# STRATIFIED-CHARGE TWO-STROKE STEPPED-PISTON ENGINE

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To my beloved late father and mother,

Your love brings me the happy and successful life

To my beloved wife and children,

Your encouragement and patience always give me spirit to achieve success

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# ABSTRACT

A two-stroke stepped-piston prototype engine, in carbureted version was designed and developed. It incorporates a unique three-port transfer system with an accumulator for high induction efficiency, so as to perform very much like a two-stroke engine but equipped with a four-stroke crankcase lubrication system. *GT-Power* software was used in the development stage to predict the engine output. The data predicted was then compared with the experimental results. A computational fluid dynamic software, *COSMOS/Floworks*, was used to develop a computational model to investigate the scavenging and compression processes of the prototype engine.The prototype was subjected to a series of laboratory trials for engine performance and emissions tests. Emission characteristics were established for regulated and unregulated gases. From the engine performance test, maximum pressures attained from *GT-Power* simulation and prototype engine were 54.62 bars at 5000 rpm and 26.12 bars at 4500 rpm respectively. Maximum indicated power produced is 11.25 kW at 8000 rpm and 3.86 kW at 4500 rpm for *GT-Power* simulation and prototype engine respectively. Torque, brake power and brake fuel consumption were also determined. For comparative reason, a *Yamaha 125Z* engine was selected as the cylinder capacity with similar working principle as the prototype engine. Torque produced by *Yamaha 125Z* was highest, followed by *GT-Power* simulation and prototype engine. The average difference of torque between *Yamaha 125Z* and *GT-Power* simulation was about 13.06%. The minimum values of brake specific fuel consumption for *Yamaha 125Z*, *GT-Power* simulation and prototype engine were 280.42 g/kWh at 3500 rpm, 351.08 g/kWh at 5000 rpm and 510 g/kWh at 3500 rpm respectively. The maximum peaks differences were attributed to the differences of combustion chamber design used and assumptions made in *GT-Power*.

# ABSTRAK

Dalam kajian ini sebuah enjin prototaip dua lejang omboh bertangga telah direkabentuk dan dibangunkan dalam versi karburetor. Ia digabungkan dengan sistem tiga liang hantaran dan pengumpul yang unik untuk kecekapan aruhan yang tinggi dan berkelakuan seperti enjin dua lejang tetapi sistem pelinciran bersifat seperti kotak engkol empat lejang. Perisisan *GT-Power* digunakan di peringkat pembangunan untuk meramal keluaran enjin. Data teramal kemudian dibanding dengan keputusan ujikaji. Perisian dinamik bendalir berkomputer, *COSMOS/Floworks*, digunakan untuk membina model komputer untuk mengkaji proses menghapus sisa dan pemampatan enjin prototaip. Kemudian prototaip menjalani satu siri ujian makmal untuk menentukan prestasi dan emisi enjin. Emisi enjin ditentukan untuk gas terkawal dan tidak terkawal. Daripada ujian prestasi enjin, tekanan maksimum simulasi *GT-Power* dan enjin prototaip masing-masing adalah 54.62 bar pada 5000 ppm dan 26.12 bar pada 4500 ppm.. Kuasa tertunjuk maksimum adalah 11.25 kW pada 8000 ppm dan 3.86 kW pada 4500 ppm untuk simulasi *GT-Power* dan enjin prototaip. Dayakilas, kuasa brek dan pengunaan bahan api tentu brek juga ditentukan. Bagi tujuan perbandingan, sebuah enjin *Yamaha 125Z* dipilih memandangkan kapasiti silinder dan pinsip kerja yang sama dengan enjin prototaip. *Yamaha 125Z* menghasilkan dayakilas tertinggi diikuti simulasi *GT-Power* dan enjin prototaip. Perbezaan dayakilas purata antara *Yamaha 125Z* dan simulasi *GT-Power* adalah 13.06%. Nilai minimum penggunaan bahan api tentu brek untuk *Yamaha 125Z*, simulasi *GT-Power* dan enjin prototaip masing-masing adalah 280.42 g/kWh pada 3500 ppm, 351.08 g/kWh pada 5000 ppm dan 510 g/kWh pada 3500 ppm. Perbezaan puncak maksimum yang terhasil adalah disebabkan oleh perbezaaan rekabentuk kebuk pembakaran dan andaian yang digunakan dalam *GT-Power*.

