



**DEVELOPMENT OF A RAM-SWIRLED HELICAL AIR INTAKE DEVICE
FOR A COMPRESSION-IGNITION ENGINE**

By

MOHD SHAHRUDIN BIN MOHD ALIAS

**Thesis Submitted to the School of Graduate Studies, Universiti Putra
Malaysia, in Fulfilment of the Requirements for the Degree of Doctor of
Philosophy**

August 2024

FK 2024 36

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August 2024

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Compression-ignition CI engines are highly efficient power devices compared to Spark Ignition SI engines and currently play a significant role in the transportation and industrial sectors. However, the engine has constraints with low engine performance, high fuel consumption and emissions levels which require continuous improvement. There are various designs of air intake systems to meet a certain engine operating range, still, there will be less actual amount of fresh charge entering the cylinder compared to the theoretical value due to the short cycle time available. This results in poor mixing of fuel-air mixture, increasing the incomplete combustion and amount of unburned fuel. The volumetric efficiency is also affected by the lower amount of intake air charged which contributed to the performance constraint. The study analyses the effect of the swirling airflow in CI engines on combustion behaviour, emissions and performance experimentally. The swirling air is generated by a novel design of ram-swirled air intake device studied with various parameters.

The parameters include helical blade, cutting angle and cylinder draft angle. To come out with the final design, a simulation is conducted in Ansys Fluent software to identify the output flow direction and swirl strength based on the base model. A parametric study is further conducted by simulation to verify the output flow characteristics by changing the model's aspect ratio. The experiment for the 3D-printed device is done on a single-cylinder, four-stroke, direct injection diesel engine connected to an eddy current dynamometer. The result shows that the addition of the basic design of the swirl air intake device increases the engine performance at full load with the brake power and torque increase by 2.9% to 5.3% at various engine speeds. The emissions levels are also improved with a significant drop by 34.6% to 55.3% in the NO_x, UHC and CO levels. The device has shown the ability to enhance the air-fuel mixing in the combustion chamber based on the in-cylinder pressure, where the SOC event starts earlier for shorter ignition delay as a result of better homogenous mixing. Experimental results of the parametric study show a consistent, good agreement and better performance with further increase in the brake power and torque by up to 5.8% to 18.4% at various engine speeds. These results conclude that the swirl air intake devices are capable of increasing the overall performance of the CI engine. This includes the improvement to the BSFC, BTE and exhaust emissions level which highly contributed by the enhancement of fuel-air mixing homogeneity.

Keywords: Air Intake, Compression Ignition, Emissions, Ram, Swirl

SDG: Goal 7: Affordable and Clean Energy, Goal 9: Industry, Innovation and Infrastructure

Abstrak tesis yang dikemukakan kepada Senat Universiti Putra Malaysia
sebagai memenuhi keperluan untuk ijazah Doktor Falsafah

**PEMBANGUNAN PERANTI PENGAMBILAN UDARA HELIKS RAM-
BERPUSAR UNTUK ENJIN PENCUCUHAN-MAMPATAN**

Oleh

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Enjin pencucuhan mampatan adalah peranti kuasa yang cekap berbanding dengan enjin cucuhan bunga api dan kini amat signifikan bagi sektor pengangkutan dan perindustrian. Walaubagaimanapun, ia perlu ditambahbaik disebabkan prestasi enjin yang rendah, penggunaan bahanapi dan paras emisi yang tinggi. Terdapat pelbagai rekabentuk sistem pengambilan udara untuk memenuhi julat operasi tertentu, namun masih terdapat kekurangan jumlah sebenar cas udara baru yang memasuki silinder disebabkan masa kitaran yang singkat. Ini menyebabkan percampuran bahanapi dan udara yang kurang baik lalu menyebabkan pembakaran tidak sempurna serta bahanapi yang tidak terbakar. Kecekapan isipadu juga dipengaruhi oleh kurangnya jumlah cas udara yang masuk ke ruang silinder enjin. Kajian ini menganalisa kesan udara berpusing secara bereksperimen ke atas kelakuan pembakaran, tahap pancaran dan prestasi enjin pencucuhan mampatan. Udara berpusing dijana oleh rekabentuk baharu peranti pengambilan udara berpusing '*ram*'

merangkumi rekaan bilah heliks, sudut pemotongan dan sudut draf silinder. Untuk mendapatkan rekabentuk akhir, simulasi dijalankan menggunakan perisian *Ansys Fluent* bagi mengenalpasti arah aliran dan kekuatan pusaran udara berdasarkan model asas. Kajian berparameterik melalui simulasi digunakan bagi mengesahkan ciri-ciri aliran udara dengan menukar nisbah bidang model yang terlibat. Percubaan pada peranti cetakan 3-dimensi dilakukan menggunakan enjin diesel silinder tunggal, empat lejang, suntikan terus yang disambungkan kepada dinamometer arus pular. Penggunaan peranti rekaan asas didapati mampu meningkatkan prestasi enjin pada beban penuh dengan kuasa dan kilas brek enjin yang meningkat sebanyak 2.9% sehingga 5.3% pada pelbagai kelajuan enjin. Kualiti tahap pelepasan juga dipertingkatkan dengan penurunan sebanyak 34.6% sehingga 55.3%. Peranti asas telah menunjukkan keupayaan untuk menambah baik percampuran bahan api dan udara dalam kebuk pembakaran di mana Permulaan Pembakaran bermula lebih awal hasil daripada campuran homogen yang lebih baik. Keputusan eksperimen peranti rekaan berparametrik menunjukkan persetujuan yang konsisten dan prestasi keseluruhan yang lebih baik dengan peningkatan kuasa dan kilas brek sebanyak 5.8% sehingga 18.4%. Keputusan ini menyimpulkan bahawa peranti udara pular mampu meningkatkan prestasi keseluruhan enjin termasuk penambahbaikan terhadap penggunaan bahanapi tentu brek, kecekapan haba brek dan pelepasan emisi ekzos ke persekitaran.

Kata Kunci: Berpular, Pelepasan Emisi, Pencucuhan Mampatan, Pengambilan Udara, Ram

SDG: MATLAMAT 7: Tenaga yang Berpatutan dan Bersih, MATLAMAT 9: Industri, Inovasi dan Infrastruktur

ACKNOWLEDGEMENTS

Thank you for the blessings of ALLAH SWT, The Almighty in the completion of my thesis. Foremost, I would like to acknowledge my supervisor, Assoc. Professor Dr. Abdul Aziz Hairuddin, who has been most encouraging, understanding and provided tremendous support, ideas, thoughts, commitment and guidance throughout her supervision of my thesis. I also would like to thank my co-supervisors, Dr. Khairil Anas Md Rezali and Assoc. Professor Ir. Ts. Dr. Mohd Khair Hassan for being supportive during the study. I would like to acknowledge the contribution and support from Universiti Putra Malaysia (Department of Mechanical and Manufacturing Engineering) in providing the financial support and facilities in order to complete the research and testing.

I wish to gratefully acknowledge my colleague for their cooperation and contribution to the case studies especially Mr. Firdaus, Mr. Zulkipli, Ir. Nik Hafiz and Mr. Muntasser. Huge thanks to those who directly or indirectly contributed in providing their support, technical knowledge, expertise and insight.

Most of all, a special thank you and my deepest gratitude to my beloved wife Siti Hanisah for all her patience and unwavering supports, which were so essential throughout the study years. To Safwan, Sufi, Syahid and Asyfina, thanks for their love. I would highly appreciate the support and prayers from my beloved parents, Mr. Mohd Alias Wagiran, and Mrs. Huzaimah Hussein. Not to forget Hj. Mohd Shariffudin Adnan and Hjh. Fatimah Omar for all their kindness and invaluable support throughout the years of study.

This thesis was submitted to the Senate of Universiti Putra Malaysia and has been accepted as fulfilment of the requirement for the degree of Doctor of Philosophy. The members of the Supervisory Committee were as follows:

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LIST OF ABBREVIATIONS

ABS	Acrylonitrile butadiene styrene
AFR	Air-Fuel Ratio
ANOVA	Analysis of Variance
ASTM	American Society for Testing and Materials
BDC	Bottom Dead Centre
BMEP	Brake Mean Effective Pressure
BP	Brake Power
BSFC	Brake Specific Fuel Consumption
BT	Brake Torque
BTE	Brake Thermal Efficiency
CI	Compression Ignition
CFD	Computational Fluid Dynamics
CO	Carbon Monoxide
CO ₂	Carbon Dioxides
CPS	Cam Profile Switching
DI	Direct Injection
DOE	Design of Experiment
ECU	Engine Control Unit
EGR	Exhaust Gas Recirculation
FAT	Focus Applied Technologies
GUI	Graphical User Interface
GVD	Guide Vanes Design
HCCI	Homogeneous Charge Compression Ignition

HHV	High Heating Value
HRR	Heat Release Rate
IC	Internal Combustion
IM	Intake Manifold
IMEP	Indicated Mean Effective Pressure
ITE	Indicated Thermal Efficiency
LHV	Low Heating Value
MAF	Mass Air Flow
NO	Nitric Oxide
NO ₂	Nitrogen Oxides
NVH	Noise, Vibration and Harshness
PRR	Pressure Release Rate
rpm	Revolution Per Minute
SN	Signal-to-Noise Ratio
TDC	Top Dead Centre
TED	Turbulence Energy Dissipation
TKE	Turbulence Kinetic Energy
UHC	Unburned Hydrocarbon
VIM/VLIM	Variable Intake Manifold/Variable Length Intake Manifold
3D	3-Dimensional

LIST OF SYMBOLS

Symbols

$^{\circ}\text{C}$	Degree Celcius
$^{\circ}\text{CA}$	Degree Crank Angle
ρ	Density
$k\text{-}\varepsilon$	k-epsilon
$\Omega(\omega)$	Angular Velocity
η_{th}	Thermal Efficiency
η_v	Volumetric Efficiency
G	Axial Flux
J	Joule
K	Kelvin
M	Moment of Mean Component
$\dot{m}_a$	Mass Flow Rate of Air
$\dot{m}_f$	Mass Flow Rate of Fuel
N	Crankshaft Angular Speed
Q	Quantity of Heat
r	Radial Component of Velocity
R	Coefficient of Correlation
R^2	Coefficient of Determination
R_s	Swirl Ratio
S	Swirl Number
u	Axial Component of Velocity
w	Tangential Component of Velocity

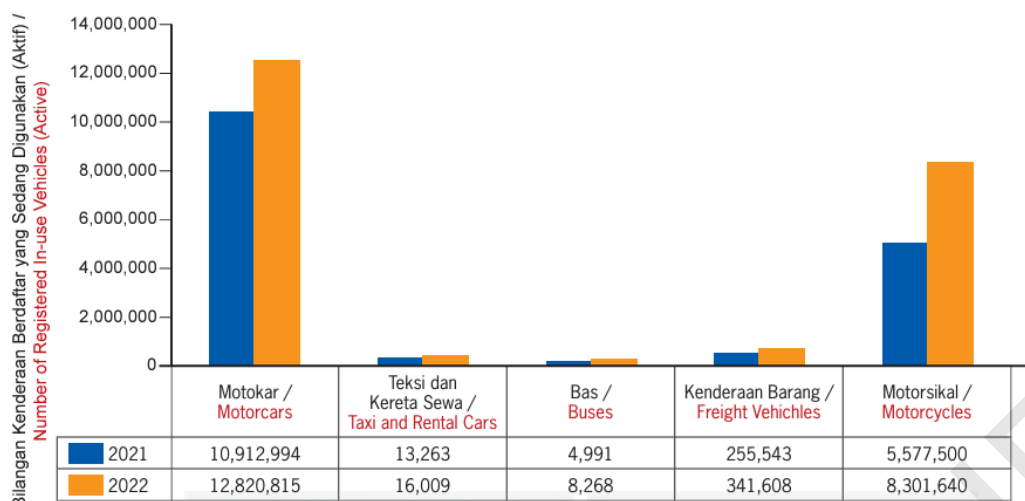
CHAPTER 1

INTRODUCTION

1.1 Background of Study

CI engine has good volumetric and combustion efficiencies. However, it produces relatively higher emissions levels of HC, CO, NO_x and Particulate Matters (PM) compared to the SI engine (Alabi, 2015). Nitrogen oxide (NO_x) for example, is emitted by a CI engine more than that of the SI mainly due to the higher combustion temperature caused by the presence of the high amount of oxygen inside the combustion chamber (K. Ramalingam et al., 2022). Moreover, due to the wider range of applications by CI engines, it is being largely used in both transportation and industrial sectors (J. Li et al., 2023). This causes the exhaust emissions produced by this engine to highly contributes to the overall increase of air pollutants.

As reported by the authorities, non-government organisations and researchers (ASEAN Secretariat, 2023; Department of Environment Malaysia, 2022; Shafie, 2024; Shafie & Mahmud, 2020), there is an increasing trend of air pollutants which is mainly contributed by the increase in motor vehicle ownership and an upward annual trend of the fuel consumption from the transport sector. Figures 1.1 and 1.2 show the trend of the motor vehicle registration and the air pollutant emission load from motor vehicles in Malaysia from year 2021 to 2022.



(Sumber: Jabatan Pengangkutan Jalan, Malaysia 2022)
(Source: Road Transport Department, Malaysia, 2022)

Figure 1.1: Number of Registered In-Use Vehicles (Active) 2021-2022.
Reproduced from Environmental Quality Report 2022
(Source: Department of Environment Malaysia, 2022)

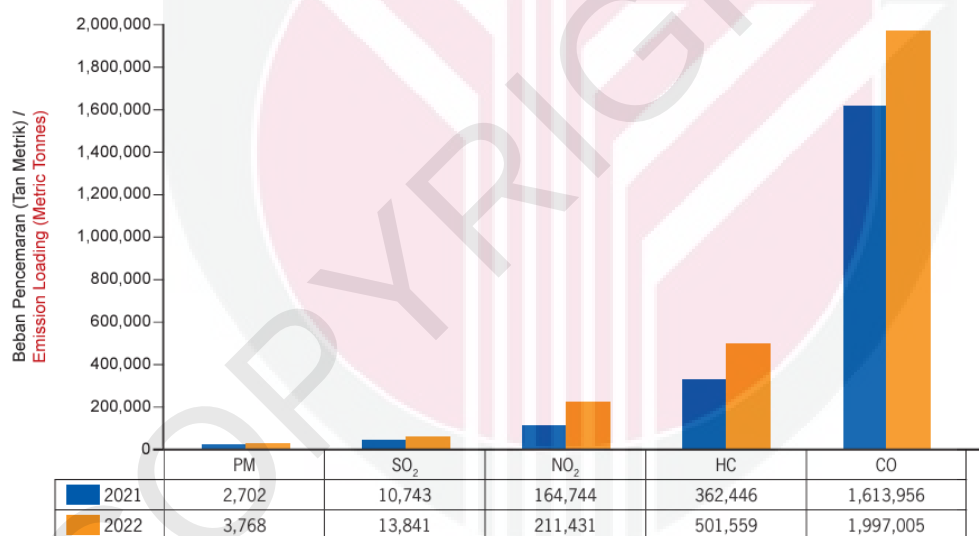


Figure 1.2: Air Pollutant Emission Load From Motor Vehicles 2021-2022 (Metric Tonnes). Reproduced from Environmental Quality Report 2022
(Source: Department of Environment Malaysia, 2022)

There is a plan to strengthen the emission regulation and legislation by the ASEAN countries such as Malaysia, Singapore, Indonesia, Philippines and Vietnam in an effort to further reduce gas emissions and air pollutants (ASEAN Secretariat, 2023; Owen & Tao, 2015). For example, Malaysia has currently imposed stricter emissions control for diesel engines that was previously planned to be effective by the year 2023. This was mentioned in the application guidelines of the 'Motor Vehicle Exhaust Gas and Noise Emissions Standard' produced by the Department of Environment Malaysia (2017) as shown in Table 1.1 involving both new and existing vehicles.

Since the enrolment of the Euro 5 diesel nationwide is already effective in October 2021 (Petronas, 2021), it is assumed that the regulation for new and existing vehicle exhaust emissions control has come into effect based on the published guidelines. Thus, the stringent emission control with the Euro 4 standard requires extensive research to improve the current internal combustion engine without delay.

