematically) the molecular transport; nevertheless, it is often convenient to parameterize it by what is called an eddy viscosity, $\nu_t$. Just like the molecular transport, $\nu_t$, the turbulent transport coefficient, is also proportional to the product of the length and velocity scales of the physical process responsible for the transport, the turbulence. Hence, $\nu_t = u \ell$. We can write

$$\frac{\nu_T}{\nu} = \frac{u \ell}{v} = Re$$

(5.10)

so that, in our two representative situations (halfway down on the intake stroke, and at TC on the compression stroke) the turbulence is approximately $10^4$ and $10^3$ times as effective at transporting momentum (and everything else) as the molecular motion. Values of $\nu_T$ in the two situations are $\nu_T = 6.9 \times 10^{-1}$ m$^2$/s and $\nu_T = 1.4 \times 10^{-2}$ m$^2$/s, far larger than the molecular values.

During a time $t$, the turbulence will carry a property (momentum, for example) a distance $L$ of roughly $[96]$

$$L \approx \sqrt{\frac{2}{3} \nu_T t} = \sqrt{\frac{2}{3} u \ell t}$$

(5.11)

Consider the situation half-way down on the inlet stroke. If we measure $t$ in terms of the time to complete the inlet stroke, say $1/3$ of that time, when the turbulence is most intense, $t = S/3V_p$, then

$$\frac{L}{b} \approx \sqrt{\frac{2}{3} \frac{u \ell}{b^2} t} \approx \sqrt{\frac{2}{9} \frac{u \ell S}{b V_p}} \approx 0.6$$

(5.12)
Thus, this intense turbulence is capable of transporting momentum across a substantial fraction of the engine cylinder in the time available. Note that it transports fuel vapor (or anything else) just as well as momentum, since the transport is physical - the fluid containing the momentum or fuel vapor is simply moved to another place, so the effectiveness of the transport does not depend on the nature of the thing being transported, and it works as well for all properties.

On the other hand, at TC, the turbulence is much weaker, and the time available is much shorter. The burn time corresponds to roughly $0.28 \sqrt{S/\bar{p}}$, so that

$$\frac{L}{h} \approx \sqrt{\frac{2}{3} \frac{u \ell S}{\bar{V}_p h h}} \approx 0.33$$

so only a fraction of the clearance height can be covered in the time available.

There are, of course, other factors involved, and these are extremely approximate values, intended only to give the flavor of what is going on. It is clear, however, that the turbulence available during the burn is marginal, considering that it is primarily responsible for the flame propagation. It is equally clear that it will be very difficult to keep the fuel vapor segregated during the inlet stroke, in the face of the aggressive transport properties of the turbulence available then.

ESP (see Chapter 8) assumes that the charge is homogeneous - that the aggressive transport during the intake stroke has resulted in uniformity. To calculate the turbulence level, it uses turbulence models that are (in some respects) a little more sophisticated than our crude approximations, and in others less so.

**TURBULENCE INDUCED BY THE INLET JET**

I will assume here that we are discussing a homogeneous charge engine, without significant swirl or tumble, which I will address later.

The flow through the inlet valve forms a hollow conical jet, roughly at the angle of the valve seat. We may expect that the Reynolds numbers will in general be high, since the viscosity of air at room temperature is relatively low. Using our estimate for the average velocity through the inlet valve Equation (2.15), and the width of the gap (assuming a valve seat angle of 45°), taking conditions at idle, $\bar{V}_p = 2 \text{ m/s}$ and $p = 0.4 \text{ bar}$, we get a Reynolds number of about $6 \times 10^5$, which will still guarantee that the jet is turbulent (since the Reynolds number based on the turbulent velocity and length scales $u$ and $\ell$, which is the one that counts, will be in the neighborhood of $6 \times 10^5 \gg 1$, [96]). The Reynolds number will be much larger at higher piston speeds. This jet is issuing into an enclosed space, so it strikes the piston crown and the cylinder walls. It forms secondary vortices from the interaction with these surfaces, and these ultimately are dissipated or break up into turbulence. The impinging jet (as it is called) also forms turbulent boundary layers on the surfaces. The jet is itself unstable, and flutters back and forth irregularly, transforming the mean flow energy in the jet into turbulent energy.