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| | | |
|---|---|---|
| $A_p$ | - | Piston area |
| AFR | - | Air-fuel ratio |
| $AFR_s$ | - | Stochiometric air-fuel ratio |
| $AFR_t$ | - | Trapped air-fuel ratio |
| a | - | Crankshaft offset |
| B | - | Bore |
| BP | - | Brake power |
| $B_{fin}$ | - | Breadth of single fin |
| b | - | Cylinder wall thickness |
| bmep | - | Brake mean effective pressure |
| bsfc | - | Brake specific fuel consumption |
| C | - | Specific heat |
| $C_p$ | - | Specific heat at constant pressure |
| $C_v$ | - | Specific heat at constant volume |
| $C_D$ | - | Coefficient of discharge |
| $CR_{cc}$ | - | Crankcase compression ratio |
| $CR_g$ | - | geometric compression ratio |
| $CR_t$ | - | trapped compression ratio |
| $C_{fl}$ | - | Low calorific value for fuel |
| $D_{cw}$ | - | Counter weight diameter |
| $D_{fw}$ | - | Flywheel diameter |
| E | - | Energy |
| F | - | Force |

| | | |
|---|---|---|
| $F_{gas}$ | - | Force of gas |
| $F_i$ | - | Inertia force |
| $F_{res}$ | - | Total force acting on the piston |
| g | - | Gravitational acceleration |
| H | - | Piston height form TDC |
| $h_c$ | - | Heat transfer coefficient |
| I | - | Inertia |
| IP | - | Indicated power |
| isfc | - | indicated specific fuel consumption |
| j | - | Cylinder perimeter |
| $K_s$ | - | Coefficient of fluctuating speed |
| k | - | Thermal conductivity |
| $k_x$ | - | Thermal conductivity in X direction |
| $k_y$ | - | Thermal conductivity in Y direction |
| $k_z$ | - | Thermal conductivity in Z direction |
| $L_{st}$ | - | Stroke length |
| $L_{ts}$ | - | Trapped stroke length |
| $L_{cr}$ | - | Connecting rod length |
| $L_{ct}$ | - | crank throw length |
| $L_{fin}$ | - | Length of single fin |
| $l_c$ | - | Clearance volume height |
| $m_{air}$ | - | Mass of air |
| $m_{ca}$ | - | Mass of cooling air |
| $m_f$ | - | Mass of fuel |
| $m_{fw}$ | - | Mass of flywheel |
| $m_{cp}$ | - | Mass of crank pin |
| $m_{crbe}$ | - | Mass of connecting rod (big end) |
| $m_{crm}$ | - | Mass of connecting rod (main) |
| $m_{crse}$ | - | Mass of connecting rod (small end) |
| $m_{cw}$ | - | Mass of counter weight |
| $m_p$ | - | Mass of piston |

| | | |
|---|---|---|
| $m_{pp}$ | - | Mass of piston pin |
| $m_{ppb}$ | - | Mass of piston pin bearing |
| $m_{tcrbe}$ | - | Total mass of connecting rod (big end) |
| $m_{rot}$ | - | Mass of rotating components |
| $m_{rec}$ | - | Mass of reciprocating components |
| $m_{ta}$ | - | Mass of trapped air |
| $m_{tf}$ | - | Mass of trapped fuel |
| $m_{tr}$ | - | Total mass of charge |
| N | - | Speed |
| n | - | Number of cylinder |
| P | - | Pressure |
| Q | - | Volumetric heat generation rate |
| $Q_{cool}$ | - | Heat to be cooled |
| $Q_R$ | - | Heat released from combustion |
| q | - | Heat flux |
| R | - | Ratio of connecting rod length to crank radius |
| $R_a$ | - | Gas constant for air |
| $R_{tr}$ | - | Gas constant at trapped point |
| $r_{fw}$ | - | Flywheel radius |
| $S_p$ | - | Mean piston speed |
| $S_{exh}$ | - | Exhaust perimeter |
| $S_{main}$ | - | Main port perimeter |
| $S_{side}$ | - | Side port perimeter |
| sfc | - | Specific fuel consumption |
| T | - | Temperature, torque |
| $T_f$ | - | Tangential force |
| $T_i$ | - | Indicated torque |
| $T_o$ | - | Specified temperature |
| $T_s$ | - | Surface temperature |
| $T_{at}$ | - | Ambient temperature |
| $T_{tr}$ | - | Trapped temperature |

