2. LITERATURE REVIEW

The problems associated with simulation of spark ignition engines have been investigated by many researchers. A great deal of analytical and experimental studies of 4-stroke engines, and highly refined models of various physical processes involved are available. Less attention has been paid to its counter-part, the two stroke engine, mainly due to its high specific fuel consumption. However, the past 15 years have witnessed a renewed interest in two stroke engines by both researchers and development communities. Yet, the literature available on various aspects of the two-stroke engine and its simulation is very little.

The research work presently available for the two-stroke spark ignition engine can be broadly divided into two parts; one is related to the open period, and the other is related to the closed period. The open period analysis relates to the gas exchange process while the closed period analysis is concerned with compression, combustion and expansion processes. A brief survey on the following aspects is given in the present chapter:

1. Wave action in inlet and exhaust Systems
2. Scavenging and charging the cylinder
3. Combustion in the cylinder
4. Full Engine simulation

2.1 Wave Action in Inlet and Exhaust Systems

Wave action phenomenon has been investigated extensively. Though the real gas flow is always three dimensional, the flow has been mostly assumed one-dimensional by many researchers as this assumption reduces the computing time drastically. The work reviewed in the present study confines to one-dimensional, unsteady, compressible flow in the engine piping system. The equations which describe such a flow are quasi-linear hyperbolic partial differential equations.

Early studies on the wave action analysis in inlet and exhaust systems were based on the small wave theory. In small wave theory, it is assumed that the flow is steady with small perturbations superimposed. This makes the flow equations
linearized and the complete solution of the problem is obtained by superimposing solutions of number of simple cases. Thus the actual pressure variation is expressed as a Fourier series and the partial solutions for each term are superimposed. Much of the development of this theory is due to Rayleigh (1877). An alternative method of solving the linearized equations in which the waves are plotted graphically, was found more convenient for engine piping system and was used by List and Reyl (1949) for exhaust systems of single and multi-cylinder engines. However, small wave theory was found to be inadequate for problems associated with engines in which the wave amplitudes and flow velocities are large.

Finite pressure wave theory was developed for solution of practical problems of engine exhaust systems by Mucklow and Banister (1948). According to this theory, waves travelling in either directions are considered as if they were simple waves. As the basic wave equation is not linear, the two sets of waves are superimposed according to non-linear laws. This method accounted for temperature discontinuities and pipe friction. This theory was applied to a naturally aspirated two stroke engine by Wallace and Nassif (1954) and later extended to pipes in which the area changes gradually by Wallace and Boxer (1956). Blair (1991) has carried out an open cycle simulation of a two stroke engine and used finite pressure wave theory for the analysis of wave action in engine ducts. Various boundaries have been treated as described by Wallace and Nassif (1954). However, no comparison has been made between the measured and the calculated results.

At the same time as the finite wave theory was being developed, the method of characteristics was being applied to exhaust system problems. Riemann (1885) was first to introduce the method of characteristics for solving one dimensional, unsteady, compressible flow problems. In this method, the hyperbolic equation is reduced to a set of simultaneous ordinary differential equations which define the characteristic curves and the solution can be obtained by integrating the resulting equations with appropriate boundary conditions either numerically or graphically.

Early studies were confined to graphical method of solution of the characteristic equations. De Haller (1945) was the first to apply the graphical method for the study of flow in exhaust systems of i.c. engines. Jenny (1950) extended this method to
include the effects of friction, entropy gradients and gradual change in pipe cross-sectional area. The method developed formed the basis of all subsequent studies. Benson and Woods (1960) successfully applied the method to study the flow in a single cylinder two stroke engine model under cold conditions. Wallace and Boxer (1956) reported wave action analysis in the exhaust system using finite wave theory, graphical method of characteristics and an analytical treatment of the basic equations. They concluded that the graphical method of characteristics gives better agreement between calculations and measurements carried out on a cold wave generator. However, the basic draw back of the graphical method of characteristics is the excessive time required for the solution though it is more accurate than the numerical solution technique.

Shapiro (1953) and Rudinger (1955) proposed numerical techniques for solving unsteady flow problems using method of characteristics. Benson et al (1964) developed a numerical method based on the method of characteristics to study flow in engine systems. Different computer programs were developed for the analysis of flow in a single and multi-cylinder engines exhaust systems. Benson et al (1970) further developed the method to include the effects of variable gas composition, variable entropy and specific heats. The accuracy of the prediction procedure was validated by comparing the results of calculations with the measurements obtained on pulse generators under cold conditions for different pipe configurations and various boundary conditions such as branch, sudden expansion and contraction and gradual area change. The studies were confined to using measured pipe pressure near the cylinder for the analysis and gas exchange in the cylinder was not considered.

Woods and Khan (1968) reported a study on gas exchange in the cylinder along with the wave action in the exhaust pipe system. Measured release cylinder pressure and temperature were used to obtain pressure variation in the cylinder and at the required positions in the exhaust pipe. The comparison of the predicted results with the measured pressure pulses using a pulse generator simulating a multi-cylinder engine discharging in a straight pipe under cold conditions gave good agreement.