Table 1.1: Exhaust emission standards for diesel engines. Extracted from the Application Guidelines of Motor Vehicle Exhaust Gas and Noise Emission Type Approval

DIESEL					
EXHAUST EMISSION		EURO II		EURO IV	
CATEGORY OF VEHICLE		M (Maximum Mass $\leq$ 3.5 tonnes) and N1	M (Maximum Mass $>$ 3.5 tonnes); N2; N3	M (Maximum Mass $\leq$ 3.5 tonnes) and N1	M (Maximum Mass $>$ 3.5 tonnes); N2; N3
IMPLEMENTATION	NEW MODEL	1 January 2017		18 months grace period after the availability of Euro 5 diesel nationwide; or Tentative date: 1 March 2022	
	EXISTING MODEL	Not applicable		18 months grace period after the availability of Euro 5 diesel nationwide; or Tentative date: 1 Sep 2023	
REGULATION STANDARD		UN R83-03; Or Directive 94/12/EC; Or Directive 96/69/EC	UN R49-02 (B); Or Directives 88/77/EEC as amended by 91/542/EEC and 96/1/EEC	UN R38-05 (B); Or Directive 70/220/EEC as amended by 98/69/EC and 2003/76/EC	UN R49-05 (B1); Or Directives 1999/96/EC; Or Directives 2005/55/EC. Or Directives 2006/51/EC
TEST REQUIREMENT		Type I Refer to Table 6	13-mode steady-state diesel engine test (To follow R49-02(B) limit values) Refer to Table 7	Type I (To follow R38-05(B) limit values Refer to Table 8 Type V Refer to Table 9	(1)* European Steady-State Cycle (ESC); and (2)* European Load Response (ELR) Test Cycle; and [(1)* and (2)* to follow UN R49-05 (B1) limit values] refer to Table 10 (3)* European Transient Cycle (ETC) [(3)* to follow UN R49-05 (B1) limit values] refer to Table 11

(Adapted: Department of Environment Malaysia, 2017)

CI engine has another major advantage for future engine developments. Researchers have seen the potential of an advanced CI engine operation in providing high engine efficiency, and at the same time, serving as the platform for other modes of combustion to be developed in reaching lower emissions

potentially with improved efficiency. Some of the advanced engine operations such as Homogenous Charge Compression Ignition (HCCI), Reactivity Controlled Compression Ignition (RCCI), Premixed Charge Compression Ignition (PCCI) and Gasoline Direct-Injection Compression Ignition (GDCI) are presented by numerous studies (Alemayehu et al., 2022; Duan et al., 2021; Duronio et al., 2020; Kumar Subramani et al., 2022; Riyadi et al., 2023; Vasudev et al., 2022).

The use of the CI engine when converted to either of these combustion modes will have further potential in exploring alternative fuels which is initially impossible in a CI engine. This also means the improved engine performance, efficiencies and emissions achieved in CI mode with the enhanced fuel-air mixing may be further improved through the other modes of combustion with more options in alternative fuels.

To be more specific on the engine parameters, fuel-air mixing plays a significant role in the Compression Ignition (CI) combustion process which proper fuel-air mixing leads to better combustion, then results in an improved performance output. In a diesel engine, the fuel is injected into the cylinder when the air is already at high pressure and temperature, near the end of the compression stroke. The combustion process starts when the portions of the developing mixture of already injected and vaporised fuel self-ignited (autoignition) with the hot air. It rapidly spreads out and later burns as fuel and air are mixed to the appropriate composition for the combustion stage. Fuel-air mixing is the primary controlling factor for the diesel-combustion heat

release. An increase in fuel-air mixing rates will increase the heat-release rate during the mixing-controlled combustion phase (Heywood, 2018).

The diesel combustion process is more complicated than the Spark Ignition (CI) combustion process because the vaporisation of liquid fuel and fuel-air mixing processes are involved (Heywood, 2018). A major diesel engine design problem is to achieve sufficiently rapid fuel-air mixing rates to complete the fuel-burning process in the time available. Thus, different combustion chamber geometries and fuel-injection characteristics are required to deal effectively. It is well known that, in diesel or petrol engines, efficiency depends on the quality of the combustion. On the other hand, the quality of the combustion process is tied directly to the oxygen-fuel mixing characteristics existing in the cylinder, resulting from the injection of the fuel during the admission stage. This is why the fuel-air mixture must be as homogeneous as possible.

This homogeneity is directly tied to the mean velocity field and to the turbulence characteristics in the cylinder (Crnojevic et al., 1999). In diesel engines, the injection of fuel droplets is not totally uniform. A motionless incoming air will have insufficient oxygen for the fuel burning, thus resulting in either slow combustion or failure. The swirling movement inside the combustion chamber will affect the fuel-air mixture inside the combustion chamber and allow the mixing of fuel droplets with air (Akshay et al., 2017).

The air intake manifold plays a greater role in ensuring an even distribution of the filtered air among all the cylinders from the air intake pipe. In a conventional CI engine, the charged air intake flows steadily from the plenum chamber to

the cylinder, where the plenum helps to ensure a consistent clean air supply to all cylinders. Apart from that, there is a more basic design of an air intake system with just an air cleaner assembled directly to the inlet port, especially for a small and lighter engine. Figures 1.3 and 1.4 show two different air intake system configurations of a diesel engine.

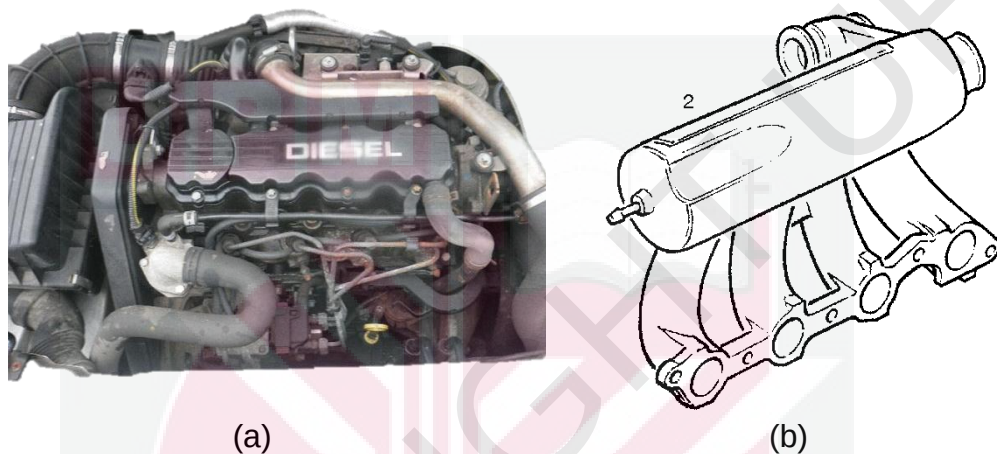


Figure 1.3: Opel Vauxhall Astra F Diesel Engine. (a) Diesel engine layout complete with an airbox assembly in the engine bay, (b) Astra F air intake manifolds schematic diagram

(Source: 247Spares UK, 2024; 7zap.com, 2024)

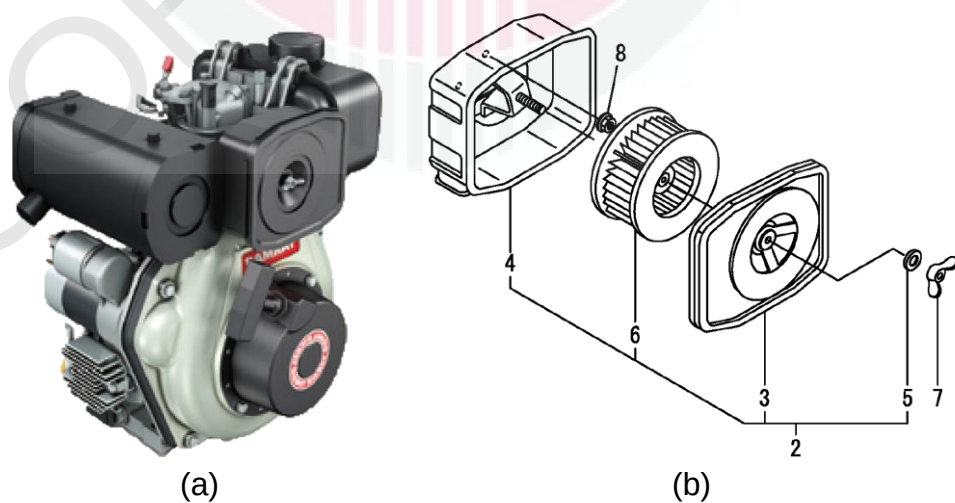


Figure 1.4: Yanmar Industrial Diesel Engine. (a) Complete assembly diesel engine layout, (b) Yanmar air cleaner schematic diagram

(Source: Yanmar, 2007, 2011, 2017)

It is commonly known that diesel engines are typically run with a maximum air intake without the requirement of a throttle body as the speed is governed by the amount of fuel that enters the engine. If equipped with a throttle body, the main reason is to provide greater control of the intake manifold pressure. Nonetheless, no further discussion regarding the throttle body mechanism as it is not a common part required to operate a diesel engine.

1.2 Problem Statement

Combustion quality and performance of CI engines have steadily improved over the years, but there are still some disadvantages and challenges associated with their combustion and performance. Combustion efficiency in CI engines is one of the challenges where the factor of fuel-air mixing, fuel atomisation and design of the engine may influence the overall efficiency. At various conditions, the combustion behaviour can be easily affected due to the variation in one of these factors. Under certain circumstances, the variations lead to poor mixing of the fuel-air mixture or poor fuel atomisation, causing more incomplete combustions. Incomplete combustion in CI engines greatly affects fuel efficiency and power output. The inefficiency of the CI engine combustion also causes poor exhaust emissions control. The failure to find a way to improve the emission will cause the engine to fail to meet the stricter emission regulations.

The application guideline for the Euro IV diesel engine exhaust emission standard was published by the Department of Environment Malaysia in 2017 as previously shown in Table 1.1. The guideline is to be implemented

tentatively for the new and existing vehicle models following the availability of the Euro 5 diesel fuel nationwide or towards the end of the year 2023. This new guideline is necessary in regulating the emission level coming from the motor vehicle. From Figure 1.5, it shows the main source of CO emission is mainly from 'Motor Vehicles' at 96.43%, followed by 'Industries' at 2.95% and 'Power Station and Others' at 0.61%. This is a clear indicator that the existing internal combustion engine needs an additional mechanism and relevant research to reduce the emission level of motor vehicles.

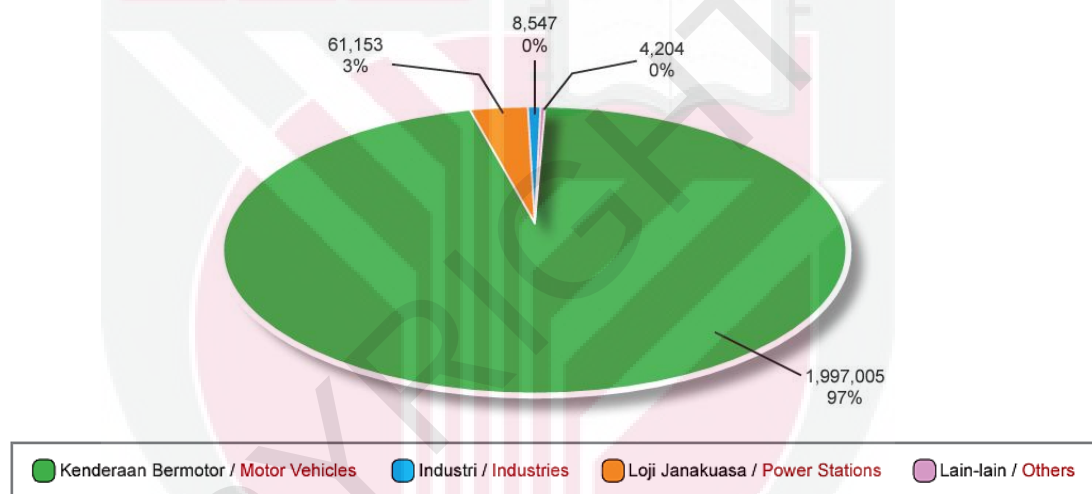


Figure 1.5: Carbon Monoxide (CO) Emission Load by Sources (Metric Tonnes). Reproduced from Environmental Quality Report 2022
(Source: Department of Environment Malaysia, 2022)

CI engines have constraints in the combustion duration due to the nature of the pre-combustion and combustion process which starts from the fuel injection into the compressed air, followed by an ignition delay. The delay period varies based on the level of the physical and chemical parameters which is very dependent on the fuel properties (Erman et al., 2020; Miron et al., 2021; Shahabuddin et al., 2013). This delay is highly crucial to combustion

efficiency where the variation of this characteristic greatly affects the energy or heat release rate profile (Erman et al., 2020). Ignition delay characterises the initiation of the combustion process and the subsequent phases where a prolonged ignition delay period leads to later SOC (Erman et al., 2020; Miron et al., 2021). Autoignition happens once the mixture of injected fuel and vaporised fuel is self-ignited with the high temperature of air. After the pre-combustion process, the CI engine then has to go through three combustion phases after the start of combustion (SOC) until the late combustion phase before starting the new cycle. The combustion duration also varied depending on the physical and chemical processes in the combustion chamber (Y. Chen et al., 2021; V. Ganesan, 2012). The requirement to complete all these processes limits the engine to run efficiently at a higher speed range.

In terms of performance, the speed range of the CI engines is typically narrower compared to SI engines. Diesel engines are commonly used for lower to mid-range speeds due to higher compression ratio, more torque, better efficiency and power-to-weight ratio at this range (Cole, 2014; Driv Inc., 2022). This makes them more suitable for heavy-duty applications such as trucks and industrial machinery. However, this engine is not able to perform well in high-speed applications such as motorcycles and performance cars, where SI engines are preferred. The limitation of the CI engine performance in high-speed applications is mainly due to the constraint in the longer combustion duration, longer ignition delay and higher proportion of combustion diffusion (Sun et al., 2017). Thus, improvement to the combustion characteristics such as optimising ignition delay and minimising incomplete

combustions should be the main focus in extending the operating range of the diesel engines, apart from the fuel properties and injection rate.

In terms of exhaust emissions, a CI engine has a relatively higher emissions level than an SI engine. Regardless of the type of fuel, in a CI engine, the fuel is injected directly into the combustion chamber at very high pressure typically around 200 bar, several degrees before Top Dead Centre (TDC). Direct injection (DI) can lead to the fuel-rich regions within the combustion chamber with a very limited time for proper fuel-air mixing, which can contribute to higher emissions. This is different from the SI engine which the fuel-air mixture is evenly mixed with air before the combustion starts. Direct injection strategy is actually contributing to a higher level of emissions based on the previous report as mentioned by Veza et al. (2020), where even the SI employing the DI strategy is more likely to emit higher Particulate Matter (PM) emissions than the PFI-SI engine. In this case, the CI engine needs an additional mechanism in the air intake system that can effectively promote better fuel-air mixing before the combustion starts.

Apart from the fuel injection rate, the effective combustion in the conventional CI engine strongly relies on the swirl motion rotating about the cylinder axis inside the combustion chamber (Heywood, 2018). For effective combustion, the fuel injection and combustion system for this engine must be capable of achieving faster fuel-air mixing rates and reaching faster fuel-burning rates. Currently, the method to achieve a faster fuel-air mixing rate is mainly accomplished by the suitable design of inlet port geometries, cylinder heads and piston shapes for producing higher rates of fuel and air mixing with the

injected fuel sprays close to TDC (Heywood, 2018). Recently, the trend of a downsized engine has become more common in a passenger car in pursuit of a lower fuel consumption and emissions level with good engine efficiency. However, diesel engine downsizing faces more challenges due to the power output requirement.

The smaller engine operates at a higher engine speed than the larger engine, thus the fuel burning available time is less (Heywood, 2018). In a typical air intake system for a CI engine, other than evenly distributing the clean air supply, there is a limited mechanism to help for better fuel and air mixing in the cylinder. The existing design for most of the intake manifolds is still lacking in the swirl mechanism to aid the existing design of the intake geometry, valve, cylinder and piston for an extremely effective fuel-air mixing. Currently, the improvement in this area is still ongoing.

Numerous studies have been conducted on the use of swirl generators in the air intake system and many positive findings published by researchers on its effect on the combustion characteristics and emissions (Arjunraj et al., 2021; Bassiony et al., 2018; Hamid et al., 2020; Sadeq et al., 2019). Despite the reported findings, there are still improvements needed to unleash the full potential of this method. This is because the contribution of the swirl airflow to the fuel-air mixing has been limited by the existence of the air resistance caused by the swirl feature itself, which obstructs the airflow. In most cases, the design of available swirl generators either published, patented or commercialised still exhibits higher air resistance based on the presented geometry of the swirl mechanism.

Some disadvantages to the airflow characteristics can be seen in several patented swirl generators. S. Y. Kim (2021) patented an intake and exhaust swirling device for an internal combustion engine to accelerate or increase air swirling as shown in Figure 1.6. The device is used as a part of the air intake system between the air cleaner and the intake manifold to reduce the fluid resistance and increase the amount of fluid to improve the output. The device can help to promote better fuel-air mixing through the swirl induced, however, the internal wings installed will limit the airflow to the intake manifold due to the obstruction of the internal structure.