| | | |
|---|---|---|
| $T_\infty$ | - | Ambient temperature |
| $t$ | - | Time |
| $t_{cw}$ | - | Counter weight thickness |
| $t_1$ | - | Thickness cylinder liner |
| U | - | Thermal load |
| V | - | Volume |
| $V_{cc}$ | - | Crankcase volume |
| $V_{cv}$ | - | Clearance volume |
| $V_{cw}$ | - | Counter weight volume |
| $V_{sv}$ | - | Swept volume |
| $V_{ts}$ | - | Trapped swept volume |
| $V_{tr}$ | - | Trapped volume |
| $\overline{V}_{fw}$ | - | Flywheel velocity |
| X | - | Piston position |

**Greek Symbols**

| | | |
|---|---|---|
| $\Phi$ | - | Angle of obliquity |
| $\gamma$ | - | Specifics heat ratio |
| $\varepsilon$ | - | Emissivity |
| $\omega$ | - | Angular velocity |
| $\rho$ | - | Density |
| $\rho_{at}$ | - | Air density |
| $\rho_{cw}$ | - | Density of counter weight |
| $\sigma$ | - | Stefan-Boltzmann constant |
| $\sigma_{fw}$ | - | Stress of the flywheel |
| $\sigma_c$ | - | Stress of cast iron |
| $\theta$ | - | Crank angle |

| | | |
|---|---|---|
| $\eta_c$ | - | Combustion efficiency |
| $\eta_i$ | - | Indicated thermal efficiency |
| $\eta_t$ | - | Thermal efficiency |

## LIST OF ABBREVIATIONS

| | | |
|---|---|---|
| AC | - | Alternate current |
| AFR | - | Air-fuel ratio |
| ATDC | - | After top dead center |
| A/D | - | analogue/digital |
| BDC | - | Bottom dead center |
| BTDC | - | Before top dead center |
| CA | - | Crank angle |
| CAD | - | Computer-aided design |
| CCD | - | Charge-coupled device |
| CDI | - | Capacitive discharge ignition |
| CFD | - | Computational fluid dynamics |
| CI | - | Compression ignition |
| CO | - | Carbon monoxide |
| CPU | - | Central processing unit |
| CVT | - | Continuous variable transmission |
| DAS | - | Data acquisition system |
| DFI | - | Direct fuel injection |
| DHA | - | Detail hydrocarbons analysis |
| ECU | - | Electronic control unit |
| EFIC | - | Electronic fuel injector controller |
| EGR | - | Exhaust gas recirculation |
| EPA | - | Environmental Protection Agency |
| EMS | - | Engine management system |

| | | |
|---|---|---|
| FEA | - | Finite element analysis |
| GDI | - | Gasoline direct injection |
| HC | - | Hydrocarbons |
| HCCI | - | Homogenous charge compression ignition |
| HPSGDi | - | High pressure swirl gasoline direct injector |
| ICE | - | Internal combustion engine |
| I/O | - | Input/output |
| MAP | - | Manifold absolute pressure |
| MTBE | - | Methyl tertiary-butyl ether |
| NO | - | Nitric oxides |
| $NO_x$ | - | Nitrogen oxides |
| PCB | - | Printed circuit board |
| PFI | - | Port fuel injection |
| PID | - | Proportional, integral and differentiate |
| PM | - | Particulate matter |
| PRV | - | Pressure relief valve |
| PWM | - | Pulse-width modulation |
| RAM | - | Read-access memory |
| ROM | - | Read-only memory |
| R&D | - | Research and development |
| SI | - | Spark ignition |
| SMD | - | Sauter mean diameter |
| SOI | - | Start of injection |
| TDC | - | Top dead center |
| TPS | - | Throttle position sensor |
| UTM | - | Universiti Teknologi Malaysia |
| ULEV | - | Ultra low emission vehicle |
| 2-T | - | Lubrication oil |