Blair and Johnston (1968) examined the wave action characteristics of two different exhaust systems of a naturally aspirated two stroke engine. The exhaust
systems consisted of combinations of straight pipes and convergent/divergent parts. They reported the existence of a numerical instability at the junction of the straight pipe with diffuser. This instability was due to very low pressure and supersonic particle velocities which occurred at the diffuser entrance. To overcome this instability, they used the shock wave relations (Rankine-Hugoniot equations) whenever the Mach number exceeded unity. Blair and Johnston (1970) further reported a study on an expansion chamber for a small two stroke engine. They developed a simplified approach based on the combined technique of using experimentally determined cylinder boundary conditions obtained on mechanical simulator. The application of this technique on expansion chambers of different sizes did not show good agreement with the experimentally measured pressure diagrams.

Blair and Cahoon (1972) studied the gas exchange process in a two stroke engine with two expansion chambers of different sizes. The flow in inlet, transfer and exhaust systems was considered to be one-dimensional and homentropic with constant specific heats. The numerical method developed by Benson et al (1964) was used for the analysis. The cylinder boundary conditions were solved by expressing them independently as a function of port/pipe area ratio. The pressures in crankcase, inlet, transfer and exhaust pipes were calculated and compared with the experimental measurements and reasonable agreement was obtained. Blair and Ashe (1976) applied the same approach as that of Blair and Cahoon (1972) to analyse the gas exchange process in a high speed two stroke engine. They treated separately the state conditions of exhaust products and fresh charge and the scavenging process was simulated assuming either the perfect displacement or the perfect mixing models. They used experimentally determined release pressure and temperature for the calculations relating to the gas exchange process. Comparisons of the experimental and theoretical pressures at different locations of the exhaust and inlet systems showed a better agreement than that reported in the previous work of Blair and Cahoon (1972).

Zucrow and Hoffman (1977) developed an algorithm which expressed the characteristics equations directly in terms of pressure, velocity and density and solved few boundaries in terms of these variables for the case of homentropic flow. However, this algorithm has not been applied for engine non-homentropic flow and engine boundary conditions so far.
Payri et al (1986) noticed some instabilities when the method of characteristics proposed by Benson et al (1964) was used for calculating the pressures in the inlet and the exhaust systems of engines. This discrepancy was attributed to the linear interpolation used by Benson et al (1964) of the Riemann variables and the non-dimensional speed of sound. Payri et al (1986) developed a modified method in which pressure and volumetric flow rate were interpolated linearly while the fluid velocity and the speed of sound were calculated as a function of pressure and entropy level. Furthermore, calculation of entropy level at the pipe ends was modified by adding a duplicated pathline with the entropy of the previous time step when the fluid motion passes from being null or going out of the pipe to entering into the pipe. For junctions between two pipes, the pathlines that pass from one pipe are taken as part of the pathlines of the next pipe so that informations about fluid state are transmitted rather than damped away. Payri et al (1986) validated their algorithm only for a four stroke engine with a straight exhaust pipe and reported that these modification resulted into removal of the instabilities.

Finite difference methods for solving the hyperbolic partial differential equations are known since the work of Courant et al (1928), but were not used frequently for engine applications, while method of characteristics was widely in use. The development of modern computers increased the interest in finite difference techniques. Several methods have been suggested, by Von Neumann and Richtmeyer (1950), Courant et al (1952), Lax (1954) and Lax-Wendroff (1960). Lax-Wendroff two-step method proposed by Richtmeyer (1967) is an effective algorithm for the solution of compressible flow problems. The finite difference solution techniques facilitate the accurate prediction of rapid changes in the flow which makes these methods attractive for applying to engine exhaust systems. However, finite difference techniques suffer from the defect that a different method is required for the solution at the boundaries, usually the method of characteristics. Further, it was observed that finite difference methods produce solutions with non-physical overshoots near the discontinuities. Many proposals have been presented to overcome these overshoots as reported by Bultay and Niessner (1984).

Chen and Wallace (1992) reviewed different finite difference methods which are used for modelling one-dimensional unsteady compressible gas flow in pipes and
their applicability to i.e. engine problems. A comprehensive prediction procedure was
developed by the above authors which has the flexibility to generate wide range of
engine systems and to select the suitable models in order to predict engine performance
characteristics and in-cylinder flow phenomena. Validation of the performance characteristics
of a four-cylinder petrol engine as well as of the predicted pressure in the inlet pipe
of a single cylinder motored engine were carried out. However, no results were reported
for the pressure diagrams in the exhaust systems of the two engines.

The above survey of the existing literature reveals that for the solution of one-
dimensional, unsteady, compressible flow in inlet and exhaust systems of i.e. engines,
the method of characteristics is used extensively. And there are three algorithms for
the solution based on the method of characteristics, these are:

1. Benson algorithm: This algorithm was formulated by Benson et al (1964)
   and solves the characteristics equations in terms of Riemann variables (\(\lambda\) and
   \(\beta\)) and the entropy level \(A_a\). Details of this algorithm are given in Appendix-
   A.

2. The Modified algorithm: This algorithm is a modification to Benson algorithm
   by Payri et al (1986) where the characteristic equations are expressed in terms
   of Riemann variables and entropy level as it is the case with Benson algorithm.
   However, during the numerical integration procedure, the Riemann
   variables are transformed into pressure \(p\) and velocity \(u\) and the integration
   is carried out for these variables along with entropy level \(A_a\). Details
   of this algorithm are available in Appendix-A.