We can make a crude estimate of the total energy introduced by the jet. If this energy were all to be converted without loss to turbulence, and spread uniformly through the cylinder, it would give us a rough estimate of the peak turbulence intensity we could expect during the inlet stroke. It would be a considerable overestimate, because we are assuming no losses; in Section 5.4 we saw that the losses (dissipation $e$) are large. We can begin from our estimate for the mean velocity through the inlet valve, Equation (2.15).
Now, we want to calculate the total kinetic energy that enters the cylinder in the jet through the inlet valve. Let us make our estimate when the jet has half-filled the cylinder, in the middle of the inlet stroke, when the turbulence from the jet is at its peak.

\[
\text{Total kinetic energy} = \frac{\text{kinetic energy}}{\text{volume}} \tag{5.15}
\]

The first term is

\[
\text{kinetic energy} = \frac{V_r^2}{2} \tag{5.16}
\]

The second term is the total volume that has entered the cylinder, which is just half the displacement volume \( Vd/2 \). Hence, the total kinetic energy is

\[
\text{total kinetic energy} = \frac{V_r^2}{2} \frac{V_d}{2} \tag{5.17}
\]

If this energy is spread uniformly over the cylinder volume, we get a total energy of

\[
\text{total tur}b \text{ energy in cylinder} = \frac{V_r^2}{2} V_d \tag{5.18}
\]

Equating expressions 5.17 and 5.18, we have

\[
\frac{u}{\sqrt{2}} = \frac{V_r}{V_d} \tag{5.19}
\]

so that

\[
\frac{u}{\sqrt{2}} = \left( \frac{b}{D} \right)^2 \frac{1}{C_i} \frac{1}{\sqrt{2}} \tag{5.20}
\]

In [47] the peak value of \( u/\sqrt{2} \) is given as about 10. Using Equation 5.20, estimating \( C_i = 0.35 \), and using the valve and cylinder diameters in [47], we obtain 12.4.

Obtaining a value that close from my simple calculation must be regarded as largely an accident. I neglected a great deal, notably the dissipation, which we have already seen in Section 5.4 is quite large, and should result in a considerable decrease in the turbulent velocity.

We have been considering an engine with one intake valve per cylinder, for the sake of simplicity. In a modern engine we are likely to have two. Roughly speaking, this will cut the velocity through the jet by a factor of two, and hence also cut the turbulent velocity by a factor of two. As piston speed drops, the turbulent velocity may no longer be large enough to provide a short enough burn time to avoid knock. The solution is to open only one of the intake valves at lower speeds, to bring the jet velocity, and the turbulent velocity up again. This is what Honda does, for example, in the VTEC engine (although that is also done to induce swirl).

Before we move on, we should consider decay of turbulence. The turbulent energy balance looks like this:

\[
\frac{dq_t^2/2}{dt} = P + T - D \tag{5.21}
\]

where \( q_t^2/2 \) is the turbulent mean kinetic energy per unit mass, \( P \) is production (the rate at which energy is being extracted from the mean flow and fed into the turbulence), \( T \) is transport (the rate at which energy is being moved to another place by the turbulence itself) and \( D \) is dissipation, \( D = e \), the rate at which the turbulent kinetic energy is being converted to heat by viscosity. We do not need to concern ourselves with the mathematical forms of these terms now. During the inlet stroke, the turbulence away from the direct path of the inlet jet is receiving energy from both \( P \) and \( T \). After the inlet valve has closed, the turbulence in the cylinder is relatively homogeneous, due to the effective
transport properties of the intense turbulence during the inlet stroke. In fact, during the last third of the inlet stroke, the jet through the inlet valve is much weaker, and the production and transport are much less important. We have seen that the dissipation term can be very large. This term will result in a rapid decrease of the turbulent kinetic energy, in the absence of terms feeding the energy. We can make a rough estimate of how fast this can happen, if we write \( \frac{d3u^2/2}{dt} \approx 3\frac{u^2}{l} \) so that

\[
\frac{d3u^2/2}{dt} = -\frac{u^2}{l} \tag{5.22}
\]

We are supposing that the turbulence in all directions is equally intense (a property called isotropy). This is probably not true, but is close enough for a crude estimate — we are interested only in orders of magnitude here. We may easily solve Equation 5.22 if we assume that the turbulent length scale \( l \) remains constant, to give

\[
\frac{u}{u_0} = \frac{1}{1 + \frac{u_0 t}{12}} \tag{5.23}
\]

where \( u_0 \) is the initial value of \( u \). If we take \( u_0 = 10\overline{V_p} \), and \( l = b/6 = S/6 \), then we have

\[
\frac{u}{u_0} = \frac{1}{1 + 20\frac{u_0 t}{S}} \tag{5.24}
\]

If we ask, "How long does it take to reduce \( u/u_0 \) to one-half its initial value?", we easily find \( \overline{V_p} t/S = 0.05 \). If \( \overline{V_p} t/S = 1.0 \), the intensity \( u/u_0 \) is reduced to approximately 0.05 of its initial value. This is something like the correct value. Actually, the turbulent intensity is observed to drop to about 0.15 by the end of the intake stroke, and then only drops to 0.1 during the entire compression stroke. We have left out some important mechanisms: the attenuation due to the expansion during the last third of the intake stroke, and the amplification for the same reason during the compression stroke. ESP does this calculation correctly; all we are trying to do here is get a feeling for the order of magnitude for some of these effects.