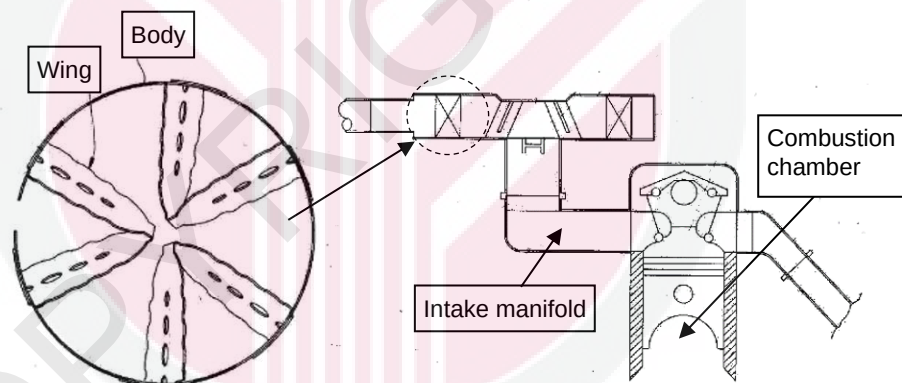


Figure 1.6: Patented design of a swirl air guiding structure
(Source: S. Y. Kim, 2021)

The position of the swirl generators before the intake manifold also adds to the existing pressure drop within the intake manifold which may compromise the actual amount of air supposed to receive by the engine. These are unfavourable conditions for the air intake system to have an increased pressure difference as described by Mamat (2009). Apart from the air resistance caused by the swirl generator itself, the effect of wall friction within

the intake manifold will cause a further reduction in the effectiveness of the swirl generated due to air dissipation before it reaches the intake port. This air resistance would cause a further pressure drop in the intake system that affects the volumetric efficiency (Mamat, 2009).

The air dissipation due to the internal structure may also limit the formation of swirling air. The swirl decay in the air intake system may be possible due to the effect of the wall friction and energy dissipation by the swirl structure as explained by Abo-Elfadl and Abd El-Sabor Mohamed (2018). In their study, the swirl produced in the cylinder was decayed due to the cylinder wall friction and the energy dissipation of the charge. This indicates the disadvantage of the swirl generator with an internal structure. The more complex the structure within the flow, there will be more wall friction that may increase the energy dissipation. Higher energy dissipation causes more swirl decay which means there will be less effective swirl airflow.

Similar to the variable swirl by S. Y. Kim (2021), the problem of additional obstacles to the airflow, air dissipation by the inner wall structures and pressure drop due to air resistance can be seen in the device patented by W. Zhang et al. (2018) and Chang (2010). Pressure drop in the intake system is very dependent on the flow resistance in the system, apart from the other engine parameters as mentioned by Mamat (2009). The additional swirl structures shown by these patents, although it was claimed to be able to increase the guiding effectiveness to form air vortex current, create an additional obstacle to the airflow. This would greatly affect the volumetric efficiency due to further reduction in the mass of airflow to the throttle body

and engine cylinder. Since the mass of airflow is directly proportional to the air density and the volumetric efficiency by referring to the relationship of the standard engine equations in numerous sources (Garcia, 2013; Heywood, 2018), a lower mass of airflow may further reduce the engine's volumetric efficiency.

There are more patented swirl generators that show similar disadvantages to the airflow characteristics as listed in Chapter 2. The majority of the devices are mainly focused on the swirl formation rather than optimising the swirl structure to minimise the impact of the flow obstruction. Higher air resistance caused by an obstruction to the airflow can reduce the amount of air volume required for complete combustion in one cycle of engine combustion. Heywood (2018) also, mention about the higher swirl during the intake process that may further reduce the volumetric efficiency. The reduction of volumetric efficiency due to the swirl is caused by the airflow restriction and pressure drop by the swirl ports, airflow disruption due to excessive turbulence, and increased heat transfer by the intense formation of the swirl (Heywood, 2018). The application of the swirl requires a comprehensive study in developing the swirl device that may sufficiently form a swirl airflow with the minimum air resistance in the swirl design while having the ram effect as an added advantage.

The application of ram air intake is one of the solutions for performance improvement in a vehicle regardless of the type of engine used. The benefit associated with ram air intake is usually the increased air density to improve combustion and volumetric efficiency. The improvement of these efficiencies may improve the power output of the engine as desired. The impact of the ram

air intake on engine performance has been investigated in some studies which resulted in higher volumetric efficiency, improved peak pressure, more efficient fuel-burning process, and increased torque and power (J. W. Lim et al., 2020). However, although the ram air intake is always associated with an improved performance output, there are some disadvantages in the engine behaviour which is indirectly caused by the ram effect.

Increased air density caused by the ram effect may increase the combustion temperature due to more fuel burning. The higher rate of fuel burning is due to the increased available oxygen in the cylinder as investigated by Baskar & Senthilkumar (2016). The higher peak cylinder temperature can lead to an increase in nitrogen oxides (NO_x) formation (Baskar & Senthilkumar, 2016; Heywood, 2018), thus increasing the emissions level. The increased intake pressure or pump work due to the ram effect also increases the knock tendency (J. W. Lim et al., 2020). This is because higher intake pressure causes higher output pressure in the engine. Thus, a suitable design of the intake manifold size, shape and structure for better airflow and fuel-air mixing should be further studied to avoid the knocking possibility.

Considering that the limitations of the ram and swirl are still there, hindering to achievement of better operations, thus the combination of the ram and swirl is essential to overcome their constraints and help to accomplish a highly effective and wider range of CI engine operations. Therefore, this study focuses on the ram-swirled effect of a novel helical air intake device design to investigate its effect on the performance, combustion behaviour and emissions level. It is hypothesized that the ram-swirled air intake produced by the novel

helical air intake device with the selected range of blade revolution, draft angle and length can promote better fuel-air mixing in the cylinder, thus improving the combustion behaviour, performance output and emissions level.

1.3 Objectives of the Research

The main goal of the study is to investigate the ram effects of the helical air intake manifold on the performance of a CI engine. The main goal can be achieved through the specific objectives as follows: -

1. To determine the parameters of a novel helical intake device that improves the flow characteristics of an air intake into the CI engines.
2. To identify the effect of the helical air intake device parameters on the swirl intensity through CFD simulation.
3. To examine the ram effect of the helical air intake manifold on the engine performance and combustion characteristics experimentally.
4. To evaluate the improvement on the emissions level of the CI engine.

1.4 Scope of the Research and Limitation

This research emphasizes on the strategies of the fuel-air mixing in improving the engine performance, combustion characteristics and exhaust emissions level. As part of the research scope, a novel swirl air intake device is designed to observe the effect of the swirl formation on the CI diesel engine behaviour.

The details of the boundaries and limitations of the study are as follows:

- In the design and simulation stage, both 3D modelling and CFD simulation are conducted using SolidWorks® and Ansys® Workbench™ software respectively.
- The parametric simulation is carried out according to the General Full Factorial approach with the aid of ANOVA analysis to identify the optimum swirl air intake parameters and study the effect of varying the level of the parameters within the specified range.
- In the General Full Factorial approach, the simulation of the parametric model involves two control variables or factors with seven (7) levels. The two selected factors are the device's 'draft angle' and 'length'.
- The research is carried out using the engine dynamometer facility at the Thermodynamics laboratory located in the Faculty of Engineering, Universiti Putra Malaysia, Serdang, Selangor, Malaysia.
- An eddy current engine dynamometer (up to 10 kW capacity) is used to collect the data on the engine brake torque and power.
- The engine used is a single-cylinder, four-stroke diesel engine (direct injection), air-cooled type Yanmar L48N model with a specially fabricated fixture for the assembly of the swirl air intake device.
- The engine is tested at full load conditions which covers the speed of 1800 to 3600 rpm according to the designated experimental parameters based on the Taguchi Orthogonal Array design. The test is performed

at ambient room temperature (27 °C) with an intake pressure of 1 atm. The engine is tested using pure or neat diesel fuel without mixing with any other blends of fuel.

- In the Taguchi methodology approach, the experiment involves two control variables or factors with three (3) levels. The two selected factors are the 'IM Helical Revolution' and 'IM Position' with the brake power as the studied response. ANOVA is used to analyse the significance of the studied factors to the response.
- The test conducted in this research only involves the 'engine level' test using the helical air intake device prototype which is made with ABS material. A road driving test is not within the scope of the research, thus no 'vehicle level' evaluation is being conducted.
- Exhaust gas emission level (CO, CO₂, UHC, NO_x, and O₂) is also measured to study the impact of the swirl air intake manifold on emission characteristics. The evaluation continues with the verification of the swirl air intake parametric model to confirm the engine performance, combustion and exhaust gas emissions level.
- For combustion behaviour, the in-cylinder pressure data is investigated to observe the trend in the maximum in-cylinder pressure, heat release rate, and pressure rise rate.

- The performance data measurement includes brake power, brake torque, brake specific fuel consumption, brake thermal efficiency, brake mean effective pressure and air-fuel ratio.

1.5 Project Significance and Contribution

This research is highly significant in identifying the optimum swirl air intake parameters for better engine performance and emissions. The study on the novel design of the swirl air intake manifold device may discover a new potential in the swirl mechanism that is effective in improving diesel engine operation, particularly in combustion behaviour, performance and exhaust emissions. This is highly essential in contributing to more advanced CI engine development in the future. Both the transportation and industrial sectors will also find a lot of benefits from the research findings in overcoming the recent increasing trend of air pollutants. Based on the previous report mentioned earlier, the air pollutant such as CO emissions increased in the year 2022 compared to 2021 with approximately 96% of the total CO emission in Malaysia. This statistic is being contributed by the transportation and industrial sectors. Apart from the emissions, the transportation and industrial sectors will also benefit from improved performance and better efficiency based on the research outcome. The increase of the performance parameters such as the maximum brake power and torque indicates the opportunity to have wider operations of diesel engines above than the current capability which is limited at low to medium-speed range.

This invention solves the problem of airflow restriction and friction that produce excessive air resistance to the intake air at the inlet manifold. The excessive air resistance is causing more pressure drop in the intake system which further affects the volumetric efficiency of the engine. The excessive air resistance happens due to the obstruction of internal structures introduced in the prior art. The greater air mass flow (which also means an increase in air-charged density) can be achieved without the use of an additional air-forced induction system.

The air swirl with the rapid air intake flow into the inlet port will increase the mixing rate of the fuel and air inside the cylinder. The increased fuel and air mixing rate then causes more complete combustion as well as reducing unburned hydrocarbon (UHC) and other exhaust emissions. To take the advantage of ram pressure effect, a larger draft angle introduced with the blade geometries will greatly increase the air intake mass and density. This is caused by the relative bulk motion of the air through the device, which means the engine will benefit from the ram-swirled air intake by the increase of both fuel-air mixing rate (due to air swirl) and mass of airflow (due to ram pressure).

This device can also be used with hydrogen fuel in the internal combustion engine. Hydrogen is the future sustainable fuel which some of the companies have already turned into hydrogen application in the development of new IC engines. Although the trend of electric vehicles has become more popular in recent years, however, the unveils of hydrogen engines by two giant automakers such as Honda and General Motors in early 2024 has suggested

a potential shift from electric vehicles to hydrogen engines in future (Leussink, 2023; Tingwall, 2024).

Not to mention the prospect of a green hydrogen economy as ventured by the State Government of Sarawak in 2019 (Sarawak Energy, 2020). The swirl air intake device may have a significant contribution to the low-carbon initiatives since it is useful for hydrogen mixing to make the mixing fuel-air mixture more homogeneous. As one of the prospects in the energy sector due to the cleaner properties, industries may shift towards hydrogen fuel. By the time of implementation, the device will be ready to be part of the energy-efficient technology for both transportation and heavy-industry sectors.

1.6 Thesis Outline

This thesis aims to enlighten the reader with the study of the ram-swirled air intake flow conditions and their contributions and effects on the performance, combustion and emissions of a diesel engine. The aims are also to understand the development and analysis of the novel swirl air intake design parameters which effectively contribute to swirl airflow formation in a diesel engine air intake system. This thesis is divided into six chapters starting with Chapter 1 which consists of the background of the study, introduction, problem statement, research objectives, research scope, project significance and contribution. This chapter shall provide an overview of the current background and challenges in a CI engine that motivates and drives the current study.

Chapter 2 reviews related subject matters such as the performance and the technological background of the current intake manifold design, patented and

commercialized swirl air intake device. This section also reviews typical air intake manifold performance in a diesel engine which includes the combustion and performance characteristics such as the brake power, torque, brake thermal efficiency, brake specific fuel consumption, brake mean effective pressure, exhaust gas emissions and engine knocking and vibration characteristics. The overview of the strategies for homogeneous mixture in CI engine is also provided with a brief review of the previous numerical study involved in the CI engine fluid flow.

In Chapter 3, the design, development, simulation and experiment of the swirl air intake device are explained in more detail. This includes the experimentation setting, equipment, and facilities involved in the engine performance test. This chapter thoroughly explained the characteristics and parameters involved in the base-swirl and parametric-swirl studies. The systematic and comprehensive method to run the simulation and experiment to explore the main effects through the statistical approach is also presented.

Chapter 4 outlines the findings and analysis of CFD simulation of the base-swirl and parametric-swirl air intake models. This includes the statistical analysis of the parametric study which employs the General Full Factorial methodology. Through the graphical diagram and ANOVA test, the Full Factorial simulation results are analysed to identify the factor significant to the swirl intensity of the swirl air intake model. The final part of this chapter presents the CFD validation against the experimental data and verification of the parametric study against the base-swirl air intake simulation.

In Chapter 5, the findings and analysis of engine performance, combustion and emissions based on the experimental run are discussed in detail. The engine operated with the base-swirl air intake device is prepared following the Taguchi Orthogonal Array design as presented in Chapter 3. The data analysis of this experiment is discussed comprehensively with the presentation of 'Main Effects and Interaction Plot for Means' and the response estimation of 'Means and Signal-to-Noise (SN) ratio' based on the Taguchi method. This includes the ANOVA test to provide the significance value of each factor based on the actual run. The performance and emissions analysis of the parametric-swirl air intake device verification run are also presented. The final part of this chapter presents the comparison of engine vibration characteristics of the swirl air intake device against the normal intake.

Finally, Chapter 6 concludes the results based on the simulation and experimental work. Some critical findings based on the activities are also addressed. At the end of this chapter, several recommendations are proposed for future work consideration and improvement involving the swirl air intake applications.

CHAPTER 2

LITERATURE REVIEW

2.1 Introduction

The performance of a Compression Ignition (CI) engine is very much related to the combustion characteristics. In a conventional four-stroke CI engine, the fuel and air are inhomogeneously mixed in the combustion chamber which is then compressed after the intake stroke. Under normal operating conditions, the combustion is initiated towards the end of the compression stroke in which the pressure rise is rapid and the rate of heat release is maximum. This stage is known as the period of rapid combustion or uncontrolled combustion.

The combustion continues to the next stage of controlled combustion where it is assumed to be ended at the maximum in-cylinder pressure and temperature. The combustion does not cease with the completion of the injection process. The unburnt and partially burnt fuel particles left in the combustion chamber will start to burn as soon as they come into contact with the oxygen and it continues for a certain duration. This duration is called the after-burning period. The rate of after-burning depends on the velocity diffusion and turbulent mixing of unburnt and partially burnt fuel with air. The after-burning period is relatively critical to the emissions level as they may not be efficient enough to let all the unburnt or partially burnt fuel particles settle before the new engine cycle begins. The presence of unburnt fuel particles after the end of the combustion cycle will appear as the UHC emission (Unburned Hydrocarbon) along with the other elements such as CO_2 , CO , and NO_x .

The main concern of the emissions levels as well as the performance of the CI engine is due to the effect of the quality of fuel-air mixing and the presence of turbulence flow in the combustion chamber after the intake stroke. Intake air without sufficient turbulence flow will not help to achieve a homogeneous fuel-air mixture in a shorter time and combustion without proper fuel-air mixing causes a lot of inefficiencies in the engine operation.

In improving the engine efficiencies and reducing the emissions, various strategies have been explored with the key focus being to observe the effect of a different setting or combination of fuel injections, ignition control and fluid flow mixing. While the fuel injection and ignition control may improve the fuel-air mixture through the combustion reactivity in the cylinder, the fluid flow mixing strategy, however, focuses on improving both external and in-cylinder flow characteristics to increase the mixing capability to promote fuel-air mixture homogeneity.

2.2 State of the Art of Current Intake Manifold in IC Engines

The fluid flow mixing strategy has attracted the attention of researchers due to its proven effectiveness in improving combustion processes and fuel-air mixing. This strategy has a lot of potential to improve engine performance and reduce emissions by implementing various designs on important components such as an air intake manifold for guiding and controlling the external airflow. For establishing the in-cylinder turbulence flow, an intake port shape, an intake valve, and a piston crown may have a different design to assist in creating the swirl and tumble motions. In comparing the performance and emissions of

current engines, this study, however, will focus on the design of an air intake system in an internal combustion engine.

Producing homogeneous mixtures to achieve near-complete combustion is the target of every engine manufacturer. In the current air intake systems, major improvements have been made to the arrangement and location of the intake pipe. The majority of the current vehicles have arranged their air duct far from the engine side. This is to minimize the effect of the hot air that surrounds the engine while it runs. The cold intake air provides denser air, which means more oxygen to be supplied to the engine cylinder. More oxygen will result in better combustion efficiency.