# LIST OF APPENDICES

# CHAPTER 1

# INTRODUCTION

## 1.1 Introduction

Internal combustion engine (ICE) technologies have undergone changes through a series of phases since the 18$^{th}$ centuries. In the old days ICE was purely on mechanical principles. With the advancement of technology, ICE had change from totally mechanical dependent to electronic and computerized controlled system. No matter what new technology was incorporated onto the engine, the main objective is to improvise and develop a more efficient ICE in terms of output and emissions.

The improvement is basically to reduce fuel consumption or in other words to utilise fuel efficiently. To achieve this goal, several concepts can be implemented. The concepts are briefly shown in Figure 1.1.

Particularly for combustion efficiency, the combustion concepts basically based on flow pattern are swirl, forward tumble and reverse tumble. Some of these concepts have already been commercialized while others are still in research stage.

In the last twenty years, the environmental implications of ICE exhaust emissions have resulted in greater effort in reducing harmful pollutants to meet stringent legislation. The investigation and the study on combustion and pollutant formation processes in ICE are clearly necessary to increase their efficiency and economic of operation. This lead to the evolution and invention of new engines with emphasise on increased efficiency and emissions reduction.

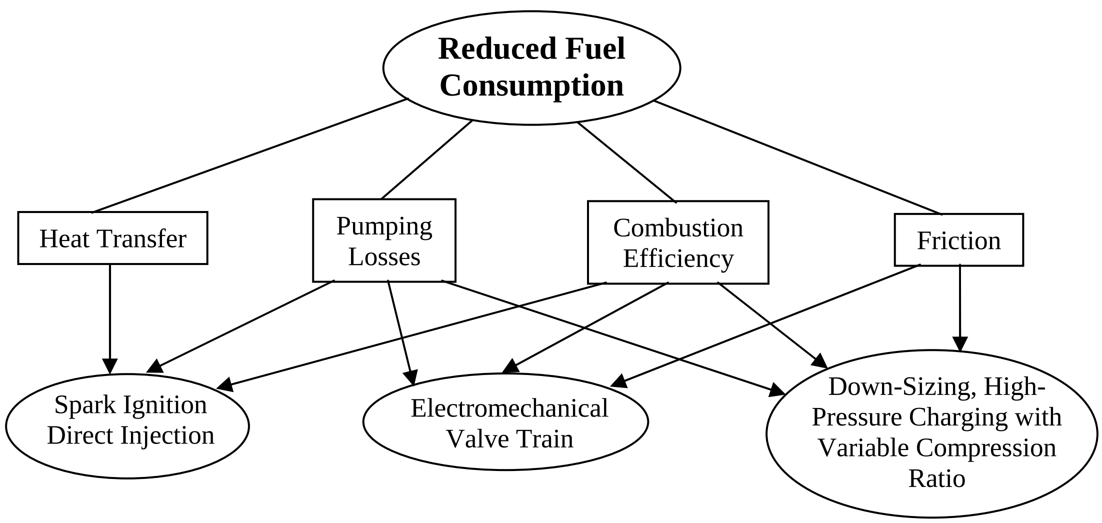


**Figure 1.1:** Concepts of current/future engine design

Motorcycles and motorised scooters are popular form of transport throughout the world. They are particularly popular in congested cities in Asia and Europe. In Asian regions they are dominant mode of transport because of their low initial and ongoing costs. Traditionally, two-stroke engines have powered the most popular models of motorcycles and scooters. Powered tools such as chainsaw, trimmer, fruit harvester, grass cutter and even bicycle utilised the two-stroke engine as their prime mover.

Two-stroke engines are popular because they produce relatively high power from a small size and are also inexpensive to manufacture and maintain. This is because the two-stroke engine requires no valvetrain which contribute in major cost, weight and complexity.