**INDUCING SWirl AND TUMBLE**

By the way in which the valves and ports are arranged, and the schedule of valve opening, mean flows can be induced in the cylinder. The flow into the cylinder is turbulent and the mean velocity is often smaller than the turbulent velocities. Motions like this are often called coherent, meaning that they are organization buried in the disorganized turbulence. When measurements are made, velocity patterns must be measured during a number of cycles (perhaps 20-30) and the results averaged to find this organized part. I reproduce here two figures from [105], Figures 5.3 and 5.4, which illustrate schematically tumble and swirl, two types of coherent motion that can be induced in the cylinder. It is also possible to induce a combination of the two motions. In fact, it is essentially impossible to generate swirl without inducing some tumble, so that the two are always associated.
It is possible to generate tumble without swirl. However, tumble is always associated with other secondary motions, since it is generated by flow through two valves. We picture in Figure 5.5 schematically this secondary motion. This secondary motion has the effect of isolating the fluid entering through the \( y > 0 \) valve (in Figure 5.5) from the fluid entering through the \( y < 0 \) valve, something we will return to later. As we have indicated in the figure, the size of this secondary motion is approximately \( b/2 \), and the intensity is also about one-half of the intensity of the main tumbling motion. Swirl and tumble, or a combination of the two, represent the most general motion that can be induced at the scale of the cylinder. It is clear from the existence of the half-scale motion associated with tumble, that many more complicated motions are possible at smaller scale.
There are a number of reasons for inducing swirl and/or tumble. High turbulence levels at ignition produce higher effective flame speeds, and more reliable combustion at very lean air/fuel ratios, or with EGR. At normal air/fuel ratios, the higher speed will allow the flame front to reach the end gas before the chemical reaction resulting in auto-ignition has time to take place, permitting higher compression ratios without knock. This was the motivation in the early days of engine development, when fuel octane numbers were low. Now the motivation is more likely to be reliable combustion with very lean fuel/air ratios, or with EGR. In any event, one possible reason for swirl or tumble is to promote high turbulence levels at ignition.

Tumble can also be used for stratification. We will discuss this later. As we have seen, the turbulence resulting from the conversion of the energy in the inlet jet decays rather fast, and not much is left at ignition. The idea of swirl or tumble is to encapsulate some of the momentum of the inlet valve jet in the organized motion (swirl or tumble), which is less dissipative than the turbulence (because of larger spacial scale), and hence will retain its energy longer. In addition, the vorticity in the tumble can be augmented by the compression process, or by squish in the case of swirl. Then, just before ignition the swirl or tumble can be induced to break up into turbulence, producing a much higher level of turbulence at ignition than would be present just from the decayed turbulence from the inlet jet.

![Figure 5.5. Schematic of secondary motion induced in tumble. Courtesy D. Haworth.](image)

Tumble appears always to break down to turbulence, because as the piston approaches TC, there is not room between the piston crown and the cylinder head for a
vortex with a diameter of the order of \( b \); only motions with scales of the order of the clearance height can survive, so the vortex breaks up into turbulence of this scale. In the case of swirl, if the combustion chamber is pancake-shaped, the swirl can survive through the burn, and we will talk about that in a moment. However, if the combustion chamber is a penta-head, with squish providing the transition from cylinder to head, and/or with the piston crown protruding into the head, the swirl must accommodate itself to the changing shape of the space available to it (from circular to rectangular, and increasingly narrow), and will also break up into turbulence.

In fact, in practical open-chamber four-valve pent-roof engines, squish is a minor factor. There is simply no room with four valves and a spark-plug to have any significant squish area. The same is true in modern two-valve engines. Hence, swirl survives to TC, and does not break up into turbulence. It is now clear from computational fluid dynamics and flow visualization that even in engines that were designed for high swirl, is principally the associated, unavoidable, tumble that yields the turbulence just before TC. There were apparent correlations between swirl and burn rate, but these appear to have held only because there was always tumble with the swirl.

The only exception to this is the high compression bowl-in-piston engine, where swirl plus the squish do result in spin-up, and the generation of turbulence.

The breakup of the tumble just before TC is not well understood, and is probably quite interesting from a fluid mechanical point of view [29], [38], [101]. Experiments on rotating flow in an ellipsoid show that rotation around the intermediate axis is not stable, and will change over to rotation around the minor axis if the ellipsoid is sufficiently flattened - that is, if the ratio of the axes is large enough. When the flow changes over, a strongly chaotic, turbulent, flow is generated. In an engine, this suggests that tumble, as the piston rises and the space available becomes more flattened, will change over to swirl, with the generation of strong turbulence. The losses will be large, probably enough so that the swirl generated will be very small.