This air duct arrangement can be seen in most of any vehicle nowadays, among others to be named, Mercedes Benz C200 (petrol or diesel), Proton Preve (petrol only), Volkswagen Golf (petrol or diesel), Honda City (petrol or diesel) and Toyota Camry (petrol only). The layout for the air duct arrangement in the Mercedes Benz C200 can be seen in Figure 2.1. There are other methods commercially available to increase the performance and fuel efficiency. Methods such as cold air intake and ram air intake are among the popular performance choices. However, the cold air intake and short ram air intake methods are mainly considered the aftermarket replacement primarily for performance upgrades due to the elimination of the standard airbox component. In consideration of the overall NVH (Noise, Vibration and Harshness) level, comfort, reliability, cleanliness and maintenance cost of the engine, the existing air intake manifold is often installed with a standard airbox that restricts the laminar airflow to the engine, thereby limiting the actual

capability of the engine. The standard airbox still can be seen in modern vehicles such as in the Mercedes Benz C200 and also in other brands.

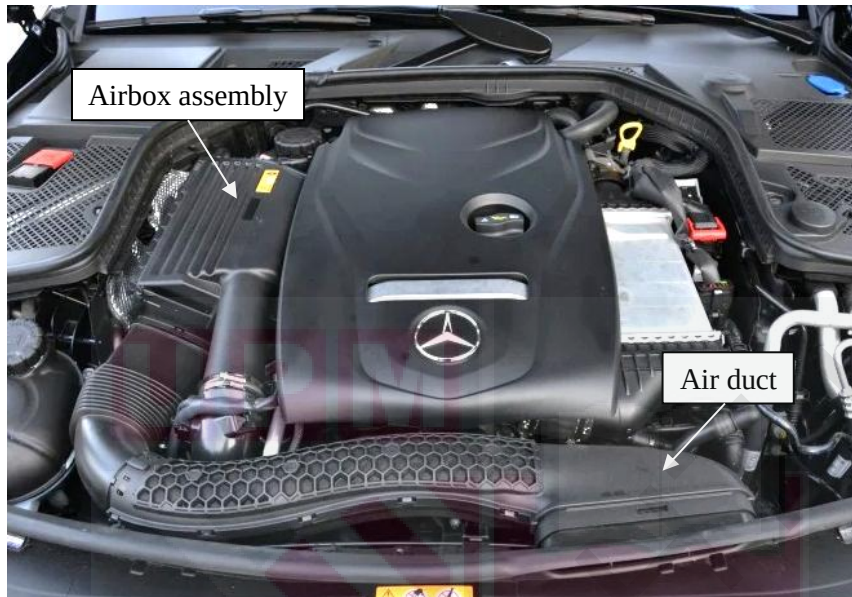


Figure 2.1: Mercedes Benz C200 air intake system
(Source: A. Lim, 2014)

To ensure optimum combustion efficiency in a standard engine, proper fuel-air mixing is therefore assisted by the use of the MAF sensor in a petrol engine to calculate the correct amount of fuel that matches the amount of incoming air. In the case of a diesel engine, the exhaust pipe will be equipped with an oxygen sensor to evaluate the amount of oxygen from the exhaust gas. The ECU will calculate a proper air-fuel ratio based on the oxygen content. This will help to achieve better homogeneous mixtures for a near-complete combustion.

In the other part of the air intake system, there are several air intake manifold designs used to optimize the flow of the intake air at both higher and lower engine speeds. The internal combustion engine manifold technology referred to as the Variable-Length Intake Manifold (VLIM), Variable Intake Manifold (VIM) or Variable Intake System (VIS) is vital to the internal combustion engine operation to optimize power and torque across the range of engine speed operation, as well as to help provide better fuel efficiency. The VLIM, VIM or VIS is operated by varying the length of the intake tract for optimized intake flow. An earlier patent issued for a variable length intake manifold was published by Daimler Benz AG in 1958 with the US Patent US2835235 as shown in Figure 2.2 (Gassmann, 1958).

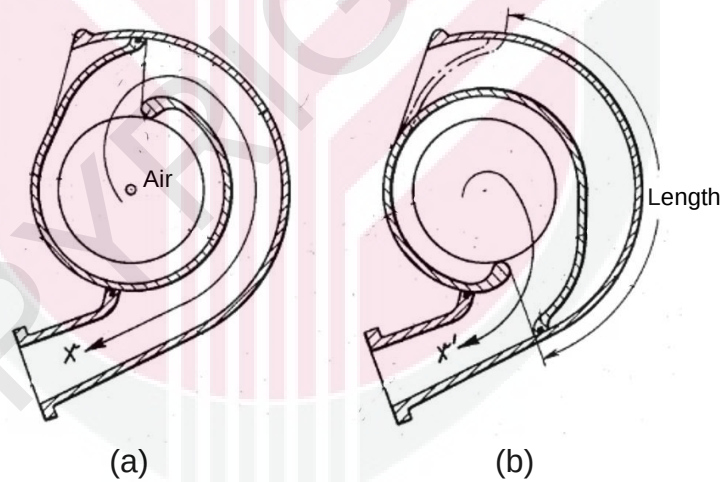


Figure 2.2: First patented design of a VLIM. (a) Cross-section of an intake pipe adjusted to the greatest length, (b) Cross-section of an intake pipe adjusted to the shortest length
(Source: Gassmann, 1958)

In maximising the volumetric efficiency through the entire rpm range, modern variable intake manifolds (VIM) nowadays utilise an ECU-controlled VIM to change the passage lengths that the intake air must pass through before

entering the cylinder head inlet port. Either vacuum or electrically operated, the transfer in the path lengths is carried out by the actuators (MS Rheinmetall Automotive, 2017). Figure 2.3 shows one of the recent VIM designs that has been used for a BMW V6 engine and other compatible engines such as Chrysler, Audi and Volkswagen V6 engines. Meanwhile, Figure 2.3 shows the condition in which the flap is open and closed to change the runner's length.

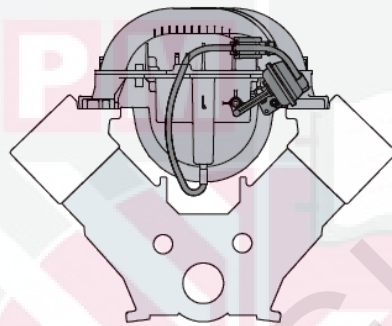


Figure 2.3: A modern variable intake manifold technology used in a V6 engine

(Source: MS Rheinmetall Automotive, 2017)

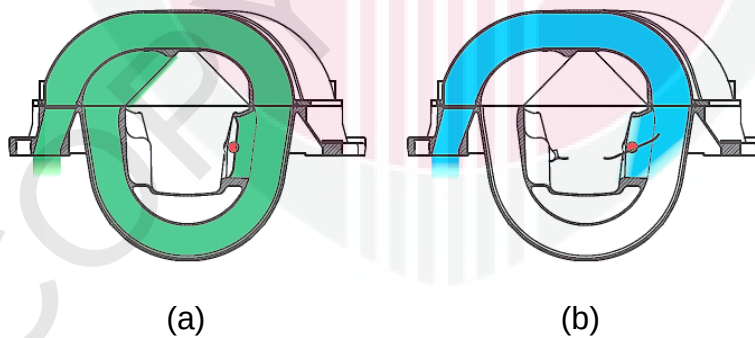


Figure 2.4: (a) Flap closed for greater length, (b) Flap open for shorter length

(Source: MS Rheinmetall Automotive, 2017)

The VLIM used to be controlled by a valve, that would open two different runners at various loads or engine speeds. At full engine load, the VLIM will work with a short runner to gain more power as shown in Figure 2.4 (b).

Meanwhile, at a lower load, it will change to a longer runner to achieve slower airflow to promote better mixing with fuel as shown in Figure 2.4 (a). The main effects of this variable-length mechanism will create beneficial air turbulence in the combustion chamber and a light pressuring effect which has further increased the engine output.

The use of the VIM or VLIM greatly affects the performance of an engine. Figure 2.5 shows the performance curve of the Mercedes-Benz V6 3.2 litre engine with VLIM as reported by MS Rheinmetall Automotive (2017). The change of the operation mode to the greater length of the inlet port facilitates high engine torque in the lower engine speed range. For a short intake manifold, the shorter inlet port allows higher performance at a higher range of speed because it allows the air to reach the inlet port much earlier than the longer inlet port. At higher speeds, there is less available time to fill the cylinder, thus shorter intake manifold helps to overcome this limitation (MS Rheinmetall Automotive, 2017).

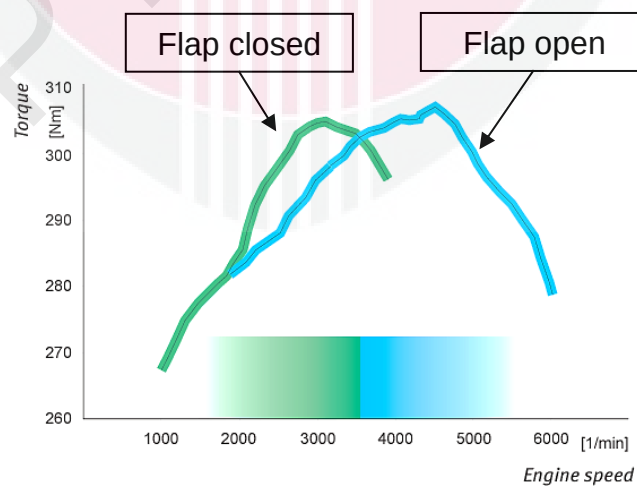


Figure 2.5: Performance curve (Torque) of the Mercedes-Benz V6 3.2 litre under two different flap positions
(Source: MS Rheinmetall Automotive, 2017).

Three (3) stages of VLIM are also introduced to broaden the torque curve compared to a typical two (2) stages of variable length (Wan, 2011). The effect of using the three stages of VLIM can be seen in Figure 2.6, where there are three different torque curves at different engine load conditions. This design was used in the Audi 4.1 litre V8 engines (Wan, 2011). However, as the 3-stage system is more complex than the 2-stage system, the 2-stage system is more common for most of the VLIM designs.

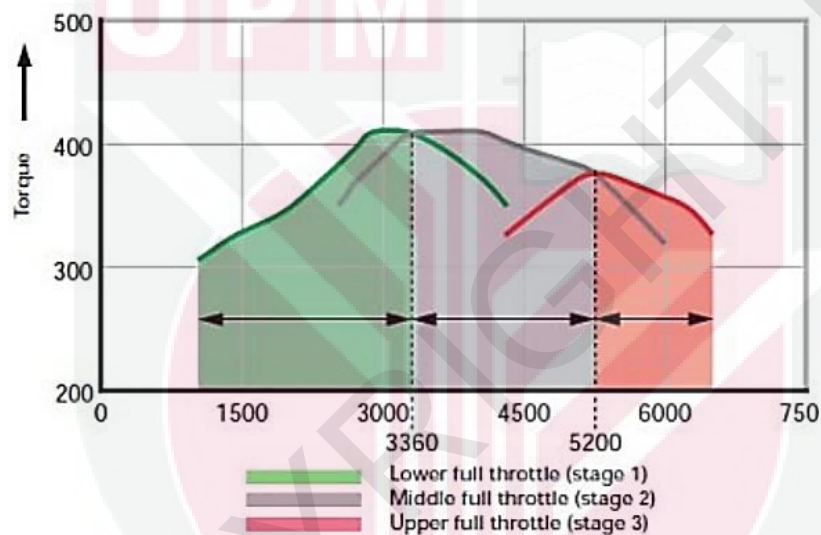


Figure 2.6: Performance curve (torque) of three (3) stages of VLIM
(Source: Wan, 2011)

For diesel engines, there is an electric drive module utilised for the operation of the swirl flaps when required (Premier AutoTrade, 2023). The combustion of a CI engine is mainly contributed by the fuel-air mixture homogeneity caused by the effect of the inducted air-charged flow. Figure 2.7 shows an OEM variable intake manifold design for a diesel engine with an intake control valve (ICV) for the swirl flaps operation.

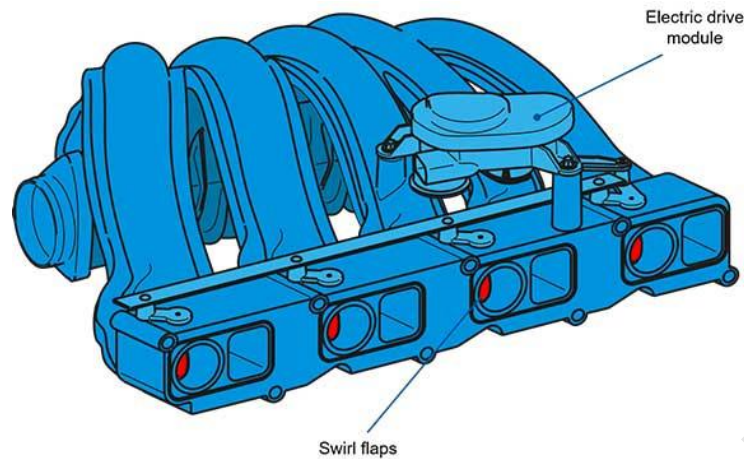


Figure 2.7: OEM variable intake manifolds with an ICV module
(Source: Premier AutoTrade, 2023)

Variable-length intake manifold technology has been used by many automobile manufacturers with different names. As example, but not limited to, the Mitsubishi Variable Induction Management (MVIM), Honda 2-Stage Variable Intake Manifold (VIM), Toyota Variable Induction System (T-VIS), Audi 3-Stage Variable Length Intake Manifold (VLIM), BMW Differential Variable Intake System, Volvo Variable Induction System (V-VIS), Mazda Variable Resonance Induction System (VRIS), Mercedes Benz M112 Variable Geometry Intake Manifold, Proton CPS Variable Intake Manifold (CPS-VIM) and Proton IAFM Variable Length Intake Manifold (IAFM-VLIM) (Edilan et al., 2014; Larsson et al., 1991; Lorio, 2019; Matsumoto & Ohata, 1986; Mitsubishi Motors, 2003; Motor Reviewer, 2022; MS Rheinmetall Automotive, 2017; Potul et al., 2014; Premier AutoTrade, 2023; Sachek, 2023; Soon, 2014). Each of these intake manifolds may have a different working mechanism but the main principle behind these variable-length intake manifolds is similar.

2.3 Patent and Commercial Swirl Air Intake Device

The intake manifold is by far the most suitable component to be continuously developed and improved since the inducted air has a direct effect on the volumetric efficiency and the engine performance. The greater the mass of the charge inducted, the higher the power produced. Volumetric efficiency is a measure of the effectiveness of an engine's induction process. Higher volumetric efficiency means the induction process is highly effective and this may increase the engine power. However, to have a near-complete combustion, the fuel-air mixture must be homogeneous and this requires a better fuel flow mixing strategy.

One of the methods, as mentioned in the previous section, is the external formation of swirl airflow generated by a certain mechanism in the intake manifold, for example, the swirl-generating device installed within the intake system. This device has been thoroughly studied by researchers with some of the variants may successfully find the market as an aftermarket part. The use of these aftermarket swirl air generating devices in internal combustion engines is mainly applied in expectation to provide swirl air for better fuel-air mixing during the premixed phase. Table 2.1 shows some of the list of patented, published or commercialized swirl air-generating devices that are substantial to the field. The detailed description of these inventions is also reviewed and presented in this section.

Table 2.1: Patented or commercialized swirl air generating device

No	Reference	Patent No./DOI	Invention/Product Name
1	Selirio et al. (2022)	US11224846	Static mixer for fluid flow in a pipeline
2	S. Y. Kim (2021)	WO2021150025A1	Intake and exhaust swirling device for IC engine
3	Y. H. Kim (2021)	KR20210103189A	Device for decreasing flowing resistance of intake air of the engine
4	T. W. Kim & Yoon (2021)	KR102250955B1	Velocity increase device of intake and exhaust pipe for automobile
5	Arjunraj et al. (2021)	10.1007/s10973-021-10817-z	Novel intake manifold
6	J. Y. Kim (2018)	KR101921023B1	Swirl generator in air intake pipe and exhaust pipe for car
7	Zhang et al. (2018)	CN108757153 (A)	Variable swirl guiding supercharger
8	Liu et al. (2018)	CN207111268(U)	Swirler structure at air cleaner inlet end
9	Kimura & Shibata (2017)	JP6252774B2	Intake device
10	Anderson (2016)	US9228542B2	Swirl vane air duct cuff assembly and method of manufacture
11	Chen et al. (2016)	CN205605339UA	Variable vortex air intake manifold structure of engine
12	Saad & Bari (2015)	10.11113/jt.v75.5217	Guide Vane Design (GVD)
13	Il & Seob (2013)	KR20130075874	A swirl apparatus for combustion engine
14	Yao (2013)	US2013255616A1	Air-Guiding Device for Vehicles
15	Chang (2011)	US2011011370A1	Air Pressure Vortex Generator Structure for an IC Engine
16	Andreas & Josef (2011)	JP2011501024A	IC engine having intake system
17	Chang (2010)	GB2461995A	Air Inlet Device with Swirl Plates
18	Deok (2004)	KR20040069119A	Duct structure of super charger for automobiles
19	Woo (2004)	KR20040096709A	Intake pipe for diesel engine
20	Wijaya (2003)	US2003221662A1	Air flow-twisting device on an air inlet system of IC engine

Some of the swirl air intake devices are available as aftermarket replacement parts. The purpose is similar to the cold air intake which prioritizes the performance upgrade and tuning from a factory standard system. Among the examples of commercial swirl air intake devices that can be easily found in the open market are the 'Flex-Force Performance Intake Kit' (SpitFire Tuning, 2023) and the 'Air Suction-Turbo' (Surbo.net, 2022) as shown in Figure 2.8. These products are designed to work with the air intake system of an

automobile. Based on the statement, these companies have claimed that their kit or device is capable of increasing the engine performance, improving throttle response, inducing more air, creating an air vortex and saving fuel (SpitFire Tuning, 2023; Surbo.net, 2022). Despite the claim, the product's effectiveness cannot be independently verified or proven by the author. However, the expected outcome, concept and basic function of the product are very similar to that of the patented design.