The current two-stroke designs are inefficient and impart environmental implications to human and plants in urban and non-urban areas where they operate. In Asian cities, they constitute the bulk of the engines for use in motorcycles and three-wheelers. With the hazardous nature of the pollutants emitted by the vehicles they are banned in Taipei, while in Bangkok the authority is seriously considering doing the same. The somewhat low combustion efficiency of today’s two-stroke engine design can be improved through various means, making it possible to compliment the four stroke version. The latter is a more complicated engine configuration having lesser power-to-weight ratio characteristic. During the last decade, there have been many improved versions of the two-stroke engines produced. A few have found commercial ventures while many are still at the prototype level or even on the drawing board.

## 1.2 Background of the Problem

The simplistic nature of the design of a two-stroke engine, with relatively simple machining process (compared to four stroke engine), will provide improved power-to-weight ratio characteristic and lower production cost. Development of new two-stroke engines must address scavenging and emission problems. This can be achieved through improvising the scavenging process. By means of multi-ports intake system and eliminating crankcase compression, better scavenging process could be achieved. The integration of transfer ports design to create better flow angle for blowdown and loop scavenging of exhaust gases will help improve fuel consumption and exhaust emissions in general. In this study the three numbers of ports was selected due to the constraints of surface area. As for the eliminating of the crankcase compression, stepped-piston configuration was selected in order to prepare the compress air-fuel mixture for the scavenging process.

## 1.3 Research Challenges

The key research challenge that must be resolved is can a multi-port intake system improve the performance and emission of a stratified-charged two-stroke stepped-piston gasoline engine.

## 1.4 Research Questions

Some of the persistent questions at this juncture would be:

i) How multi-ports intake system and stepped-piston can fit in a two-stroke engine?
ii) How is the optimise condition be achieved?
iii) What effect will it have on performance and emission?
iv) What aspect of performance (torque, brake power, sfc, etc)?
v) What speed regime? Low speed? Typically idle to maximum speed?
vi) What load range?
vii) What are the impacts towards the unregulated and regulated exhaust emission?

## 1.5 Hypothesis

i) Multi-ports and stepped piston provides an effective mean of discarding exhaust gases by introducing fresh charge through the circumference of the cylinder.

ii) It also improves the scavenging process by reducing the short-circuiting of fresh charge air-fuel mixture.

## 1.6 Objectives of the Study

By introducing the multi-ports concept into the design of stratified-charged stepped-piston engine the associated problems of the conventional design of two-stroke engine can be addressed. The objectives of this research are as follows,

i) to analyse the impact of multi-ports and stepped-piston configuration towards improving engine performance and emission,

ii) to determine the appropriate ports design configuration and location, and

iii) to compare the deviation between simulation and experimental results.

## 1.7 Importance of the Study

The study could benefit toward enhancing stratified-charged, two-stroke, stepped-piston gasoline engine design with the potential for application in transport, industrial and agriculture sectors. Furthermore, the future of the two-stroke engine will depend upon how far a successful combination of scavenging, charging, combustion technology improvement (this study) can be made without jeopardising its classical advantages in terms of power density and simple construction.

## 1.8 Scope of the Study

Scopes of the study are as follows:

i) Design and simulation.

Preliminary study of the workability of the device is vital before the actual device is being fitted to the actual engine. The best configuration of the device will be design based on the actual boundary condition from the actual engine operation.

ii) Development and testing the prototype engine in laboratory for performance and emission.

A two-stroke engine will be developed to distinguish the workability of the system.

iii) Engine testing for performance and emission.

In this exercise two intensive testing will be conducted. Firstly the developed engine characteristic will be defined and secondly an engine which is available in the market which has the same capacity will be undergoing the same testing for comparative study.

iv) Data analysis.

All the data obtained from the testing will be analysed in order evaluate engine performance and emission.

In general, the outline constitutes three major areas in meeting the objectives and they are:

i) Simulation of flow and engine performance

To observe the flow characteristic during scavenging of exhaust gases and also the compression of fresh air-fuel mixture by the steppe-piston. As for the engine performance simulation is to observe the typical engine characteristics based on the engine geometry.

ii) Engine development

To look into the appropriate design features in order to have a optimize engine. This exercise will involve decision making in choosing the right method in producing each component of the engine without compromising with the vital features.

iii) Engine performance testing.