The other possible reason for inducing swirl is to stratify the charge; that is, to keep the fuel-rich charge segregated, so that it does not mix with the remainder of the air in the cylinder. In order to do this, it is necessary to damp some of the turbulence, selectively. It turns out (as we shall see) that one of the effects of swirl is to damp strongly the turbulence near the edge of the swirl, and if the combustion chamber is pancake-shaped, so that the swirl can persist until ignition, the swirl vortex at ignition consists of a high turbulence core surrounded by a very low-turbulence annulus. If the fuel-air cloud can be induced to enter this low-turbulence annulus (and it can, as we will see) then it will stay there, and will not be spread throughout the cylinder. Ignition will not be a problem, because the fuel/air mixture is locally near stoichiometric in the cloud. At ignition, the flame front will at first be in the low-turbulence annulus, and will spread slowly, but it will soon leave for the leaner mixture in the high-turbulence core, where it will burn reliably and spread rapidly.

Another type of stratification associated with swirl is important in direct injection engines. The hot residual gases are considerably lighter than the cold incoming charge (perhaps 0.25 of the density), and will move toward the axis of rotation. Fuel vapor is of considerably higher density than air, and with direct injection engines it is possible to obtain a rich mixture with a density ratio of perhaps 1.5-2.0, which will gravitate to the outside of the swirl. Whether these will stay put depends on whether the density gradients are large enough to damp the turbulence, which would otherwise mix them. These density gradients are definitely large enough.

Even if the swirl is broken up by squish and a penta head, if this can be delayed until close enough to ignition, a similar scenario will hold. At ignition, the turbulence will be
high, so flame propagation will be rapid. What is important is to not provide enough time for the turbulence to spread the fuel cloud before ignition. If breakup can be delayed until roughly 15 CAD before ignition, then the resulting turbulence cannot spread the fuel cloud more than 1/3 of the available vertical distance (perhaps 0.25 S) before ignition.

Swirl and tumble are usually present together. The combination of tumble and swirl results in a tilted axis of rotation of the secondary motion in the cylinder, and this tilted axis precesses. So long as the tilt of the axis is not too great (say, 1:3), the suppression effect on the turbulence near the edge of the swirl is still present [80].

Both swirl and tumble are normally specified by a swirl ratio, or tumble ratio. In either case, the angular velocity of the solid-body rotation with the same angular momentum as the actual velocity distribution in the swirl or tumble is compared to the angular velocity of the crankshaft.

\[
\frac{R_s}{2\pi N} = \frac{\omega_s}{2\pi N}
\]

\[
\frac{R_t}{2\pi N} = \frac{\omega_t}{2\pi N}
\]

Where \( R_s \) and \( R_t \) are respectively the swirl ratio and the tumble ratio, and \( \omega_s \) and \( \omega_t \) are respectively the angular velocities of the solid-body rotations that have the same angular momentum as the real swirl or tumble flow. \( N \) has units of revolutions/sec.

Production engines usually have values of the swirl or tumble ratios between 1.0 and 2.0. Experimental engines achieve numbers up to perhaps 6.0. If we call the tangential velocity at the edge of the solid-body rotation \( v_0 \), so that \( \omega_s = 2 v_0 / b \), then we can write

\[
\frac{R_s}{2\pi N} = \frac{v_0 S}{b/2} = \frac{2}{\pi} \frac{S}{b} \frac{v_0}{V_p} \]

Hence

\[
\frac{v_0}{V_p} = \frac{\pi}{2} \frac{b}{S} R_s
\]

For a square engine, roughly 3/2 of the swirl ratio \( R_s \) gives the ratio of the tangential velocity to the mean piston speed. Thus, in production engines we expect tangential velocities below three times the mean piston speed, while experimental engines may achieve nine times the mean piston speed. We can say the same things for tumble.

The swirl and tumble ratios are normally measured in steady flow on a test rig consisting of the cylinder head and valves, with a tube replacing the cylinder. A paddle wheel, or similar device is placed at the exit from the tube, and the rotation of the device is used as a measure of the swirl or tumble. This approach is sound for swirl, but is less satisfactory for tumble. It is used because measurement of the actual swirl or tumble in a motoring engine, or an operating engine, is extremely difficult and expensive. This can be done, using various optical techniques (particle image velocimetry [80], laser doppler velocimetry [100]) if an experimental engine has been specially prepared for optical access to the cylinder. The match between the steady flow swirl ratios and the swirl ratios measured in a motoring engine is good if the actual velocity distribution is integrated to obtain the angular momentum. For example, [80] had a steady flow swirl ratio of 6.0, and the PIV results build from near zero at TC on the intake stroke to about 6.5 at BC, then fall slowly to about 5.0 at TC on the compression stroke. On the other hand, estimating the swirl ratio from the maximum tangential velocity gives low values; this gives 2.93 instead of 5.0 for the conditions of [80]. In [100], at TC the swirl ratio (estimated from the maximum tangential velocity, measured by LDV) was 1.27, while the steady flow value was about 2.6. At 150 CAD before TC, the estimated value was a little closer, at 1.75. It
appears that estimating the swirl ratio in this way (from the maximum tangential velocity) gives about 0.6 of the true value. This number would vary for different velocity distributions in the swirl; however, the velocity distributions are usually quite similar, at least for open chamber four-valve pent-roof engines. Presumably the same remarks can be made about the tumble ratios. For two-valve engines, and/or bowl-in-piston designs, it is more difficult to generalize.