3. Zucrow–Hoffman algorithm: This algorithm is proposed by Zucrow and
   Hoffman (1977) and solves the characteristics equations in terms of pressure
   \(p\), velocity \(u\) and density \(\rho\). Details of this algorithm are given in Chapter 4.

The survey shows that Benson algorithm results into numerical instabilities when
employed for solving the flow equations in exhaust systems having divergent parts.
No reference was made in literature for employing the Modified algorithm to this
case. In the present work, we found that this algorithm also results into the same sort
of instability when employed for a high speed two stroke engine with expansion
chamber as will be discussed in more detail in Chapter 5. The survey also shows that
Zucrow-Hoffman algorithm has not been employed so far for complete analysis of engine flow. Therefore, the present work is majorly concerned with employing Zucrow-Hoffman algorithm for the solution of engine flow problems, particularly for exhaust systems having divergent parts.

2.2 Scavenging and Charging the Cylinder

The performance and emissions of internal combustion engines are dependent mainly on the thermodynamic properties of the mixture trapped inside the cylinder at the commencement of compression. In a two stroke engine, these properties are closely related to the efficiency of the charging process. In ideal scavenging, the fresh charge acts as a piston in pushing the burned gases out of the cylinder without actually mixing with them, this represents the case of perfect displacement. If the entering fresh mixture mixes instantaneously and uniformly with the cylinder contents, then this is called perfect mixing. Perfect mixing and perfect displacement are two limiting ideal conditions which are not possible in a real engine. In real engine scavenging, mixing occurs as the fresh charge displaces the burned gases and some of the fresh charge may be expelled, i.e., short circuited to the exhaust port. Various scavenging models have been proposed to represent the gas exchange process within the cylinder, some are discussed below:

Maekawa (1957) proposed an isothermal two zone model for simulation of the scavenging process in a two stroke engine. It was assumed that the cylinder volume is subdivided into two zones: a mixing zone and a fresh charge zone. The stream of the entering fresh charge splits into three sub-streams, one part is short circuited to the exhaust port, another part mixes in the mixing zone and the remaining forms a pure fresh charge zone. At the same time, gas leaves the cylinder from the mixing zone first. No mass or heat transfer was considered between the two zones and the scavenging process was supposed to occur at a constant cylinder volume, pressure and temperature. This model did not include the displacement scavenging phase.

Benson and Brandham (1969) reported a method for analysing quantitatively a mixing displacement scavenge model for the gas exchange process in a two stroke engine. It was assumed that the cylinder is sub-divided into two zones; a mixing zone adjacent to the scavenge port and a burned gas zone adjacent to the exhaust port.
No short circuiting of fresh charge to the exhaust port and no mass or heat transfer were considered between the two zones. The model was incorporated in the thermodynamic cycle calculations using wave action and gas exchange relations and the results of this model were compared with the results of mixing scavenging. It was concluded that at high charging efficiencies there is an improvement in engine performance compared with using the mixing process alone.

Streit and Borman (1971) applied the isothermal two zone model of Benson and Brandham (1969) to a special design of opposed piston engine. During the scavenging period, the cylinder is thought to consist of two thermodynamic subsystems. System (I) is always connected to the scavenge port and system (II) is always connected to the exhaust port. It was also assumed that in each system a uniform mixture and temperature exist and that both systems are at the same pressure. Heat transfer to or from the walls in each system was taken into consideration but no heat transfer between the two systems was assumed. The mass exchange from system (I) into system (II) and vice-versa was specified separately as function of inlet port, mass flow rate and delivery ratio.

Dedeoglu (1971) performed experimental study of the gas exchange process in two stroke spark ignition (s.i.) engines of different port designs. The experiments indicated that the scavenging process in cross scavenged engines occurs in two principal phases. In the first phase the air first penetrate the cylinder gases and displaces them towards the cylinder cover and then towards the exhaust ports. During this phase there is some jet mixing at the boundaries of the air jets, and at the end of the phase, some of the incoming air passes straight through the exhaust ports causing short circuiting. In the second phase the scavenge air mingles with cylinder contents and a mixture of scavenge air and gas leaves the cylinder through the exhaust ports. The proportion of air and products of combustion in the exhaust ports varies with time. Further, the scavenging phases depend on the port design.

Benson (1977) proposed a gas dynamic model to represent the gas exchange process in a cross or loop scavenged engine based on the experiments of Dedeoglu (1971). The scavenging process is considered to be in three phases: a displacement phase, a short circuiting phase and a mixing phase. The cylinder was subdivided into
three zones: an air zone, a gas zone and a mixing zone. Two models were proposed for calculations of the proportion of air and gas in the mixing zone. The first model assumes a constant gas entrainment fraction by the incoming air jet. The second model assumes a linear drop in the volume ratio of the gas zone. Calculations presented for a crankcase scavenged engine have shown that a good representation may be made of the scavenging process by the new model, although the constant entrainment ratio mixing model is less sensitive to input parameters than the second model.

Baudequin and Rochelle (1980) proposed a three parameter model which is a synthesis of the two zone model of Benson and Brandham (1969) and two zone model of Maekawa (1957). The model consists of a displacement scavenging period followed by a period which includes the cylinder short circuiting of part of the delivered fresh gases, the existence of a dead zone of burned gases in the cylinder and the perfect mixing scavenging of the rest of the volume. The model was found to follow closely the experimental measurements and to permit the correlation of the fundamental parameters of the scavenging curves with the variation of cylinder geometry.