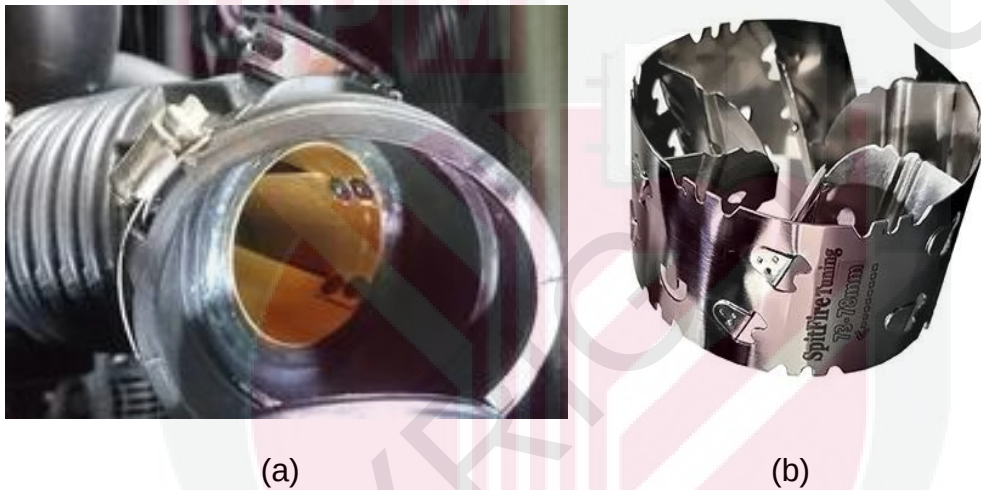


Figure 2.8: Aftermarket swirl air intake device. (a) Air-Suction Turbo, (b) Air Intake Performance Kit
(Source: Surbo.net, 2022; SpitFire Tuning, 2023)

The Flex-Force intake kit, as an example, has similarities to the design patented by Zhang et al. (2018) in terms of the tubular air-guiding pipe and the air-guiding structure. Zhang et al. (2018) patented a tubular air-guiding pipe, an airflow baffle plate and a swirl air-guiding structure to form a variable swirl. The swirl air-guiding structure comprises multiple flow-guiding vanes which are connected to form a radial central symmetry structure as shown in Figure 2.9 (a). The use of the internal guiding vanes with the baffle plate will allow the air-flow amount to be varied at different engine speeds. S. Y. Kim (2021) also has

patented almost a similar concept of a swirl guiding pipe as shown in Figure 2.9 (b) with an intake and exhaust swirling device for an IC engine that can accelerate or increase air swirling in addition to the reduction of the surface resistance of prior art, and increase flow velocity and flow rate.

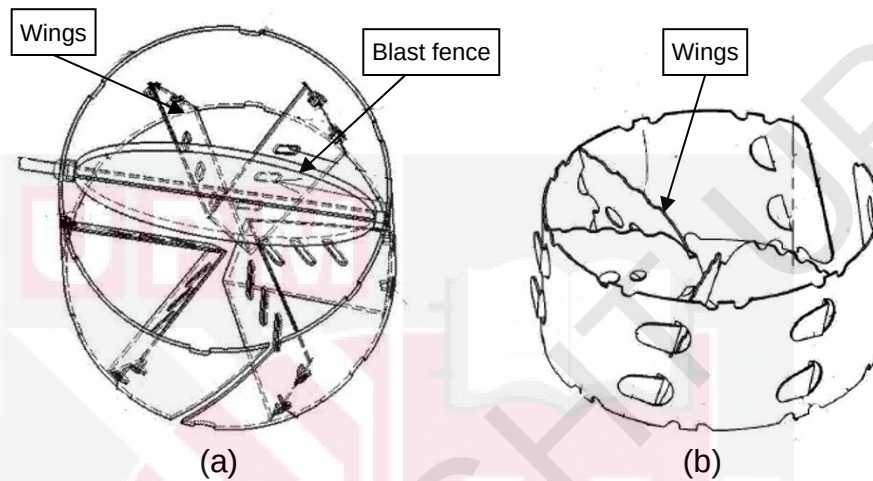


Figure 2.9: Patented design of a swirl guiding pipe. (a) Swirl air guiding structure (W. Zhang et al., 2018), (b) Intake and exhaust swirling device (Source: W. Zhang et al., 2018;S. Y. Kim, 2021)

The swirling device by (S. Y. Kim, 2021) has a main body and a plurality of wings installed to be radially inclined in the main body. The top and bottom of each wing have a bent portion that is bent in the same direction as the inclined direction of the wing to generate an eddy in the bent portion vicinity. This device is installed inside an air intake system between the air cleaner and the intake manifold to reduce the fluid resistance and increase the amount of fluid to improve energy efficiency by complete combustion and improving the output (S. Y. Kim, 2021).

As shown in Figure 2.10, another patented air intake device was designed by Y. H. Kim (2021) to be used in the intake pipe to reduce the intake airflow resistance of an engine by concentrating the airflow to the centre. The device comprises inner hollow rings with a plurality of turning guide wings and a tapered body shape to increase the concentration of airflow to the centre. The inner diameter gradually decreases so that the first entrance on one side thereof is smaller than the first exit on the other side. This part can be easily mounted in the intake system by inserting it into an intake pipe connected to the internal combustion engine.

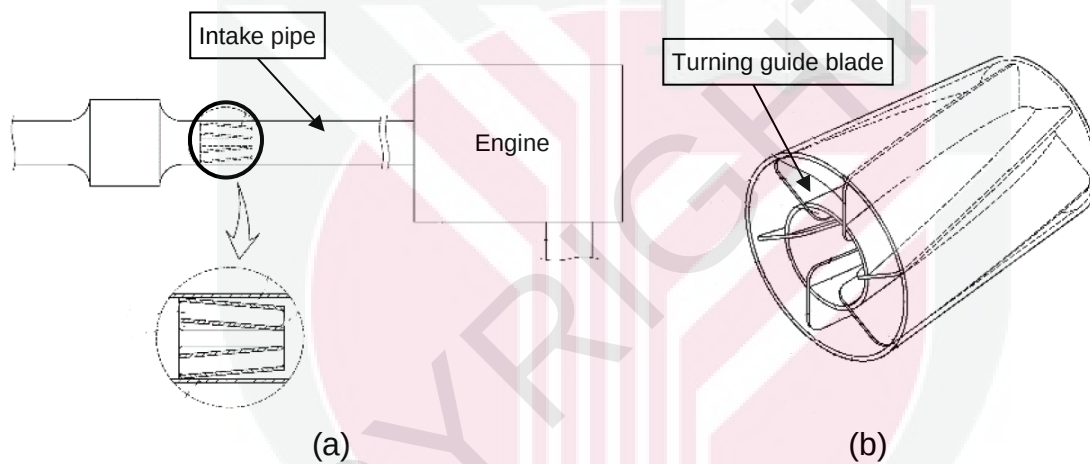


Figure 2.10: Patented design of an air intake device. (a) Pipe assembly schematic diagram, (b) An air intake device for IC engine
(Source: Y. H. Kim, 2021)

Figure 2.11 is a turbulence-forming unit, which is part of an air intake system that forms a swirl in a power unit such as an internal combustion engine or a gas turbine engine to achieve optimal combustion. The device consists of a cylindrical body having a diameter to be inserted into an intake passage of any engine. The impeller-type turbulent guide surface is formed in an inner diameter of the body which is connected to the central shaft of a venturi tube. An inclined surface is formed around the upper-end opening of the body.

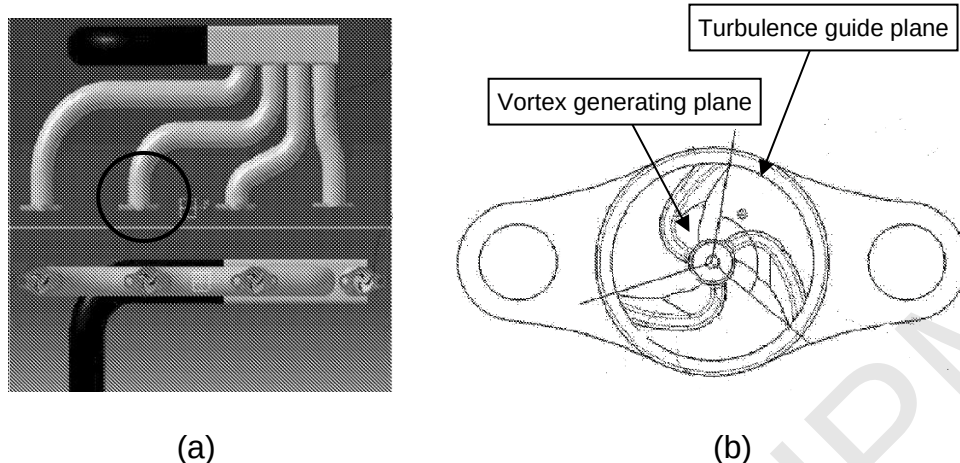


Figure 2.11: A turbulence forming unit for IC engine. (a) Schematic diagram of the air intake manifold and (b) An impeller-type turbulent guide device
(Source: Il & Seob, 2013)

This device is applied to intake systems of various types of internal combustion engines, where it can be separately installed between the existing intake manifold and the intake port. The main purpose of this invention is to improve the combustion efficiency of the engine and help to achieve complete combustion.

Figure 2.12 shows an air guiding device, which is a part of the air intake system that efficiently mixes air with fuel to improve combustion to reduce the deposit of carbon in the internal combustors of vehicles. The device is a tubular element made of a flexible metal sheet and includes a positioning section and an air-guiding section. The position section is fit in the intake manifold. The air guiding section includes helical blades that converge as they extend from the end of the positioning section.

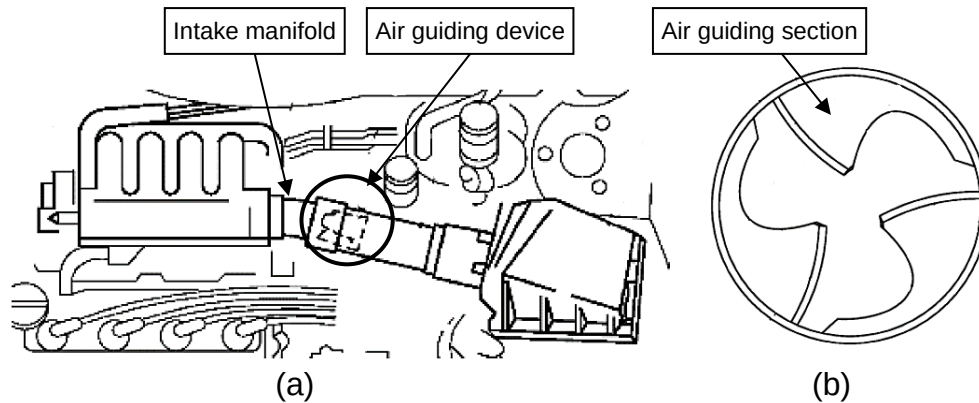


Figure 2.12: Air pressure vortex generator. (a) Schematic diagram of the device assembled before the air intake manifold (b) A converged helical blade feature

(Source: Yao, 2013)

This device is used in an intake manifold arranged between an internal combustor and an air-filtering device. The main purpose is to develop turbulence airflow for better combustion with an easily made, inexpensive, environmentally friendly and turbulence-efficient air-guiding device for vehicles.

Figure 2.13 presents the air pressure vortex generator structures which are installed within an intake manifold between an air filter and an internal combustion engine. The structure comprises a tubular body with one end which has a plurality of guide plates inwardly or outwardly bent from the tube wall. The guide plates also are arranged to be equally spaced and are inclined in the same direction. This is another invention that also uses the guiding structure to naturally form a plurality of spiral air currents at an accelerated speed, where the main purpose is to achieve a uniform mixing of air and fuel for better combustion.

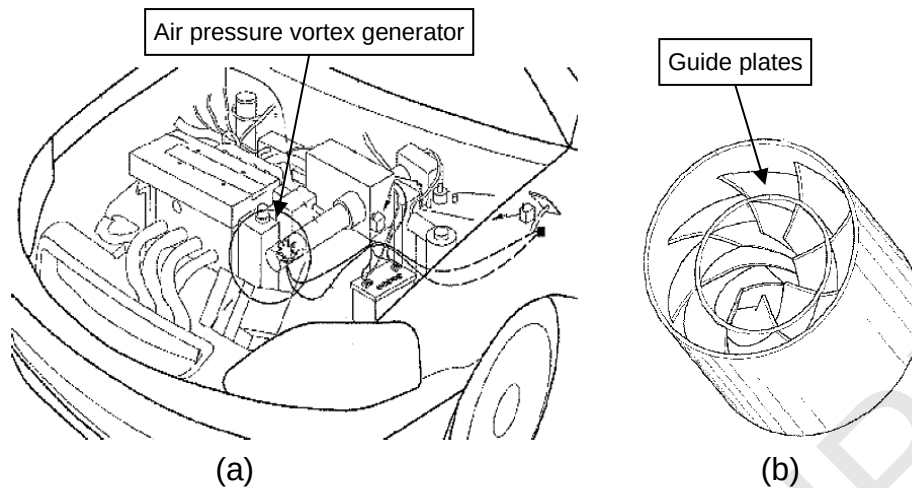


Figure 2.13: Air pressure vortex generator. (a) Schematic diagram of the device assembled before the air filter (b) A air pressure vortex generator
(Source: Chang, 2011)

Figure 2.14 shows three variations of the intake manifold designed in a helical shape to produce higher turbulence in the combustion chamber. The helical intake manifold pipe is provided with 30° in the output angle that can create high disturbance and a solid swirl of the initiated airflows. This intake manifold needs to be installed near the engine intake port, where the main purpose is to create high turbulence and solid swirls in the combustion chamber.

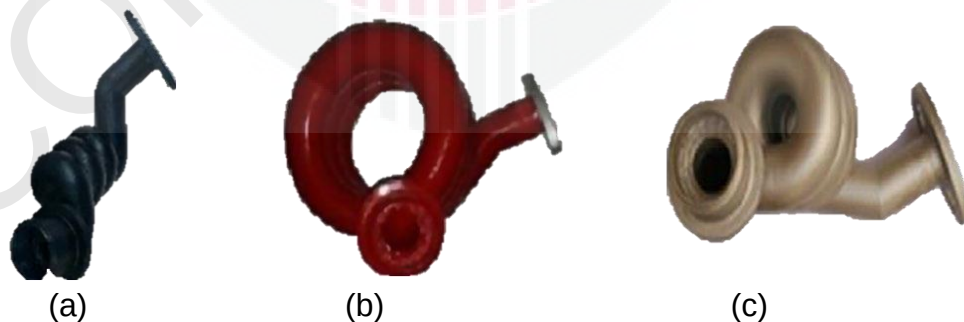


Figure 2.14: Three optimized helical-spiral manifolds with different helical diameter variations. (a) 1D helical diameter, (b) 2D helical diameter, (c) 3D helical diameter
(Source: Arjunraj et al., 2021; Sadeq et al., 2019)

The Guide Vane Swirl and Tumble Device (GVSTD) shown in Figure 2.15 is part of the air intake system that will develop an organized in-cylinder turbulent flow and break up the heavier molecules to mix with air. The device consists of four simple fins imposed inside the intake system with a twist angle of 35° and the height is varied from 25% to 75% of the intake runner radius. This device is used as a part of the air intake system and installed in between the intake runner and intake port, where the main purpose is to improve the fuel-air mixing for better combustion.