Swirl and tumble ratios can be obtained quite accurately for both production and research engines using computational fluid dynamics. This is not a cheap solution, however, because it requires the construction of a grid for the inlet manifolds, ports, valves and combustion chamber, unless this has been constructed for some other purpose. Although the swirl velocity field is often more complex during the intake stroke and early on the compression stroke, by TC of the compression stroke, the mean velocity field has settled down to a single vortex, centered in the cylinder. I include here (Figure 5.6) PIV measurements of the velocity field at TC from [80]. Bear in mind that this is a particularly simple engine.

**Lift Strategies**

As can be seen from Figure 5.3, the four-valve head lends itself to producing tumble, which can be optimized by modifying the angle of the inlet runners. This is probably one explanation for the popularity of the four valve head, even though other valve arrangements might have heat transfer or Mach index advantages. On the other hand, if both inlet valves are opened, it does not produce swirl (presuming that the head is reflectionally symmetric about the plane between the two valves). In order to produce swirl, it is necessary to keep one of the inlet valves closed, or nearly so. In [105], they experimented with variable valve lift in a 79 mm × 76.2 mm engine at \( V_p = 5 \) m/s. With equal lift they attained \( R_s = 0 \); with one valve closed, they obtained \( R_s = 0.85 \). The \( R_s = 0.52 \) with the valves equally open, but rose to about \( R_t = 1.0 \) with one closed. In the Honda VTEC engine, for example, this strategy is used to obtain high swirl and tumble. Usually, the valve that is not opening is actually opened just a crack to allow the fuel sprayed by the injector into the port to enter the cylinder. Because the air flow is very
small, this is sometimes used to keep a fuel-rich cloud near the cylinder head, to produce a stratified charge.

**Port and Valve Configurations**

Swirl can also be generated by the configuration of the inlet port, or by shrouding the valve, or masking it.

I reproduce here Fig. 5.7 from [47], showing various inlet port configurations, which produce similar values of swirl ratios, between 2.5 and 2.9. The deflector wall port uses the port inner side wall to force the flow preferentially through the outer periphery of the valve opening, in a tangential direction. The directed port brings the flow toward the valve opening in the desired tangential direction (see [47]). These various ports pay various penalties in lower discharge coefficients. The helical ramp ports appear to have higher discharge coefficients, because the whole periphery of the valve open area can be used [47]. The directed port, with its straight passage, has the most restricted flow area, and the lowest discharge coefficient. Since only one wall is used in the deflector wall port, the area is less restricted, and the discharge coefficient is higher.

Port design could probably be approached rationally. For example, it seems possible that there is a design producing a minimum discharge coefficient for given swirl or tumble, a design that could be found by variation of design parameters using computational fluid mechanics. However, in real engine design the final (suboptimal) port choice is made based on non-fluid-mechanical considerations of packaging, fuel injector targeting, and the like, so that concern with optimal ports is probably not productive.

Another way of inducing swirl is by shrouding or masking the valve. Figure 5.8 from [47] illustrates shrouding and masking.

Shrouding is often used on experimental engines, but never on production engines, since it requires that the valve be prevented from turning. Normally, a valve is turned slightly by the lifter each time it is raised, and this has advantages in uniformizing valve head temperature and seat wear. Some special provision has to be made to prevent this rotation if shrouding is used. Masking avoids this problem, and can easily be included in a production engine. However, both shrouding and masking have the disadvantage that the effective valve open area is reduced, which reduces the volumetric efficiency and increases the Mach index.
EFFECT OF COMPRESSION

To understand what happens to either swirl or tumble during compression, we have to talk about fluid mechanics for a moment. Conservation of angular momentum is the principle that we will apply. We are all familiar with the illustration of the ice skater who spins on one toe with her arms outstretched, and then pulls in the arms, reducing the radius of gyration (as it is called), and increasing the angular velocity substantially. This is conservation of angular momentum - her angular momentum with arms out and with arms in is the same, but the moment of inertia is larger with the arms out, and smaller with the arms in, so the angular velocity is smaller with the arms out, and larger with the arms in.