Sato and Kido (1983) applied a version of Maekawa model (1957) to a small crank-case scavenged engine. They considered that the cylinder volume was subdivided into two zones; a mixing zone and a stratified zone. The stream of the fresh charge entering in the cylinder splits into two sub-streams; one enters the mixing zone and the other enters the stratified zone. The exhaust gas consists of two streams; one from the mixing zone and the other from the stratified zone. Heat transfer between the two zones was considered and calculated using empirical correlation. It was recommended that 85% of the charge enters the mixing zone and the remaining, the stratified zone and the mass leaving was completely considered from the mixing zone.

Sher (1985) proposed a semi-empirical model to simulate the scavenging process in cross, loop or uniflow scavenged engines. The model was based on the assumption that the time variation of the mass fraction of fresh charge content in the gas passing through the exhaust port, \( \beta \), exhibits a sigmoid curve (s-type). It was assumed that the scavenging process occurs at a constant cylinder volume and pressure. \( \beta \) was defined as:

\[ \beta = \frac{C}{1 + e^{-k(t-t_0)}} \]
\[ \beta = 1 - \exp \left[ -c \lambda \left( \frac{\alpha - \alpha_{so}}{\alpha_{sc} - \alpha_{so}} \right)^b \right] \]

where \( b \) and \( c \) are the form and shape factors, respectively. An appropriate selection of \( b \) and \( c \) can provide any curve between perfect mixing and perfect displacement processes.

Chen and Wallace (1987a) suggested a three zone model wherein cylinder pressure and volume during the scavenging process were assumed to be constant. This model is supposed to give the thermodynamic description of any possible scavenging method, provided the coefficients of intake and discharge proportions are known. Using this model the scavenging formulations of number of well known workers, for example Maekawa (1957), Benson et al (1969) and Baudequin and Rochelle (1980) were derived. The model could precisely represent all characteristics of experimental charging trends. In another work Chen and Wallace (1987b) have proposed a phenomenological model for uniflow scavenging process based on the laws of conservation of mass, momentum and energy for compressible flow taking into consideration the unsteady jet theory to give an approximate description of velocity and concentration fields. The model is a multizone model with spatial and temporal history of mass, local temperature and concentration within every zone. However, the pressure was assumed to be constant throughout the cylinder. The model has two versions one is based on mass entrainment rate in which the jet is considered as an entity, and the other is based on eddy diffusivity theory in which the jet is divided into several strips. Good agreement between experimental and computational results were found for both the versions, and the model predicted well the effect of swirl, which characterizes the uniflow scavenging, on the scavenging effectiveness.

2.3 Combustion in the Cylinder

It is well established that the burning mechanism in s.i. engine is a turbulent combustion process. The mechanism of flame propagation is governed by the flow motion in engine cylinder. However, the flow in engine is turbulent and involves a complicated combination of turbulent shear layers, recirculating regions and boundary layers. Further, the flow is also unsteady and may exhibit substantial cycle to cycle
fluctuations and both large-scale and small-scale turbulent motions are important as they control the mechanism of flame propagation. However, the existence of identifiable structures in the turbulent flow has prompted many researchers to propose turbulent combustion models based on the characteristics of these structures. A critical review of some theories of turbulence and their compatibility with different models of turbulent flame propagation has been made by Andrew et al (1975). The application of such models to spark ignition engines has been reviewed by Tabaczynski (1976). A brief review of some of these models is given below.

2.3.1 Modelling of the Combustion Process

The first structural turbulence model was proposed by Tennekes (1968). He postulated that vorticity is concentrated in vortex tubes which are characterized by the Kolmogorov scale $\eta$ and that the characteristic spacing of adjacent vortex tubes is the Taylor microscale $\lambda$, as shown in Fig. (2.1). Using Tennekes turbulence structure, Chomiak (1970) suggested a mechanism of turbulent flame propagation based on fast flame propagation along the vortex tube of the Kolmogorov scale $\eta$. The change of density due to combustion causes the vortex to collapse and the effect of this combustion bursting allows the flame to travel along the vortex tubes at a much higher turbulent flame velocity than the laminar flame velocity. The flame propagation proceeds through the space between the vortex tubes characterized by the Taylor microscale $\lambda$ with the slow laminar flame velocity. Daneshyar et al (1987) proved the compatibility of the Tennekes model with established results for locally isotropic turbulence and with experimental measurements of various length scales in turbulent flow field. They also reviewed visual observations of flames and concluded that a turbulent structure exists which is strongly dependent on the turbulence level. At low turbulence, flame exhibits a thin wrinkled flame while at high turbulence a thick combustion zone exists which is a mixture of unburned, burning and fully burned zones. Further, turbulent flame fronts exhibit zones of strong curvature and the typical distance between these regions of high curvature decreases sharply with engine speed.