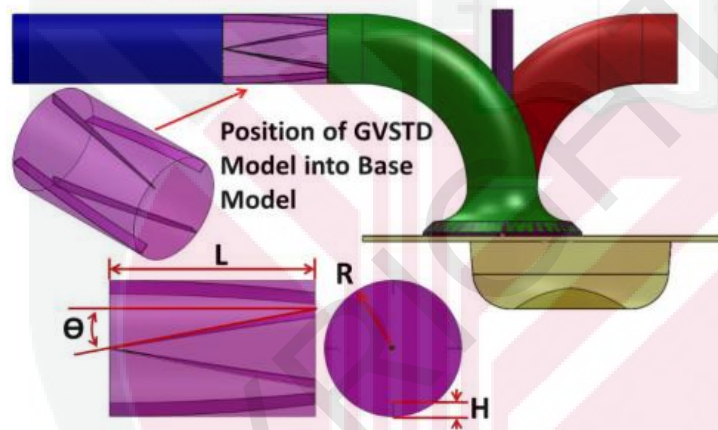


Figure 2.15: GVSTD consists of four simple fins with a twist angle with various heights
(Source: Saad & Bari, 2015)

2.4 Effect of Swirl Air Intake on the CI Combustion and Performance Characteristics

The combustion efficiency of an internal combustion engine (ICE) is strongly dependent upon fluid flow through the chamber (Payri et al., 2004). The fluid flow in the engine affects the output due to changes in fluid structures in the engine cylinder and finally changes in the combustion (Zaidi, 2019). Secondary flow such as turbulence, swirls and tumble are important for adequate mixing of air-fuel, appropriate distribution of fluid in the chamber as well as for

increasing the efficiency. The flow in the inlet manifold is one particular flow that has a large effect on engine combustion (Zaidi, 2019). A better swirling motion of fluid or strong vertical structures in the inlet manifold can lead to an efficient mixing of fuel and air and better combustion (Giannakopoulos et al., 2017). The swirl air which initially developed in the inlet manifold can be characterized by the swirl number based on several definitions that will be explained in the next section.

2.4.1 In-Cylinder Pressure and Temperature

The performance of a CI engine operated with swirl air intake is expected to be affected by the increase of the fuel-air mixing rate before combustion. The variation in heat release rate due to the change in the fuel-air mixing rate will also affect the in-cylinder peak pressure and temperature. Typically, higher loads and richer mixtures will also produce higher peak pressures. Based on the report by Arjunraj et al. (2021) and Sadeq et al. (2019), the peak pressure increases with increasing the engine load due to the increased amount of the injected fuel, thus resulting in higher in-cylinder pressure and temperature at a higher engine load. Other than the engine load and speed, the pressure also depends on the energy of the fuel, intake temperature and pressure.

The effect of swirl air intake on the in-cylinder pressure and temperature needs to be properly investigated with consideration of turbulence. From the previous investigations, swirl intensity will always be associated with turbulence kinetic energy (TKE) when studying its effect on the peak pressure (Arjunraj et al., 2021; Bassiony et al., 2018; Hamid et al., 2020; Sadeq et al., 2019). The intake

manifold that generates a strong swirl and a maximum TKE inside the cylinder may lead to more complete combustion as this reduces the physical ignition delay which includes the atomisation, penetration, evaporation and mixing of the fuel with air. Reduction in the ignition delay, which is due to the enhanced fuel-air mixing may cause a reduction to the maximum pressure. This is because a shorter ignition delay reduces the amount of burnt fuel during the premixed phase of the combustion process.

In terms of the turbulence level, higher swirl intensity does not mean the TKE will be higher due to the variation in the fluid flow characteristics. Depending on the shape, structure and length of the device, the flow characteristics such as the turbulent drag and diffusion may be affected by the confined inflow condition and this will reduce the TKE further, even with the higher swirl intensity (Arjunraj et al., 2021; Sadeq et al., 2019). The effect of lower TKE inside the cylinder by changing the swirl air intake device, for example, may cause a different peak pressure level for the same engine. For such conditions, the effect of fuel properties such as high viscosity and low cetane number that affect the fuel atomisation and droplet size becomes dominant in extending the ignition delay and producing high peak pressure (Sadeq et al., 2019). It should be mentioned that the turbulence energy level in the cylinder may further change with different designs of piston-bowl, combustion chamber and other parameters.

Arjunraj et al. (2021) investigated the effect of different sizes of intake manifold helical pipe diameter that can generate different intensities of swirl air (Intake manifold type 1D, 2D and 3D) on a CI engine. For the testing with a neat diesel

fuel, intake manifold 1D results in a lower peak pressure of 6.2% as compared to the standard intake manifold, but the peak pressure for intake manifold 2D and 3D have higher peak pressure by 3.4% and 6.3% respectively. Intake manifold 1D has a strong swirl intensity with high TKE, while intake manifold 2D and 3D, have stronger swirl intensity but with lower TKE based on the previous CFD simulation work (Bassiony et al., 2018; Sadeq et al., 2019). These findings show the importance of both strong swirl and turbulence intensity in achieving better combustion efficiency.

The effect of swirl air intake on the maximum temperature in the cylinder, by far, is not fully reported by the researchers. However, according to the gas law, pressure is directly proportional to the absolute temperature when the volume is constant (Oxford Reference, 2023). Therefore, it is expected with the decrease in maximum pressure, the maximum temperature inside the cylinder will also decrease.

2.4.2 Brake Power and Torque

In terms of engine performance, there is much evidence of the increase in overall performance for an engine induced with swirl air intake (Hairuddin & Shafanejad, 2018; Naga Deepthi & Govinda Rajulu, 2019; Sharma et al., 2017). As reported by Hairuddin and Shafanejad (2018) and Sharma et al. (2017), there is about a 1% to 6% increase in the performance output as observed in their investigations. The optimized swirl increases the volumetric efficiency and consequently improves the combustion characteristics. Figure 2.16 shows one of the improvements in the brake power and torque after the

modification of the cylinder head intake port design for the optimised swirl intensity and flow coefficient.

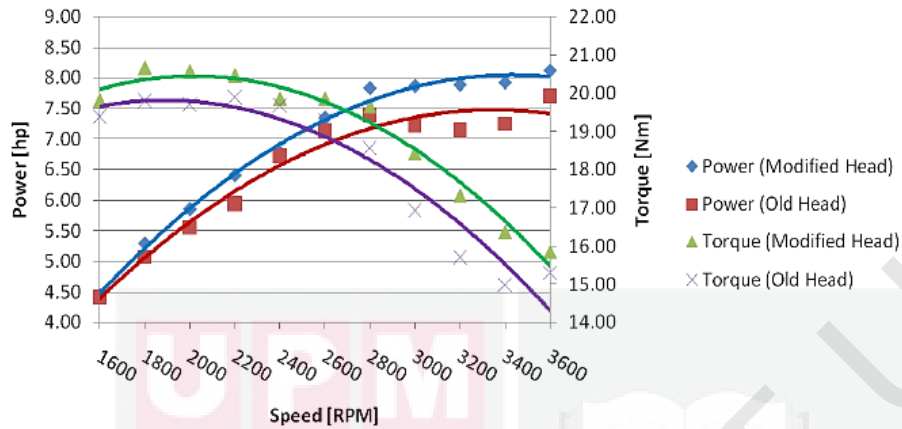


Figure 2.16: Comparison of engine performance before (old head) and after the optimisation (modified head) of the swirl intensity
(Source: Sharma et al., 2017)

However, several researchers also reported on the limitation of the fluid flow mixing method which may result in poor engine performance. An investigation by Sharma et al. (2017) concluded that a reduced flow coefficient in the intake port design reduces the performance output, even with a higher swirl intensity. The swirl air intake device, for example, may also create undesirable air restrictions due to the internal swirl geometries and sizes as reported by Saad and Bari (2014) and Cahyono et al. (2016). With the swirl device obstructing the flow of the inducted air, the resistance of airflow increases at higher engine speed due to the increase of airspeed entering the combustion chamber (Cahyono et al., 2016).

It is important to consider all the factors that can cause air restriction in applying any swirl air intake mechanism into the air intake system. An overly design piston bowl geometry and helical intake port inducing more swirl and

reversed squish formations may also have an adverse effect on the combustion characteristics and performance (Y. Chen et al., 2021; Kang et al., 2022). In a diesel double swirl combustion system (DSCS) engine as reported by Kang et al. (2022), although the air utilization is improved, however, the design may cause fuel accumulation problems due to air entrainment and wall-flow guiding effects led to incomplete combustion and soot formation.

2.4.3 Brake Thermal Efficiency

The increase in combustion efficiency can be achieved by higher turbulence and swirl air intake. The turbulence increases the rate of reaction by accelerating the chemical reaction by intimate mixing of fuel and oxygen in the air. Turbulence also produces a higher flame speed and increases the heat flow to the cylinder wall. The increase in flame speed due to the turbulence reduces the combustion duration and also minimizes the tendency of abnormal combustion. All these factors may contribute to the increase of BTE in a CI engine. The fuel-air rapid mixing has a direct effect on the brake thermal efficiency (BTE). BTE is the ratio of brake output power to input power. It describes the brake power produced by an engine concerning the energy supplied by the fuel.

Numerous investigations have been done to study the impact of the swirl air intake on the engine and based on the observation, the modification of the swirl air intake increases the brake thermal efficiency by 13.0 % to 25.6 % compared to the normal engine (A. R. Kumar et al., 2016; V. Prasad & Rangadu, 2011). The configuration of using a modified inlet manifold offers

better thermal efficiency than a normal engine due to the enhanced mixing rate carried by the swirl effects.

2.4.4 Brake Specific Fuel Consumption

Brake specific fuel consumption (BSFC) is inversely related to BTE, whereby the lower the BSFC, the higher the thermal efficiency (Sadeq et al., 2019). It is expected to have a lower BSFC when the BTE is higher. Based on the previous findings, it is observed that the modified intake manifold with swirl air intake is 4.9 % to 25.8 % lower in BSFC when compared to a normal engine (Arjunraj et al., 2021; A. R. Kumar et al., 2016; Naga Deepthi & Govinda Rajulu, 2019; NShrirao & Sambhe, 2012; V. Prasad & Rangadu, 2011; Sharma et al., 2017). They conclude that this is due to more complete combustion by the enhanced swirl in the combustion chamber which increases the fuel-air mixture quality.

However, the engine running from high to low load may produce a different combustion characteristic. There are cases with no clear effect of the swirl air intake on the BSFC found when the engine load increases (Arjunraj et al., 2021; Sadeq et al., 2019). This is due to the considerable effect of the fuel density and calorific value when a high amount of fuel is injected at a higher load. The effect of fuel properties will be more dominant with diesel-biodiesel blends. Arjunraj et al. (2021) and Sadeq et al. (2019) have investigated the effect of various diesel-biodiesel blends in a CI engine with swirl air intake and observed that the fuel properties have a considerable effect on the BSFC at higher load compared to the swirl effect.

2.4.5 Brake Mean Effective Pressure

BMEP is defined as the average mean pressure in the cylinder that would produce the measured brake output, where the pressure is calculated as the uniform pressure in the cylinder as the piston rises from Top-Dead-Centre (TDC) to Bottom-Dead-Centre (BDC) during the power stroke. Based on previous investigations of the swirl air induction in the CI engine conducted by Reddy et al. (2014) and Sharma et al. (2017), the engine operated with the swirl intake manifold results in a higher BMEP compared to the normal intake. Modification to the existing design of the cylinder head inlet port design to optimise the intake swirl also produces good combustion efficiency which consequently leads to a better BMEP. Higher BMEP is normally associated with the higher brake power and torque that can be produced by the engine.

2.4.6 Exhaust Gas Emissions

Exhaust emissions of a CI engine can be further reduced with the swirl air intake. Since better fuel-air mixing is generated with the swirl and turbulence flow inside the cylinder, more complete combustion of the fuel in the combustion chamber can be achieved. Unburnt hydrocarbon (HC) is the emission element that is a direct result of incomplete combustion (Arjunraj et al., 2021; NShrirao & Sambhe, 2012). As the engine load increases, the hydrocarbon emission will slightly increase as two factors work against each other; more fuel injected, and reduction of quenching (Sadeq et al., 2019). More injected fuel increases incomplete combustion which causes more unconsumed hydrocarbon. At the same time, a higher engine load increases the temperature inside the cylinder which reduces quenching. Flame

quenching is an undesirable condition that happens during the oxidation process which causes the unburnt hydrocarbon to remain unconsumed (Heywood, 2018).

The hydrocarbon substantially decreases with the increase of turbulence inside the cylinder which results in more complete combustion. This is consistent with the previously reported results, where, in most cases, they found lower unburnt hydrocarbon emission using the swirl air intake manifold when correlated with the ordinary manifold (Arjunraj et al., 2021; A. R. Kumar et al., 2016; Naga Deepthi & Govinda Rajulu, 2019; NShrirao & Sambhe, 2012; Reddy et al., 2014; Sadeq et al., 2019).

Incomplete combustion also produces carbon monoxide (CO). CO is resulted as the product of incomplete combustion which increases with an increase in engine load. In a swirl air intake, generally, lower carbon monoxide levels can be achieved for most cases as compared to a normal intake manifold due to the improvement in fuel-air mixture and better combustion. This is consistent with the results from the similar reports. Meanwhile, Nitric Oxides (NO_x) emission increases with the engine load due to the high combustion temperatures generated by the increased amount of fuel.

2.4.7 Effect of Ram Air Intake on the Operation of CI Engine

An IC engine with a normal air intake system typically runs with a negative pressure in the intake manifold due to the vacuum pressure created by the piston movement in the cylinder during the intake stroke. The vacuum pressure then draws the fresh air into the cylinder before the compression stroke in the

combustion cycle. The pressure drops in the air intake system cause the volumetric efficiency to fall around 75 to 80% unless, with a supercharged or turbocharged air intake, an increment of up to 29% can be achieved. In the case of supercharged and turbocharged engines, the volumetric efficiency may exceed 100% due to the boost pressure in the air intake (Rajoo et al., 2003).

The 'Ram Air Intake System' was established from the idea of the air being pushed with supercharging which also can be achieved with a ram pressure (AutoExe, 2022). With the ram effect, a positive pressure could be possibly applied to a naturally aspirated engine. However, in reality, the positive pressure is rarely applied in an engine especially for the engine running in a static condition, causing such an application to be more common to a moving vehicle. The effect of efficiently taking in air with high oxygen density will be greater when the vehicle speed is increased, given a large-capacity cleaner box, an optimized intake pipe length and diameter, and a larger intake duct opening which promotes more airflow (AutoExe, 2022; Lehenbauer, 2019).

As shown in Figure 2.17, the application of the ram air intake system is usually found as an upgrade to the existing system to improve the performance of a vehicle without sacrificing any of the factory's drivability (Lehenbauer, 2019).



(a)



(b)

Figure 2.17: Ram Air Intake System. (a) An installed system with a larger air cleaner and duct, (b) Aftermarket ram air intake and associated parts
(Source: Lehenbauer, 2019)

At present, the effect of ram on the operation of a static engine can be more associated with a pressurised air intake system that is comparable to the effect given by the supercharger and turbocharger. The effect of the pressurised air intake on a diesel engine fuelled with diesel and biodiesel blends has been investigated by Seetharamaiah et al. (2018), where the pressure is supplied by the air compressor and adjusted using the pressure regulating valve while the engine is running at full speed. Based on the outcome of the study, the increase of inlet pressure enhances the brake power due to more complete combustion. The findings of this study are explained in the next section which discusses the effect of the pressurised air intake on the homogeneous mixture in CI engines.

2.4.8 Knocking in CI Engines

In the CI engine, the ignition of fuel relies on the heat generated by the compression of air. Autoignition happens when the chemical processes of precombustion reactions of the fuel, air and residual gas mixture take place

before releasing a significant fraction of the injected fuel's chemical energy (Heywood, 2018). Apart from that, the physical processes such as the atomisation of the liquid fuel, vaporisation of the fuel droplets and the mixing of fuel vapour with air must also take place before the significant chemical energy is released. Due to the autoignition phenomenon, there will be results from the unintended autoignition of a portion of the precombustion reaction mixtures. Whilst these mixtures cause a knock effect by spontaneously igniting ahead of the advancing flame propagation in the combustion chamber for SI engine, diesel knocking occurs when excess quantities of the injected fuel, under various circumstances, auto-ignites and combusts in the premixed combustion stage (Lowe et al., 2011).

According to Lowe et al. (2011), knock in CI engines is also referred to as 'Diesel Knock' or 'Diesel Detonation'. The knocking is clearly audible once the abnormal combustion releases a large amount of energy to produce the pressure waves. This causes unwanted vibrations in the chamber and the cylinder block, affecting the power output and directly impacting the engine efficiency, where the knock limits the maximum compression ratio that can be used by the engine (Heywood, 2018; Moshrefi & Shoaee, 2017; Yin et al., 2016).

Moshrefi & Shoaee (2017) further explain that these vibrations will lead to several detectable phenomena which include cylinder pressure oscillations, and engine vibrations with an audible noise. Knock detection can be done by measuring the intensity of the vibration by a specific knock sensor, where the most accurate sensor according to Moshrefi & Shoaee (2017) is the cylinder

pressure sensor. Figure 2.18 shows the signal of the cylinder pressure sensor involving knock and no-knock conditions.