In the engine cylinder, the swirl vortex is being squashed lengthwise, while the tumble vortex is not having its length changed (we will consider the effect of squish separately in a moment - for now, assume there is no squish). However, the situation in the engine cylinder is complicated by the fact that the density is changing, as well as the length of the vortex. We need a more general statement of conservation of angular momentum with density change. Deriving this here would take us far from our subject; the interested reader can consult a good fluid mechanics text, such as [12]. What we need is called Cauchy’s equation, which states (in a simplified form) that, in an inviscid fluid, if a vortex of initial vorticity \( \omega_0 \), initial density \( \rho_0 \) and initial length \( \ell_0 \) is stretched to a length \( \ell_f \), and changed to a density \( \rho_f \), then the final value of the vorticity \( \omega_f \) is given by

\[
\frac{\rho_0 \omega_f}{\rho_f \omega_0} = \frac{\ell_f}{\ell_0}
\]

The same equation can also be obtained by starting with the integral form of angular momentum conservation, and assuming solid-body rotation. Equation 5.28 can now be applied in the engine cylinder.

Effect on Swirl and Tumble

First, consider swirl. If we start at the beginning of the compression stroke, \( s \) is the stroke plus the clearance height, \( S + h \); at TC, \( \ell_f \) is simply \( h \). The ratio
where \( r \) is the compression ratio. The density ratio, on the other hand is

\[
\frac{\rho_f}{\rho_0} = r
\]

so that the ratio of the final to the initial vorticity is

\[
\frac{\omega_f}{\omega_0} = r \cdot \frac{1}{r} = 1
\]

Hence, there is no change in the value of the swirl vorticity during compression.

Tumble, on the other hand, is another story. Here, the axis of the vortex is transverse to the cylinder, so that there is no change in the length of the vortex during the compression (again, assume no squish). Thus,

\[
\frac{\omega_f}{\omega_0} = \frac{\rho_f}{\rho_0} = r
\]

so that the effect of the density change is to spin up the vortex quite strongly. The tumble vortex will break up before the piston reaches TC. Also, in both cases, the vortices are losing energy to the cylinder walls during the stroke, and we have not taken this effect into consideration. Nevertheless, it is clear that the tumble vortex should contribute a considerably higher turbulence level at ignition, if this is what is desired.

We can make a crude estimate of the effect of friction on a swirl vortex. The wall shear stress is approximately

\[
\tau_w = \rho u_r u_r = \rho \frac{u_r^2}{v_\theta^2} \approx \rho \frac{v_\theta^2}{900}
\]

where \( u_r \) is the friction velocity (defined by the first part of Equation 5.33), and we have made the crudest possible approximation: \( u_r/v_\theta = 1/30 \), which is approximately valid over a very wide range of Reynolds numbers (see [96]). We are, after all, only trying to find out if this effect is important or not. Using the fact that the rate of change of the angular momentum is equal to the applied torque, and supposing that the piston is stationary (another crude assumption which will greatly simplify the problem), we can obtain the equation

\[
\frac{dv_\theta}{dt} = -\frac{v_\theta^2}{900b}
\]

This may be immediately integrated to give

\[
\frac{v_\theta}{v_\theta^0} = \frac{1}{1 + \frac{\rho v_\theta^2}{900b}}
\]

where \( v_\theta^0 \) is the initial value of \( v_\theta \). If we take \( S=b \), and assume that \( v_\theta^0 = 3\bar{V}_p \) (corresponding to \( R_\infty = 2 \), a reasonable value, we obtain

\[
\frac{1}{S} \bar{V}_p = 37.5
\]

as the time required for the initial tangential velocity to drop by one-half. That is, it will take some 38 strokes for this to happen. Despite all the approximations, it is clear that this is not an important effect. There were more approximations that we did not mention - we neglected the torque on the ends of the mass of gas. However, it is obvious that adding these will not make enough difference to make this important.

Tumble is more difficult to treat, because the losses are not so simple to parameterize. We have a cylindrical vortex with its axis transverse to the cylinder, and as the piston
comes up, and the vortex is compressed between the piston crown and the cylinder head, the geometry is difficult to model in any simple way. The only thing that is clear is that the losses are considerably greater than they are for swirl. This is clear, for example, in [80], where the initial swirl ratio is 6, and the tumble ratio is 2. At the end of the compression stroke, the swirl ratio should still be 6, but the tumble ratio should have gone up to 16. However, from the observed tilt of the vortex axis, the tumble ratio is evidently still of order 2. The simplest way to view this, is that the losses are so great that all the additional energy (put in by the compression) is converted into turbulence.

If there is squish we can take that into account using the same equation. Consider the effect on swirl. We can do the calculation in two steps. First, consider a simple compression without squish. The combustion chamber is cylindrical and the same diameter as the cylinder. The piston will stop with a relatively large clearance volume (a low compression ratio) to leave room for the squish. According to our equation (above), Equation 5.31, there will be no change in the vorticity. Now bring in the sides of the combustion chamber to produce the squish. During this phase there will be no change in the length of the vortex. If, for example, the radius is reduced to one-half its value, the area is reduced to one-quarter of its initial value, so the density increases by a factor of four, and hence the vorticity is up by a factor of four. I have sketched this in Figure 5.9. We can apply the compression and the squish sequentially, because Equation 5.28 relates only the initial and final states. Recall that squish is negligible except in bowl-in-piston or bowl-in-head engines.