Fig. (2.2) shows in a simplified way the flame front propagating at turbulent speed $S_\lambda$ followed by a thick zone in which there is a mixture of burned gas and still burning pockets initially of typical size $\lambda$ (Taylor microscale). This model supposed
to depict the implications of Tennekes model, i.e., slow burning on the \( \lambda \) scale and fast burning on the \( \eta \) (Kolmogorov) scale. Experimental as well as theoretical evidence reported by Abraham et al (1985) and Heywood (1989) suggested that engine combustion in the range of low to mid engine speed and low to high engine load occur in the flamelet burning mode. Flamelet combustion corresponds to chemical reaction occurring at fast time scales and short length scales relative to the turbulence time and length scales. In Flamelet regime, turbulent flame zone consists of laminar flamelets separating regions of uniform composition and temperatures; the reactants and the products. The instantaneous behaviour of flamelets is same as those of laminar flames. The local burning rate is determined by the structure of the laminar flamelets, the local curvature and the velocity gradient tangential to the flame front (strain rate).

Turbulence not only act to produce distortions in the flame front and to cause flame stretching but turbulence itself is also generated due to the process of flame propagation. However, it is difficult to explain as to how flame-generated turbulence can influence burning velocity because the very process of its generation follows and does not precede the flame propagation. In s.i. engines the effect of flame on turbulence is still debated and the literature available is contradictory. Witze et al (1984) reported that turbulence is enhanced ahead of the flame front due to compression of the turbulent unburned mixture. This result is consistent with rapid distortion theory of Wong and Hoult (1979) which indicates that the turbulence intensity in certain directions should increase due to rapid compression. On the contrary, Borgnakke and Xiao (1991) stated that compressibility strain rates in engines are not large enough to satisfy rapid distortion theory. Witz et al (1984) reported that compression may either amplify or destroy the turbulence depending on the inlet turbulence intensity. On the other hand, Hall et al (1986) stated that there is little or no increase in turbulence intensity ahead of the flame.

**Correlations of Turbulent Flame Speed**

It is clear from the above that theories of turbulent flame propagation do not permit direct evaluation of the turbulent flame speed \( S_T \), but provide a variety of descriptions of the flame-turbulence interactions that can be used as a basis for empirical relations. It is generally accepted that turbulence increases the ratio of turbulent flame speed \( S_T \) to the laminar flame speed \( S_L \).
Collectively, the correlations proposed for the calculation of turbulent flame speed \( S_T \) may be expressed as follows:

\[
S_T = a S_L + b u'
\]  (2.3.1)

where \( S_L \) is the laminar flame speed

\( u' \) is the absolute turbulence intensity, i.e., the r.m.s. velocity fluctuation about the mean

\( a, b \) are constants.

This correlation implies a linear variation of the flame propagation rate and the turbulence intensity. The first term \((a S_L)\) represents the effect of chemical kinetics which enters via the laminar flame speed, and the second term \((b u')\) gives the effect of turbulence level on the turbulent flame speed. The constant \( a \) and \( b \) were assigned values based on the analysis of data from a variety of engine operating conditions. It was found that the second term is usually more than the first term. That is, the effect of the chemical kinetics on the combustion process is relatively unimportant compared to the effect of turbulence levels prevailing in practice.

There have been other correlations which predicts a departure from linear dependence of \( S_T \) on \( u' \), for example; Hires et al (1978) correlated the turbulent flame speed as

\[
\left( \frac{S_T}{S_L} \right) = C_S \left( \frac{u'}{S_L} \right)^{1/3} \left( \frac{\rho_u}{\rho_i} \right)^{1/3} \left( \frac{u'L}{v} \right)^{1/3}
\]  (2.3.2)

where \( C_S \) is an empirical constant.

Chen et al (1991) proposed a turbulent flame velocity model based on dimensional reasoning of the classical turbulence structure. This model suggests three stages of development of turbulent flame speed depending on the size of the flame with respect to a characteristic turbulence scale called the inertial scale, which is the size of the middle scale eddies. The ratio of turbulent burning speed to the laminar flame speed in these three stages is as follows:

\[
\left( \frac{S_T}{S_L} \right) = 1 \quad \text{For } (R_f/L) \leq r_{i,L}
\]

\[
\left( \frac{S_T}{S_L} \right) = 0.8717 \ Re_l^{3/8} (R_f/L) \quad \text{For } r_{i,L} \leq (R_f/L) \leq 1
\]
\( \frac{(S_{T,S}}{S_L}) = 0.8717 \, \text{Re}_L^{3/8} \quad \text{For} \quad 1 \leq (R_f/L) \quad (2.3.3) \)

\( r_{t,L} = (1.1472/\text{Re}_L^{3/8}) \)

\( \text{Re}_L = (u'L/v) \)

where \( R_f \) is flame radius

\( L \) is integral length scale

\( r_{t,L} \) is critical ratio of flame size to integral scale (less than this value the flame is laminar)

\( \text{Re}_L \) is Reynolds number based on the integral scale

\( u' \) turbulence intensity

\( v \) kinematic viscosity

\( S_{T,S} \) Saturated turbulent flame velocity for fully developed flame

This model has been employed in the present study for estimation of turbulent flame speed during the combustion process.

**Early Flame Development**

Early flame development is defined as the initial stage of burning from the time of spark breakdown to a noticeable departure of the cylinder pressure from the compression pressure. Comparatively, this combustion phase has a long duration as compared to the small fraction of the charge that is burned. Typically, it takes 20% of the combustion time to burn 0.1 percent of the charge, corresponding to a burned volume of 1 cm³ (Baritaud, 1987). Therefore a small change of initiation duration leads to a large variation of overall combustion duration. This influence is well supported experimentally by Kalghatgi (1985) and Swords et al (1982).