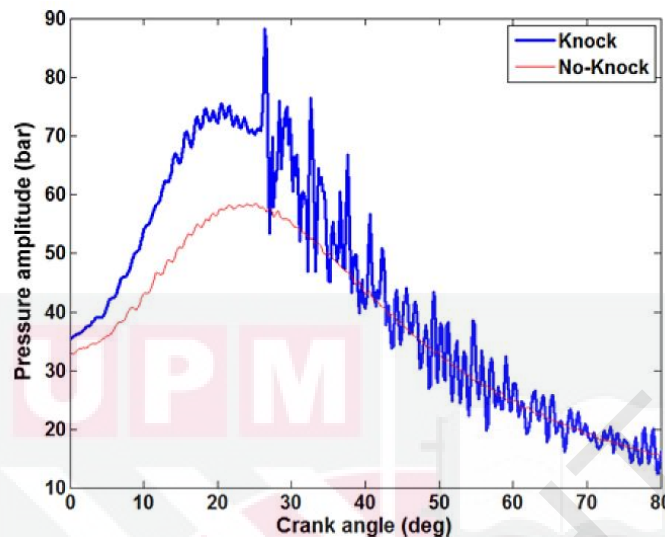


Figure 2.18: In-Cylinder Pressure Amplitude with the Knock and No-Knock conditions

(Source: Moshrefi & Shoaee, 2017)

2.5 Strategies for Homogeneous Mixture in CI Engines

The effect of fuel-air mixing on the efficiencies of the CI engine has been widely discussed and investigated. The objective of the investigation is mainly to achieve a better homogeneous mixture of the fuel and air before the start of combustion. Table 2.2 shows some of the recent studies of the fuel-air mixing effect using various strategies on the CI engine characteristics. In general, the homogenous fuel-air mixture in the combustion chamber involves with the fuel injection, combustion control and fluid flow mixing strategies.

Table 2.2: Investigation of fuel-air mixing effect using various strategies on the CI engine efficiencies

No	Reference	Mode of Combustion	The strategy to form a better air-fuel mixture	Effect on engine characteristics/ efficiencies
1	Kesharvani et al. (2023)	Compression-Ignition	Fuel pre-mixed with nanoparticles	Approx. 2% increase in BTE
2	L. Guo et al. (2023)	Compression-Ignition	Dual fuel pre-injection	Approx. 2% increase in ITE
3	Qian et al. (2023)	Gasoline Compression-Ignition	Fuel dual-direct injection	Min. 7.7% increase in ITE
4	Vasanthakumar et al. (2023)	Compression-Ignition	Hydrogen-enriched intake air and ethanol blend	Min. 3.8% increase in BTE
5	Honecker et al. (2023)	Compression-Ignition	Molecularly-Controlled high swirl Combustion	Approx. 2.5% increase in ITE
6	Kang et al. (2022)	Compression-Ignition	Double swirl combustion system	Max. 4.8% increase in air-fuel uniformity index
	Z. Wang & Li, (2022)	Compression-Ignition	Fuel Pre-Injection	Max. 61.18% decrease in CO
7	Y. Wang et al. (2022)	Gasoline Compression-Ignition	Fuel pre-injection	Min. 14.4% increase in BTE
8	Şener & Gül (2021)	Compression-Ignition	Double swirl combustion system	Approx. 3.2% increase in BP
9	Hamid et al. (2020)	Compression-Ignition	Guide vane intake manifold	Max. 5% increase in In-Cylinder airflow efficiency

2.6 Fuel Injection Strategy

Achieving a better air-fuel mixture is possible by extending the ignition delay time. When the ignition delay is longer, the fuel mixes more homogeneously with the air, resulting in more fuel being burned together after ignition. Ignition characteristics are largely affected by the fuel injection timing, where the timing of the injection as well as the fuel properties significantly influence ignition delay time and combustion duration (Erman et al., 2020). Ignition delay can be extended by applying different fuel injection strategies as performed by Qian et al. (2023) and Wang et al. (2022).

2.6.1 Fuel Dual Injection

Qian et al. (2023) state that the medium-pressure single-direction injection mode is suitable for IMEP of 4 to 10 bar. However, at higher loads, the indicated thermal efficiency is decreased. At a medium injection pressure, single DI mode combustion is limited by the fuel mixing rate. Thus, to improve the efficiency at high loads, a medium-pressure dual direct injection is much preferable for IMEP of 10 to 12 bar. The dual injection strategy is causing an improvement in the fuel-air mixing rate by having a slightly shorter ignition delay than that of a single DI mode. This strategy increases the heat release rate and further improves the engine efficiency (Qian et al., 2023). Another study conducted by L. Guo et al. (2023) introduces the ammonia-diesel dual fuel (ADDF) in a CI engine through simulation work. The use of ADDF in the CI engine has better overall performance compared to pure diesel mode at 70% ammonia energy ratio, which increases the indicated thermal efficiency by about 23%.

2.6.2 Fuel Pre-Injection

Pre-injection strategy has been proposed by some researchers in addition to the strategy of changing the injection timing of the single-injection method to improve the fuel-air mixing process (Guo et al., 2023; Y. Wang et al., 2022; Z. Wang & Li, 2022). As suggested by Y. Wang et al. (2022), the pre-injection strategy directly affects the stratification of the mixture which provides a significant impact on the combustion reaction. The pre-injection could reduce the overall inhomogeneity in the cylinder and reduce combustion instability. This improves the combustion quality and characteristics which further

improves the torque and brake thermal efficiency accordingly. In the simulation work of L. Guo et al. (2023), the addition of a pre-injection strategy in the CI engine mode resulted in a more concentrated exothermic process with a more fixed-volume combustion process, leading to an improvement in the indicated thermal efficiency by 2% compared to single-injection.

In terms of emissions gases, the use of a pre-injection strategy improves the combustion and emissions characteristics as investigated by Z. Wang & Li (2022) in the diesel engine. Increasing the pre-injection timing and the pre-injection mass ratio increases the in-cylinder pressure and temperature as well as decreasing soot, HC and CO emissions. The increase in peak in-cylinder pressure was also reported by L. Guo et al. (2023) and Y. Wang et al. (2022) from the previous explanation.

2.7 Combustion Control Strategy

As the CI engines rely on the self-ignition of the fuel-air mixture at high temperature and pressure during compression, a strategy that can control the behaviour of the ignition may produce a significant improvement to the overall combustion efficiency. Optimising the ignition control greatly helps in achieving better fuel efficiency, power output and reduced emissions levels across a wide range of operating conditions. As a key element to have better ignition control, the strategy should have control over combustion timing with an impact on any of the atomisation, penetration, evaporation and mixing of the fuel with air for homogeneous mixing. This will be better presented with the following

strategies demonstrated by numerous investigations in the CI engine operation with the improvement in the fuel-air homogeneity.

2.7.1 Pre-mixed fuel

The study by Kesharvani et al. (2023) evaluated the effect of cerium oxide nanoparticles doped water diesel emulsion fuel on multi-cylinder compression ignition engines. The investigation found that the nano-particles help to improve engine performance with approximately a 2% increase in brake thermal efficiency while lowering harmful engine pollutants. This strategy results in a faster mixing of fuel with air by the increases of the evaporation speed due to the nanoparticle sheath.

A similar strategy of using premixed fuel in improving the fuel-air mixing is also investigated by Vasanthakumar et al. (2023). They work on a diesel engine operated on dual fuel mode by ethanol-diesel blends with hydrogen-enriched intake air which the aim is to replace conventional diesel fuel with low and no-carbon fuels. The use of the premixed fuel enriched with hydrogen results in an increase of a minimum of 3.8 % in brake thermal efficiency. The improvement of engine efficiency is found to be due to the high diffusivity and volatility attribute of hydrogen that ensures the formation of a better homogeneous flammable mixture, leading to faster flame propagation and better combustion.

The excess oxygen content in ethanol also improves the air-fuel mixture formation. Based on Vasanthakumar et al. (2023), the lower heating value (LHV) of ethanol induces a cooling effect on the intake charge compared to

diesel. This, in turn, increases the charge density, leading to better fuel-air mixture formation.

2.7.2 Pressurised Air Intake

Pressurised air intake can be achieved by increasing the inlet air pressure by means of supplying higher air pressure to the intake manifold. This is typically achieved by the application of the supercharger and turbocharger that is most preferred to any IC engine (Hairuddin et al., 2014; Seetharamaiah et al., 2018). The application of these devices provides a higher intake pressure into the combustion chamber. The pressurised intake air may increase the charge density, leading to a significant increase in engine performance and better efficiency (Hairuddin et al., 2014).

The effect of the pressurised air intake on the performance and emissions of a CI engine has been investigated by Seetharamaiah et al. (2018). Based on the findings, the brake power is enhanced with the increase of inlet pressure from 1 to 2.5 bar due to more complete combustion. This inlet pressure range is still within the boost pressure typically provided by a turbocharger. For example, the range of boost pressure controlled by a poppet-valve-type wastegate for a 2.38-litre six-cylinder automotive diesel engine is normally up to 0.75 bar above the atmospheric at full load, while higher boost levels can be provided in more recent turbocharged diesel engine beyond 4 bar (Heywood, 1988, 2018). Therefore, a slightly higher inlet pressure range that is controlled by the pressure regulator should be able to produce more clear behaviour for a better comparison at different pressures. Since each of the fuel

molecules is surrounded by the air (oxygen) molecules due to higher air intake pressure, more complete combustion could happen, leading to an improved brake power of the engine. This investigation also found lower CO and HC emissions levels at higher inlet pressure, but a slight increase in NO_x level as shown in Figure 2.19.

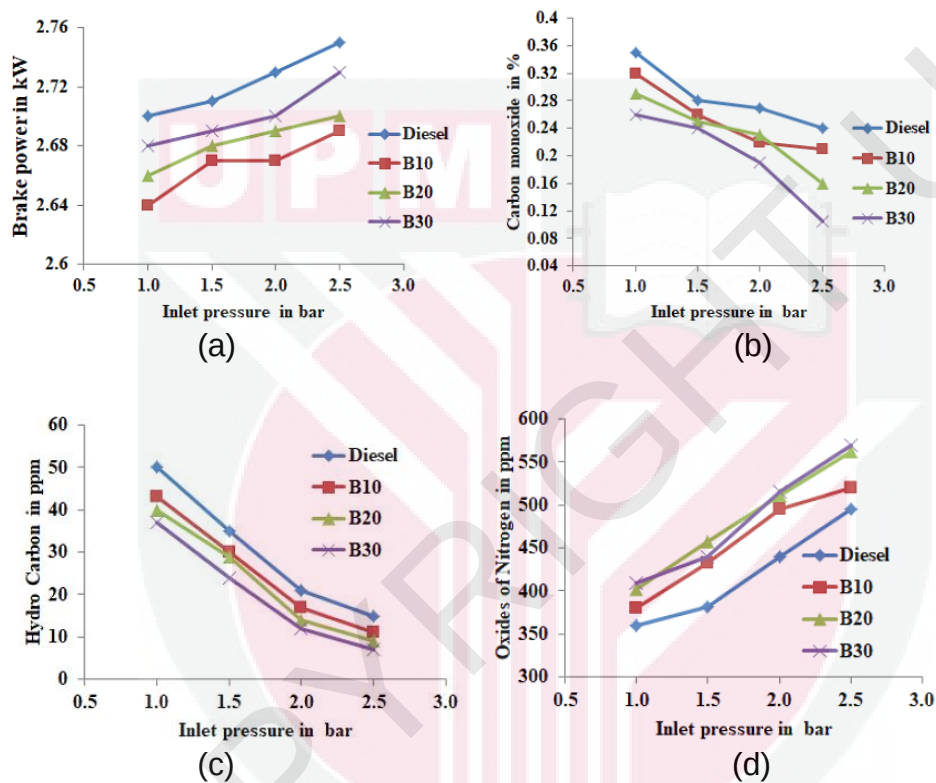


Figure 2.19: Effect of inlet pressure variations to the CI engine. (a) Brake Power, (b) Carbon Monoxide, (c) Hydrocarbon, (d) Nitrogen Oxides.
(Source: Seetharamaiah et al., 2018)

2.7.3 Hydrogen Addition

Hydrogen, by far, still represents a good alternative fuel that can be used in an IC engine due to its higher Lower Heating Value (Net Calorific Value) and good burning qualities compared to other fuels such as diesel fuel. Table 2.3 shows the properties of hydrogen and diesel fuel as reported by Cernat et al. (2023).

The use of hydrogen may improve the diesel engine combustion process because of the quality of the burning, where the use of hydrogen as an addition to the diesel fuel in the CI engine operation led to the maximum pressure increase by 17%, decrease in BSFC at 7.8%, and decrease in CO₂ and NO_x emissions. The effect of hydrogen addition through the experimental study has been previously done by Dimitriou et al. (2018) in a hydrogen-diesel dual-fuel engine. The trend of the emissions observed by Dimitriou et al. (2018) is identical to the one reported by Cernat et al. (2023) in the reduction of carbon and NO_x emissions. With the increase of hydrogen addition, the increase in the maximum in-cylinder pressure is due to the higher maximum pressure rise rate. The higher combustion speed of an air-hydrogen homogeneous mixture against the normal air-diesel leads to an increase in the heat release rate, combustion speed and a reduction of the combustion duration.

Table 2.3: Properties of diesel and hydrogen fuels

Properties	Unit	Diesel	Hydrogen
Density at 16 °C	kg/m ³	833-881	0.0838
Auto-ignition temperature	K	530	858
Flammability limits	Vol.% in air	0.7-5	4-75
Stoichiometric air mass quantity	kg in air	14.5	34.32
Flame speed	cm/s	30	265-325
Heating Value (LHV) / Net Calorific Value	MJ/kg	41.855	119.6178
Cetane Number	-	44-55	-
Octane Number	-	30	130

(Source: Cernat et al., 2023)

2.7.4 Exhaust Gas Recirculation (EGR)

There are numerous studies on the implementation of EGR in the CI engine operation which predominantly reduce the NO_x emission (Fayad et al., 2022; Joshi, 2018; Pan et al., 2019; Wozniak et al., 2023). The reduction of NO_x is achieved through the reduction of the combustion temperature while improving the fuel evaporation with the increase in the inlet air temperature (Wozniak et al., 2023). Increasing the inlet air temperature through the EGR is possible since the exhaust gas flow is being recirculated back to the engine cylinder from the exhaust manifold via an air intake pipe. By adding the EGR back into the combustion chamber, the amount of oxygen O_2 inside the combustion chamber is reduced with a higher amount of carbon dioxide CO_2 (Fayad et al., 2022). The higher amount of CO_2 with reduced O_2 level produces lower combustion temperature and thereby reduces the NO_x emissions.

As a control technology for NO_x emissions, EGR may also control the combustion duration via the high and low-pressure cooled EGR which eventually causes a reduction in the combustion temperature, leading to the NO_x and CO emissions reduction (Pan et al., 2019). Wozniak et al. (2023) further classify the EGR into two types of EGR systems; hot and cooled EGR. Hot EGR normally employs a direct recirculation of the exhaust gases to the intake pipe, while the gas-coolant heat exchanger is utilised for the operation of the cooled EGR. Hot EGR can effectively control the ignition as an increase in the inlet charge temperature may result in a shorter ignition delay, thereby improving the thermal efficiency. On the other hand, a cooled EGR increases the charge density, thus, improving the volumetric efficiency (Hussain et al.,

2012). Joshi (2018) explains that the cooled EGR has a more significant impact on the NO_x reduction and is more effective compared to the hot EGR according to their study. This is because the increase in the combustion temperature due to the higher EGR gas temperature will lead to an increase in NO_x formation.

2.7.5 Charge Stratification

Charge stratification, also referred to as Stratified Charge Compression Ignition (SCCI) is one of the strategies to control the ignition of CI combustion with the fuel composition stratification that may affect the auto-ignition characteristics. It also can minimize NO_x, particulate matter and soot. However, excessively higher stratified charge leads to a significant increment in NO_x emissions (Gupta & Mogra, 2020). Thus, the parameter setting for stratified charge engines must be properly studied and tested.

As specified by Heywood (2018), stratified-charge engines may achieve charge stratification when the mixture within the prechamber is richer than in the main chamber due to the fuel injection or with a separate prechamber intake valve. This is attained by the combustion in the main chamber initiated by the flame originating from the prechamber, where the flame is created by the ignition process and successive energy release inside the prechamber. The application of a prechamber or auxiliary combustion chamber is typically found in small high-speed compression-ignition engines to achieve adequate fuel-air mixing rates (Heywood, 2018). In achieving different fuel mixture stratification, Gupta & Mogra (2020) state that the change in stratification level can be attained by varying the timing of the fuel injection.

2.8 Fluid Flow Mixing Strategy

Better fuel and air mixing can also be achieved by applying different fluid flow mixing approaches. This approach can be done at various locations from the intake system to the in-cylinder area through a certain mechanism. For the external approach, the air intake will be guided to the inlet port through a certain intake manifold design or with a separate device to automatically form the air swirl. The air swirl generated by the device will be drawn into the cylinder and expected to increase the fuel-air mixing rate for a homogeneous mixture.

2.8.1 Swirl and Tumble Motion Inside Cylinder

For the in-cylinder approach, this method will involve the motion of the gas charge within the cylinder. To achieve adequate fuel-air mixing, it is necessary to use a swirling air flow rotating about the cylinder axis, which is created by force with the suitable design of the helical inlet port and valve. The initial in-cylinder flow pattern set up by this intake process may be substantially modified during compression by the piston bowl. The use of bowl-in-piston influences the movement of air and fuel during the compression stroke, and increases the swirl rate, thereby affecting the fuel-air mixture (Caeses, 2017; Heywood, 2018).