When tumble and swirl are present simultaneously, Equations 5.30 and 5.31 are simultaneously valid.

![Figure 5.9. Cartoon of a swirl vortex being first compressed and then squished.](image)

**Effect on Turbulence**

A swirling flow has interesting effects on turbulence. The radial distribution of angular momentum can act dynamically like a stably or unstably stratified temperature distribution in a gravitational field (see [96]).

We will have to stop for a moment and talk a little about turbulence dynamics. See Equation 5.21 and the discussion immediately following it. We need to concern ourselves only with the production. In a mean flow on circular streamlines, with tangential velocity \( U \), the production term is

\[
P = -\overline{U \nu} r \frac{\partial}{\partial r} \left( \frac{U}{r} \right)
\]  

(5.37)

where \( \overline{U \nu} \) is called the Reynolds stress. It is the mean value of the product of the fluctuating turbulent velocities in the streamwise and radial directions. This product is not
zero, because the velocities are correlated - that is, when one goes up, the other tends to go up (or perhaps down) also. We will get back to that in a moment.

The term (y) corresponds to the velocity gradient in a parallel flow. If the mean flow is a solid body rotation, so that the speed \( U \) is proportional to the radius, then this vanishes. There is no shear in a solid body rotation.

In a turbulent flow, lumps of fluid are continually changing position. Although they are interacting with their surroundings and interchanging momentum with the fluid with which they are in contact, they are trying to conserve angular momentum also, just as a lump of fluid in a parallel flow is trying to conserve linear momentum, while interacting with its surroundings. We can make a very crude estimate of \( \overline{u \nu} \) if we suppose that a lump having the local tangential velocity of the swirling flow is displaced radially outward (a positive fluctuation \( v \)) to a new, larger radius, and approximately conserves its angular momentum during this displacement. Conserving angular momentum would mean that its velocity would be \( \propto r^2 \), so its swirl speed will drop as it moves to its new radius. A velocity field in which \( U \propto r^2 \) is called a free vortex; this has the same angular momentum at all values of \( r \). When the displaced lump arrives at its new (larger) radius, if the lump finds its swirl speed to be slower than the mean swirl speed of the surroundings at the new radius, it produces a negative velocity fluctuation \( u \); this gives a value of the product \( \overline{u \nu} < 0 \). On the other hand, if the displaced lump finds itself moving faster than the local mean swirl speed, it will have a positive fluctuation \( u > 0 \). Hence, we expect \( \overline{u \nu} < 0 \) if the mean swirl speed has a radial slope less negative than that of a free vortex, and \( \overline{u \nu} > 0 \) if the mean swirl speed has a radial slope more negative than that of a free vortex.

Consider radial profiles of mean swirl speed that lie between a free vortex and solid body rotation (that is, \( U \propto r \)). This will cover any profile that we will find in a swirl vortex in an engine cylinder. Then we expect that \( \overline{u \nu} < 0 \), and the mean shear \( r \frac{\partial}{\partial r} \left( \frac{U}{r} \right) < 0 \). Hence, we expect the production to be negative - that is, it will take energy out of the turbulence and put it back into the mean flow. It will act to suppress the turbulence. It will be a stabilizing influence, like a stable temperature distribution.

![Graph showing the distribution of tangential speed relative to the vortex center in the swirl of Figure 5.6.](image)

In Figure 5.10 I have plotted the distribution of tangential speed in the vortex of Figure 5.6.
It is clear that the swirl vortex is almost in solid body rotation out to a radius of approximately 28 mm, and that the tangential velocity has an essentially constant value after that. The solid body rotation is due to a high turbulence level. Intense turbulence is like a very viscous fluid - it will actively transport momentum in such a way as to try to bring the mean profile to solid body rotation (a state of zero shear). We expect that there will be very little production out to a radius of 28 mm, and strong negative production beyond that point.