Different opinions exist concerning the speed of flame during the early stages of development of the flame. Chomiak (1979) considered that the flame kernel formation to be consisting of two periods. **The first period** is highly reproducible and it is independent of flow turbulence and mixture composition for mixtures undergoing easy ignition. The length of this reproducible period is 1.5 msec. **The secondary period** is a non-repeatable and the percent coefficient of variation of the kernel size is a linear function of the turbulent velocity fluctuations, and for mixtures undergoing easy ignition it does not depend on mixture composition nor on spark energy or spark gap
width. While in contradictory to Chomiak (1979), measurements in engine by Namazian et al (1980), Smith (1982), ZurLoye et al (1985) and Baritaud (1987) have shown that flame kernels are influenced by turbulence as early as 0.1 msec after spark onset. Theoretical analysis of kernel formation carried out by Bradley and Lung (1987) has shown no turbulence interaction with flame formation within the first 0.2 msec.

Based on comparisons of "flame shadow" radii measured optically and burned gas radii determined from pressure measurements, Keck et al (1987), Taglian and Heywood (1986) and Psichinger and Heywood (1988) have shown a laminar like burning process immediately following the spark discharge and an expansion speed of the flame kernel which increases approximately linearly with the burned gas radius. They have reported that the most important parameter controlling the initial flame growth are the laminar flame speed at the spark plug and the size of the first eddy burned. They postulated that in the early stages the turbulence has a convective effect of the initial flame and does not immediately affect the laminar structure of the flame. Betev and Karpov (1990) observed a quasi-laminar behaviour in the earliest stages of the flame development; the kernel becomes mildly affected by turbulent structure with a characteristic length scale comparable to the size of the flame. Herweg et al (1988) based on optical measurements, reported that at low turbulent intensities, the expansion velocity of the flame drops within 0.4 msec from the spark induced high values to an initial velocity very close to that of a laminar flame. Therefore the laminar flame chemistry controls the flame kernel formation and increasing the turbulence intensity leads to a steady increase of the initial expanding speed due to wrinkling of the flame area. Daneshyar and Hill (1987) postulated that the effect of small scale structure of turbulence on early flame development is to delay the combustion time randomly. Since the first ignited flame kernel might be located anywhere inside a \( \lambda \)-sized eddy and would burn in a laminar fashion till it reaches the zones of turbulent burning along the vortex filaments of scale \( \eta \) (Kolmogorov scale). While the effect of the large scale turbulence on early flame development as reported by Gatowksi et al (1984) is to randomly blow flame zone towards or a way from the wall so that the degree of partial quenching of the flame zone during its early growth varies coincidently from cycle to cycle. LeCoz, J.F. (1992) based on cyclic-resolved processing of the velocity measured near the spark plug, have reported that the small scale
turbulence affects the combustion rate during the initiation phase only for a fuel/air equivalence ratios higher than 0.8, while for lean mixtures convection of the flame kernel by the large-scale fluid motion is the only cause for cyclic variation of the initial flame.

Bianco et al. (1991) have characterized the early flame development by an expansion speed which describes its growth rate and a convection velocity which describes its overall movement. The value of the expansion speed is 2 to 3 times larger than the laminar expansion speed which implies a significant enhancement of the burn rate by turbulence. Further, they have observed a correlation between 0-2 percent mass burn durations and the expansion velocity (i.e. early flame development) and also a correlation between the direction of the initial convection velocity and the 10-90 percent burn duration.

However, for modelling the early flame development the recent approach is to bridge the gap between the general combustion theory and the small flame kernel development by incorporating the recent results of combustion research into the model of early flame kernel development, such as flamelet concept, concepts of flame quenching by turbulence, early flame extinction due to heat losses into spark electrodes and flame/vortex interactions. Examples of these models, are reported by Herweg and Maly (1992) and Dulgar et al. (1994).

It is obvious from the above survey that much of the endeavour of the researchers is focused on studying the early flame development in premixed flames. As early flame development is the determining factor for the later propagation of the flame and consequently on the pressure development in the cylinder during the combustion process. However, detailed modelling of early flame development is not carried out in the present work due to lack of informations about the ignition system for the engine simulated.

2.3.2 Combustion Models for S.I. Engines

Based on the above review of the work concerning the thermodynamic modelling technique of the combustion process, it may be observed that turbulent flame propagation in engines comprises both wrinkled flame regimes (with or without pockets) and the
thickened wrinkled flame regime. In these different regimes turbulent propagation velocity shows different behaviour. Therefore, the difficulty in modelling the combustion process in s.i. engines arises because of a need for a model to incorporate more than one regime of turbulent combustion. However, many models are in use for prediction of the burning rate in s.i. engines, starting from simple cosine law to models which embody recent theories of turbulent combustion. The available models may be categorized into the following two distinct groups:

(a) Physical Models: These models incorporate the basic turbulent flow quantities such as Integral length scale \( L \), Taylor Microscale \( \lambda \) and Kolmogorov length scale \( \eta \) and turbulence intensity \( u \), along with the laminar flame speed \( S^l \).