The testing comprises of conventional two-stroke and prototype engine. The purpose of the conventional engine testing is intended for comparative study.

The overall perspective of the research program is shown in Figure 1.2.

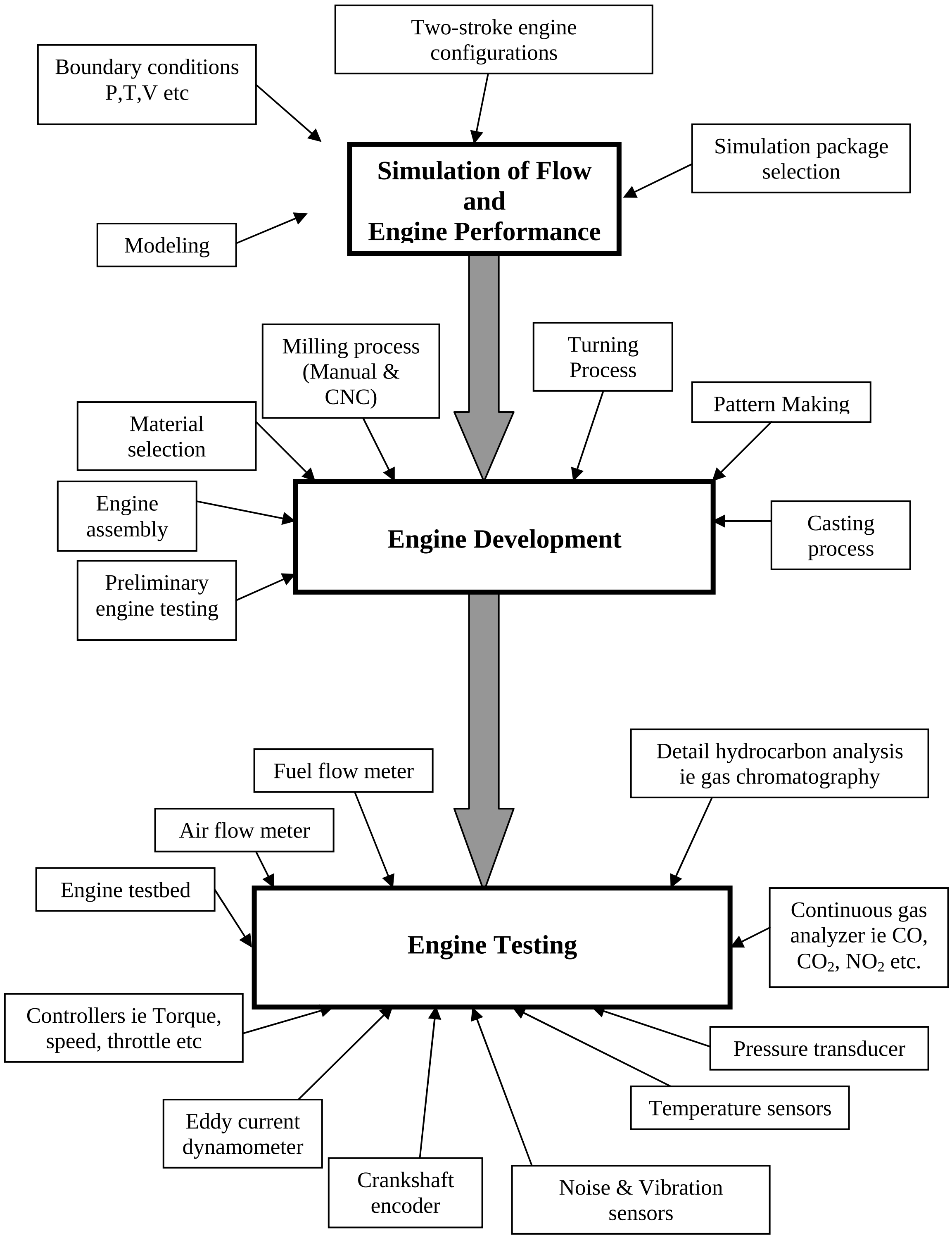


**Figure 1.2:** Overall perspective of research program

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