We can make a crude estimate of the strength of this negative production. We have determined that $\overline{uv} < 0$; $u$ and $v$ should be well correlated, if the fluctuations in $u$ are essentially caused by the fluctuations in $v$ combined with the conservation of angular momentum. Hence, we should expect $\overline{uv} = -u_0^2$, where $u_0$ is the r.m.s. value of the turbulent velocity fluctuation. Let us take $U \propto r$ out to a certain radius, say near 28 mm and constant = $U_0$ thereafter. Then the production will be equal to (in the region where it is negative and non-zero, for $r > 28$ mm),

$$P = -\overline{uv} r \frac{\partial}{\partial r} \left( \frac{U}{r} \right) \approx -u_0^2 \frac{U_0}{r} \frac{1}{3}$$  \hspace{1cm} (5.38)

We should compare this with the dissipation, which is given by $D = \epsilon = u_0^2/\nu$; let us take $\epsilon = b/6$ and evaluate the ratio $P/D$ at $r = b/2$:

$$P/D \approx \frac{u_0^2 U_0 \epsilon}{r \nu^2} = \frac{U_0}{u} \frac{1}{3}$$  \hspace{1cm} (5.39)

Looking at Figure 5.10, we see that $U_0 = 17$ m/s. Figure 5.11, which we will describe in a moment, gives the peak value of the measured turbulent fluctuating velocity in this flow at $u = 3.46$ m/s. As a result, we have $-U_0/3u = 17/10.4 = 1.6$. This estimate is extremely crude, but it suggests that the suppressive effect of this negative production term may be important. In this region we have, according to our estimate, destructive forces (negative production plus dissipation) = $-2.6\epsilon$. This means (approximately) that the fluctuating velocity in the peripheral region will decay to a value $1/2.6$ of the fluctuating velocity in the core in the same time, or that the fluctuating kinetic energy in the peripheral region will be about 15% of the core kinetic energy after the same decay period. This is numerically approximately what Figure 5.11 shows.

Let us look at Figure 5.11, reproduced from [80], which presents the measured and calculated turbulence distributions corresponding to Figure 5.6. Look first at the measured
distribution. In the center of the cylinder, the turbulence level is roughly 3.4 m/s, or about the mean piston speed. The turbulence level is thus about twice what can be achieved without swirl and tumble. It is also evident that, from a radius of about 26 mm out to the cylinder wall (at least to the resolution of the measurements - a word about that in a moment) there is very little turbulence, in the neighborhood of 1 m²/s². This is exactly the region in which we have estimated that there is a negative production roughly 1.6 times the dissipation. Note: if there is no turbulence, there is no negative production. The negative production we have calculated was present when there was turbulence, and it is what killed the turbulence, but now that the turbulence is dead, the negative production has died also. In Figure 5.11 we are looking at the flow just before TC, but the value of negative production we have calculated corresponds to the conditions during the compression stroke.

The calculated distribution does not reproduce this. The turbulence falls off in a sort of Gaussian shape close to the cylinder wall, but does not fall to zero.

This is because the turbulence model used cannot reproduce the behavior of the turbulence in a swirling flow. Developing a turbulence model that is capable of reproducing such behavior is on the frontiers of research at the moment (see, for example, [55, 83, 84]; these references refer to rotating flows, but rotating and swirling flows are closely related).

Very close to the wall there is a turbulent boundary layer. This layer is not resolved by the PIV technique. The cylinder has a bore of 92 mm, which gives a radius of 46 mm. The PIV data are limited to a circle of radius roughly 35 mm, as can be seen in Figure 5.11, so there is a layer next to the wall of thickness roughly 11 mm that is not observed. In this region, there is a relatively thin turbulent boundary layer trying to propagate against the stabilizing negative production at its edge. This boundary layer is unstable, and is dominated by shear, unlike the flow just above it, which is dominated by rotation. Boundary layers like this in the atmosphere, propagating upward against buoyant stably stratified fluid, are well-understood, but our boundary layer in this rotating flow is not, and we cannot predict the thickness of this boundary layer, other than to say that it will be thinner than a similar boundary layer in a non-rotating flow. The turbulence in the boundary layer will be relatively weak, approximately \( \nu/30 \), which makes it about one-tenth of the intensity of the turbulence in the center of the cylinder. It will also be of relatively small scale, roughly 1/3 the thickness of the boundary layer, which is a small fraction of the bore. The effective diffusivity in the boundary layer, \( \nu \zeta \), will therefore be roughly 2% (or much less) of the diffusivity in the center of the bore.

In this flow of [80] the initial tumble ratio was 2, while the swirl ratio was 6. As we have seen, the swirl ratio after compression should still be 6, but the tumble ratio should be 16 (\( r = 8 \)). However, as we noted above, the tumble ratio stays roughly constant, indicating that all the additional work done on the tumble vortex by the compression has been transformed to turbulence. However, it is evident from Figure 5.11 that the negative production in the outer 1/3 of the radius has managed to kill the turbulence put in by the breaking up tumble as well.

In several recent works [9], [44], [57], detailed budgets of angular momentum, mean and turbulent kinetic energy have been calculated in the engine cylinder. These suffer from being modeled turbulence, and fairly simple models. However, this would not be a problem - our analytical models are also quite simplistic. Unfortunately, the budgets presented are global budgets, integrated over the cylinder volume, so that no information is available on local differences in production.

It is clear that there is a lot going on here that would repay closer attention.