Blizard and Keck (1974) were first to formulate a combustion model for s.i. engines based on physical parameters of turbulence. The flame propagation was treated as a two-step mechanism in which the rate of combustion is governed by the entrainment of unburned gases by the flame due to the fluctuations in the flow field, and the subsequent burn up of these engulfed gases. The rate of entrainment is assumed proportional to the turbulent intensity \( u \) at the time of ignition and is given by:

\[
\frac{dm_{en}}{dt} = \rho_u A_f S_{en} \quad (2.3.4)
\]

where

\( \rho_u \) is the gas density of the unburned gas

\( A_f \) is the spherical flame front area

\( S_{en} \) is the flame entrainment velocity and is equal to the turbulent flame speed \( S_t \)

The entrained mass is assumed to burn in a manner characterized by the following relaxation expression.

\[
\frac{dm_b}{dt} = \frac{(m_{en} - m_b)}{\tau_b} \quad (2.3.5)
\]

\( \tau_b \) is the characteristic burning time and defined as:

\[
\tau_b = \frac{L}{S_l} \quad (2.3.6)
\]

where \( m_b \) is the burned mixture mass

\( m_{en} \) is the mixture mass engulfed into the flame front
\( \lambda \) is the Taylor microscale

\( S_L \) is the laminar flame speed

McCuston et al (1977) observed that the Blizzard and Keck model (1974) did not adequately predict the effect of equivalence ratio and engine speed on combustion. The disagreement was attributed to the assumption of constant entrainment speed and constant characteristic burning time. Based on the turbulent flame propagation model proposed by Chomiak (1970) Tabaczynski et al (1977) postulated that ignition occurs across the highly dissipative regions (the order of Kolmogorov scale, \( \eta \)) and burning on these scales is assumed to be instantaneous. The rate of propagation of the ignition sites \( S_{en} \) (entrainment) is function of both chemistry and turbulence and is given by:

\[
S_{en} = S_T = u' + S_L
\]  

(2.3.7)

The burn up of the charge behind the ignition site propagation front is governed by the spacing of the dissipative regions (Taylor microscale \( \lambda \)) and the laminar burn up across the spacing. The characteristic burning time \( \tau_b \) is given by

\[
\tau_b = \left( \frac{\lambda}{S_L} \right)
\]

(2.3.8)

Tabaczynski et al (1977) model predicted correct trends of ignition delay and combustion duration for different equivalence ratios, exhaust gas recirculation, spark timing and engine speeds.

Hires et al (1978) separated the burning process into a developing turbulent flame which was taken as the process of burn up of a single eddy (ignition delay) and a fully developed turbulent flame in which entrainment of eddies of size \( L \) (the integral scale) were engulfed into the flame front. Two semi-empirical formulae for predicting the ignition delay and the combustion duration were proposed. According to Hires et al (1978) model, the ignition delay is assumed to be the time of burning an \( L \)-sized eddy while, the fully developed flame propagation is supposed to be laminar burning of the spacing of \( \lambda \)-size eddies. These assumptions did not allow a smooth transition from the time of spark to fully developed flame propagation. Therefore, Tabaczynski et al (1980) further modified the model of Tabaczynski et al (1977) by defining the delay period as the time to burn the small eddy of the size of Taylor microscale. This means that the flame is never fully developed but rather a continuously developing process.
Fig. (2.3) depicts Tabaczynski et al (1980) model when a large (L-sized) eddy is entrained within a thick turbulent flame. It contains several λ-sized regions in which the burning rate is slow and occurs at the laminar flame speed $S_L$ and the boundaries of these regions of scale $\eta$ are rapidly inflamed at the turbulent flame speed $S_T$.

Chen and Veshagh (1992) concluded that Tabaczynski et al (1980) model could not fully explain the entrainment history. It led to predicting a slower burning process and a thicker flame than suggested by the Tennekes-Chomiak mechanism as reported by Groff (1987). The discrepancy was attributed to the fact that the history of entrainment as implied by Tennekes-Chomiak model was not taken into consideration. Chen and Veshagh (1992) expressed the ordinary differential equation, Eqn. (2.3.5) proposed by Tabaczynski et al (1980) as a difference equation for the calculation of the burn rate ($\frac{dm_b}{dt}$) as given below:

$$m_b = \sum_{n=1}^{i} m_{en} \Psi^n$$  \hspace{1cm} (2.3.9)

where, $\Psi^n$ is a shape function and defined as:

$$\Psi^n = 0 \quad \text{for} \quad i \Delta t - n \Delta t \leq 0$$

$$\Psi^n = 1 \quad \text{for} \quad i \Delta t - n \Delta t \geq \tau^n$$

$$\Psi^n = \left( \frac{i \Delta t - n \Delta t}{\tau^n} \right) \quad \text{for} \quad 0 \leq i \Delta t - n \Delta t \leq \tau^n$$

where, $\tau^n = (\lambda / S_L)^n$

Chen et al (1992) model has been adopted for modelling the combustion process since it involves the latest modification to the eddy entrainment model which has been developed over years by Tabaczynski and Co-workers and is well known for predicting the correct trends of mass burning rates and combustion durations, see for example Tabaczynski et al (1980).

(b) **Empirical Models**: It may be concluded from the review of above work that, the development of combustion models for spark ignition engines based on fundamental principles is a difficult task. Therefore, researchers such as Rashidi (1981). Keck
(1982), Gatowksi et al (1984) focussed on development of empirical models based on detailed correlations of experimental data. They used pressure measurements and high speed photographs of flame propagation to derive empirical equations for calculations of burning rates and relating the parameters in the burning equation to the engine geometry and operating conditions. Some of the correlations proposed by Keck and Co-workers such as Keck (1982), Beretta et al (1983a) and Keck et al (1987) are given below:

\[
\frac{dm_b}{dt} = \rho_U A_b S_L + \frac{\mu}{\tau_b} \tag{2.3.10}
\]

\[
\frac{d\mu}{dt} = \rho_U A_b U_T \left(1 - e^{-t/\tau_b}\right) - \frac{\mu}{\tau_b} \tag{2.3.11}
\]

\[
\mu = m_e - m_b = \rho_U (V_f - V_b) = \rho_U L_T (A_L - A_f) \tag{2.3.12}
\]

where

- \(m_b\) : is the mass burned
- \(m_e\) : is the mass engulfed by the flame front
- \(\mu\) : is a dummy variable with the dimensions of mass
- \(\tau_b\) : a characteristic burning time = \(L_T / S_L\)
- \(L_T\) : Characteristic length scale = \((V_f - V_b) / (A_f - A_b)\)
- \(U_T\) : a characteristic speed \(\approx\) turbulence intensity \(u'\)
- \(A_f\) : flame front area
- \(A_b\) : equivalent spherical burning area
- \(V_f\) : volume of engulfed mass
- \(V_b\) : volume of burned mass
- \(A_L\) : laminar burning area, the flame would have if it burned at the laminar flame speed and defined as:

\[
A_L = \left(\frac{dm_b/\, dt}{\rho_U \, S_L}\right)
\]

The mass burning rate equations (2.3.10) and (2.3.11) contain three parameters, \(S_L\), \(U_T\) and \(L_T\) which may be obtained from experimental measurements. \(U_T\) may be obtained from engine data where \(U_T\) is correlated to the mean piston speed. \(L_T\) may
be found as a function of density ratio $\rho_u/\rho_{ui}$ during combustion process, where $\rho_u$ is density of unburned mixture and $\rho_{ui}$ density at the time of ignition.

The advantages of the above models are that they can be used both for immediate practical applications in engine design and analysis and to obtain physical insight useful for developing and testing more fundamental models of turbulent combustion. However, the limitation of these models is that it is not possible to carry out investigations on all possible engine geometries and operating variables to check the tentative correlations for $U_T$ and $U_T$.

2.4 Cycle Simulation Models

Literature on the simulation of a small two-stroke crankcase scavenged engine including both the power cycle and the open cycle is limited. Benson et al (1975b) reported a model to simulate the power cycle and the gas exchange process in a crankcase compression two stroke spark ignition engine having straight intake and exhaust pipes. The effect of carburettor in the inlet pipe was included. For the closed cycle a simplified combustion model was used where the turbulent flame speed was assumed to be linearly related to the laminar flame speed and the multiplying factor was called flame factor. For the open cycle analysis, three scavenging models were used, these are mixing, displacement and 50% mixing and 50% displacement. The effect of different scavenging models and different flame speed on the release pressure and temperature as well as on the indicated power was examined. However, no comparison of theoretical and experimental results of the cylinder pressure for the entire cycle (closed and open cycle) or the inlet and exhaust systems was reported.

Blair (1976) examined the theoretical performance characteristics of a two-stroke cycle engine. For the closed period heat release pattern suggested by Lyn (1960) was used instead of a combustion model. The wave action in engine piping system was analysed using Benson algorithm. Besides comparing the measured and calculated cylinder pressure-time diagrams, performance characteristics were also compared. No comparisons of pressure variation in inlet and exhaust systems were reported. However, Blair and Johnston (1970), Blair and Cahoon (1972), Blair and Ashe (1976) carried out only homentropic calculations of the cycle and the cylinder boundary conditions were solved and expressed in tabular form as a function of port/pipe area ratio and
then used for actual calculations. Furthermore, calculations were carried out only for the open cycle while the closed cycle was simulated by defining the release pressure and temperature at exhaust port opening. Although good predictions were obtained, the discontinuity of the simulation of the whole engine cycle reduces the validity of these predictions.

Malik et al (1988) reported a simulation study on a two stroke spark ignition engine coupling the scavenging and combustion processes and neglecting wave action in the intake and exhaust systems except for the transfer pipe. Scavenging process was analysed using a three phase model including displacement, short circuiting and mixing. The flow of the fresh charge in the exhaust during the mixing phase was based on the s-shape model proposed by Sher (1985). Combustion was analysed using Benson et al (1975a) model and turbulent flame speed was calculated as proposed by Hires et al (1978). Cylinder pressure was predicted and compared with experimental results under different operating conditions. It may be noted here that the impact of the wave action phenomenon in inlet and exhaust systems, on the cylinder pressure development during open period as well as during the closed period was neglected which is a major factor in influencing the performance characteristics of the two stroke engine.

The above survey shows that a complete satisfactory simulation model of a two stroke engine including divergent and convergent parts in the exhaust system has not been completed and to achieve this, a successful wave action analysis in engine piping system coupled with realistic in-cylinder phenomena models such as scavenging and combustion models has to be formulated.
Fig. (2.1) Model of Concentrated Vorticity Region as Proposed by Tennekes (Daneshyar and Hill, 1987)

Fig. (2.2) Flame Propagation with Thick Combustion Zone (Daneshyar and Hill, 1987)
Fig. (2.3) Model of Burning, Turbulent, Small-scale Structure (Tabaczynski et al, 1980)