Among the studies that focus on fuel-air mixing through the fluid flow mixing strategies are done by Honecker et al. (2023), Kang et al. (2022), Şener & Gül (2021) and Hamid et al. (2020). Kang et al. (2022) investigated the effect of optimizing the double swirl combustion system (DSCS) diameter in a diesel

engine on the fuel-air mixing process. The improvement to the fluid swirl motion characteristics has contributed to the fast fuel-air mixing rate in the outer chamber and the clearance. High indicated power and low soot generation are also observed based on the study. They found that the whole-chamber uniformity index by optimizing the DSCS has increased up to 4.8 %.

The increase is due to the improved air entrainment and the reverse squish together with the wall-flow-guiding effects and in-cylinder motion. The effect of DSCS on the CI engine was also studied by Şener & Gül (2021). They observed that by the enhanced in-cylinder air utilization and rapid mixing-controlled combustion in the optimized DSCS design, there are improvements in combustion efficiency and engine performance with a reduction in exhaust emissions.

2.8.2 Swirl Generation in The Air Intake System

In the study conducted by Hamid et al. (2020), they incorporated the guide vane design (GVD) with a shallow depth re-entrance combustion piston (SCC) to improve the engine performance. The GVD is a separate device installed in the intake runner of the engine to guide the airflow before it is induced into the cylinder. The GVD vanes guide the airflow and consequently generate higher turbulence and create sufficient momentum to sustain the swirl up to the end of the expansion stroke (Hamid et al., 2020). In-cylinder turbulence airflow tends to accelerate and improve the fuel-air mixing process due to its capability of breaking down heavier molecules and higher viscosity (Hamid et al., 2020; B. V. V. S. U. Prasad et al., 2011).

One of the measurable variables that determine the improvement in the in-cylinder turbulence flow is the turbulence kinetic energy (TKE). Higher turbulence airflow can be indicated by the higher TKE. Higher TKE will increase the speed and the ability to aid the atomisation of molecules during spraying and, thus, have a greater ability to break down the fuel molecules (Hamid et al., 2020). The use of GVD and SCC pistons has shown a higher in-cylinder TKE compared to that of the engine without the GVD. The increase of TKE, among others, contributes to the result of a 5% increase in the in-cylinder airflow efficiencies.

2.9 Numerical Study and CFD Environments in CI Engines

In predicting the fluid flow characteristics in internal combustion engines, numerical study has been used extensively to simulate and analyse the behaviour of fluids inside the engine. The numerical method is used to discretise the governing equations and solve these equations numerically. Simulation works are based on mathematical models to describe the behaviour of the fluid flow and its interaction with the solid body depending on the boundary conditions and assumptions. Three categories of simulation are very common in the numerical study which are zero-dimensional, quasi-dimensional and multi-dimensional simulations (Hafizil et al., 2017; D. Zhang et al., 2023).

Zero-dimensional often referred to as '0D' simulation provides a simplified representation of the entire engine system, where the engine is considered as a single control volume with the fluid flow and combustion process represented

by a set of simplified models derived from the governing equations. However, zero-dimensional simulations have limited accuracy due to the simplified model which does not accurately represent more complex fluid flow and combustion process. This limitation can be improved by necessary modifications to the model in calculating the heat release rate in diesel engines (Feng et al., 2019).

Quasi-dimensional simulations provide a compromise between computational efficiency and spatial details where the simulation is done according to the multiple interconnected control volumes. Each simulation of the quasi-dimensional considers the inclusion of additional models to capture specific phenomena. In the fluid flow study, quasi-dimensional provides a more accurate prediction with an additional model for capturing the turbulence within the intake pipe and cylinder. For a swirl prediction in a CI engine, the zero-dimensional model can be further developed with a sub-model working under the swirl environment in the heat release model (Broatch et al., 2019; Wu et al., 2022).

Multi-dimensional simulation provides the highest level of detail where the complex flow structures such as swirl, tumble and turbulence can be captured to study the effect on combustion and emissions. Computational Fluid Dynamics (CFD) is used in the multi-dimensional simulation to solve the governing equations of fluid flow and combustions in three-dimensional distribution. CFD is extensively used in fluid flow analysis with the ability to provide a detailed and accurate representation of flow physics. This is a well-known method which has been widely used to simulate and analyse the flow

characteristics, pressure difference, and other effects within the cylinder as well as in the intake systems.

2.10 Combustion and Fluid Flow Modelling in CI Engines

2.10.1 Multi-Dimensional Simulations in the Prediction of Combustion

Process with Swirl.

CFD is a multi-dimensional simulation that has been used to study the flow behaviour inside the air intake system, where the analysis is done by focusing on the physical properties and flow conditions of the fluid with the specified wall boundary conditions. From the previous discussions, the flow behaviour of the inducted air charge has a significant effect on the quality of the fuel-air mixing in the cylinder. Hence, the fluid flow study is important to be conducted in identifying the flow characteristics of the inducted air charge before any work related to combustion kinetics and thermodynamics.

In predicting the effect on emissions and combustion efficiency due to swirl formation in CI engines, the work can also be performed with the guidance of CFD. The CFD-based simulation has been employed by numerous researchers in analysing the combustion behaviour inside the combustion chamber involving a specific transport equation based on the energy, momentum and continuity equations. This includes the transport of species which comprise the turbulence characteristics for detailed analysis of the swirl effects (Sener & Gul, 2021).

Figure 2.20 shows the multi-dimensional simulation results as performed by Sener & Gul (2021) in investigating the effect of swirls on the in-cylinder temperature distribution through the injection parametric study. The combustion temperature and velocity distributions shown in Figure 2.20 are the successful cases of the air-fuel mixing process which have been effectively improved inside the cylinder with the effect of swirl. This shows the effectiveness of the multi-dimensional model in accurately predicting the behaviour of the combustion.

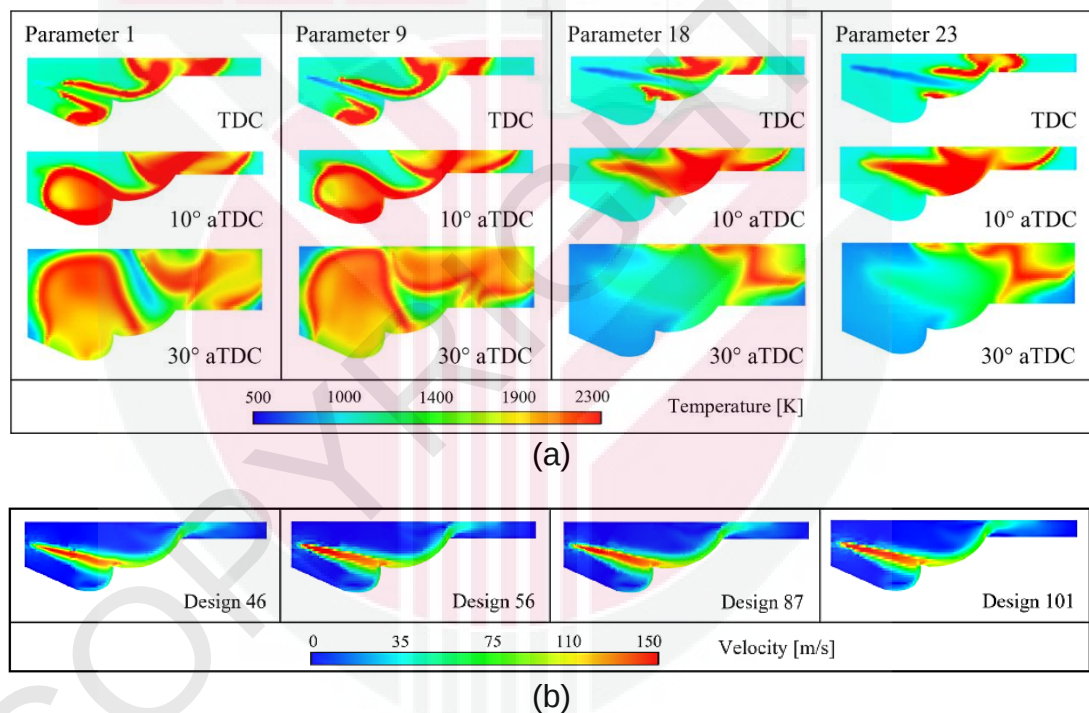


Figure 2.20: (a) In-cylinder temperature and (b) velocity distributions due to swirl based on the selected cases using a Multi-Dimensional model
(Source: Sener & Gul, 2021)

The study that focuses on the fluid behaviour inside the cylinder has been conducted by Oddiwar and Koshti (2017). Figure 2.21 shows the CFD result of the streamlined flow during the air induction process with the swirl measured inside the cylinder. The change of the valve lift changes the swirl ratio, TKE

and TED of the air intake charged inside the cylinder. The flow starts from the intake runner directly to the intake port and ends inside the cylinder. Based on the intake port analysis of a diesel engine during the air induction process, they observe the increase of the swirl value and mass flow rate induced by the increase of the valve lift. The study also concludes that CFD is capable of predicting the value of the swirl with 5% error between CFD and experimental results.

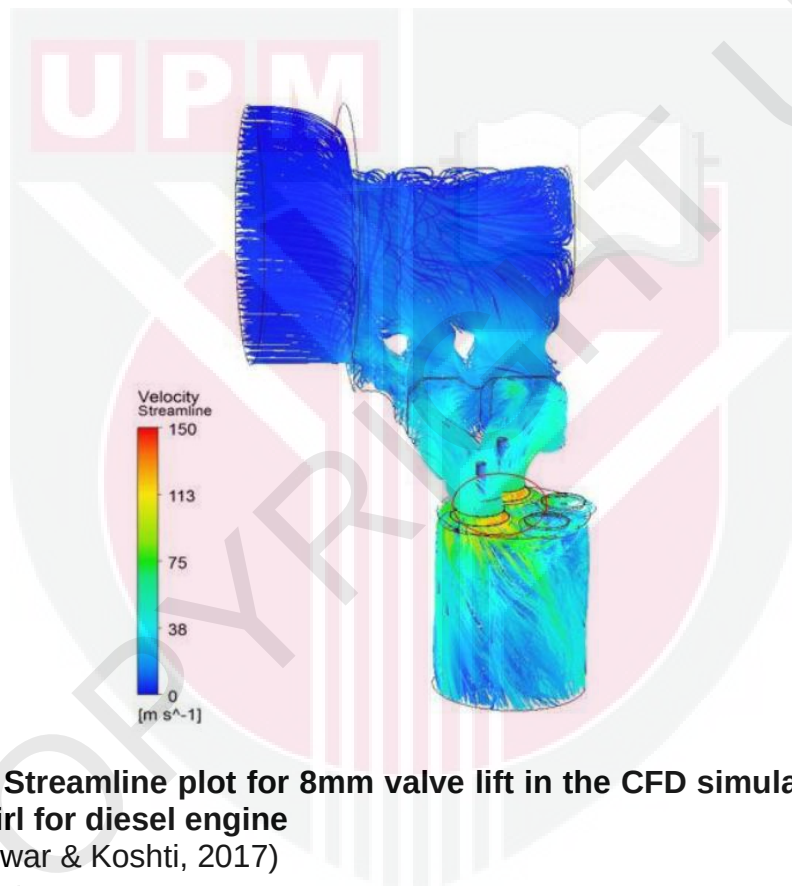


Figure 2.21: Streamline plot for 8mm valve lift in the CFD simulation of intake air swirl for diesel engine
(Source: Oddiwar & Koshti, 2017)

There are also simulation works related to the fluid flow study before the air reaches the intake port (Gocmen & Soyhan, 2020; Ruqaiyah et al., 2023; Xu, 2017). Figure 2.22 shows the simulation work done by Ruqaiyah et al. (2023) in predicting the vortex phenomenon and the unevenness of the design which affects the intake airflow into the cylinder. Meanwhile, Figure 2.23 shows the result of the new intake manifold geometry designed by Gocmen & Soyhan

(2020) which is expected to improve the swirl intensity in the cylinder, thereby reducing exhaust emissions. All presented simulation works as discussed in this section indicate the effectiveness of CFD as the simulation tool for fluid flow modeling in the CI engines.

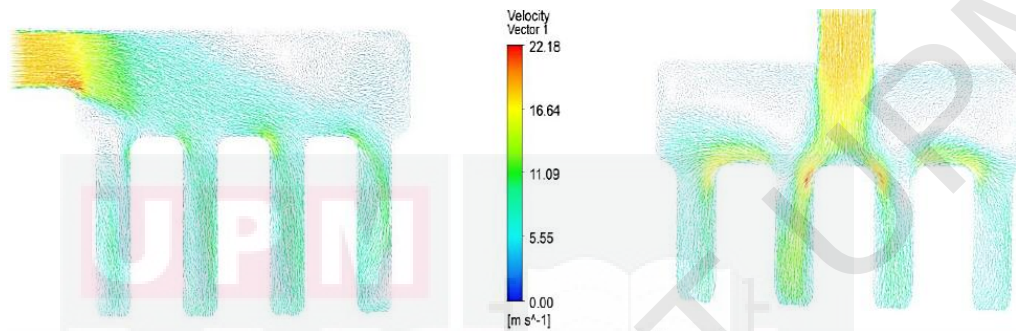


Figure 2.22: Velocity vector of the intake manifolds in CFD simulation
(Source: Ruqaiyah et al., 2023)

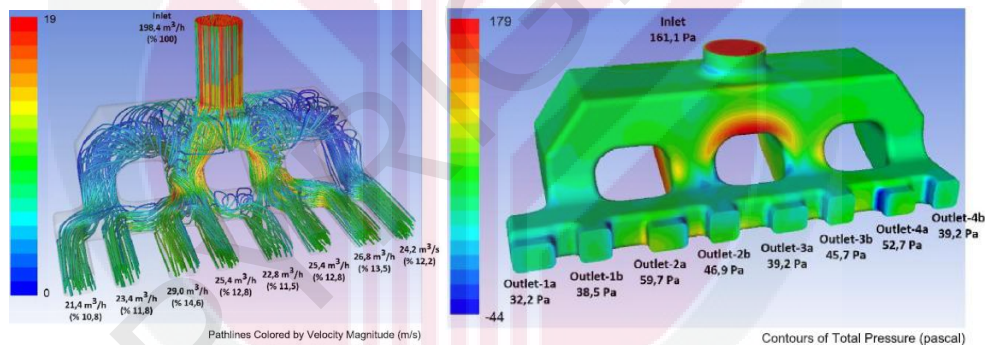


Figure 2.23: Velocity and pressure distribution of the new intake manifold geometry
(Source: Gocmen & Soyhan, 2020)

2.10.2 Turbulence Models

The k-epsilon ($k-\epsilon$) model is regularly used as a standard turbulence model in the CFD in many simulations to predict the airflow characteristics inside the intake manifold and engine cylinder (Gocmen & Soyhan, 2020; Oddiwar & Koshti, 2017; Xu, 2017). In previous investigations, researchers have pointed

out that the k - ϵ turbulence model is considered satisfactory compared to the experimental results (Gocmen & Soyhan, 2020). However, the k - ϵ turbulence model accuracy may be limited in predicting complex flow phenomena such as flow separation and highly swirling flows, where this model is mainly applied in the steady-state simulations, and thus may not accurately capture the transient or unsteady flow phenomena.

The realizable k - ϵ model has been found to be more accurate than the other k - ϵ models for boundary layer flows involving high-pressure gradients and complex flow structures (Araoye et al., 2017; Qu et al., 2019). This turbulence model has been specifically employed by several fluid dynamics simulation works that cover various swirl flow strengths or intensities (Araoye et al., 2017; Kozlov et al., 2016; Pofalkar et al., 2017; Qu et al., 2019).

Another turbulence model may also perform better compared to the standard k - ϵ turbulence model. Turbulence models such as the RANS-RNG k - ϵ turbulence model (Sener & Gul, 2021) and RNG k - ϵ turbulence model (Z. Wang & Li, 2022) are more appropriate for more accurate and detailed turbulence modelling involving complex flow phenomena. K-omega (k - ω) turbulence model (Zaidi, 2019), Shear Stress Transport (SST) (Ruqaiyah et al., 2023) and Transition-SST (k - ω) turbulence model (K. Kumar et al., 2023) are the other types of turbulence models that are available to be used in simulating the fluid flow involving swirl airflow study. However, the choice of turbulence model is very much dependent on the specific flow characteristics aimed to be studied and higher computational cost and time should also be considered.

2.11 Governing Equations of Swirl

Swirl has several definitions according to some areas of analysis. Different definitions will implement different concepts of swirl measurement. In engine operation, a swirl is usually defined as an organized rotation of the charge about the cylinder axis. It is created by bringing the intake flow into the cylinder with an initial angular momentum. This generated swirl usually persists through the compression, combustion and expansion processes while some decay in swirl occurs during the engine cycle due to friction.

According to Abo-Elfadl and Abd El-Sabor Mohamed (2018), the swirl ratio is calculated by applying the common momentum-based definition on the velocity field inside the cylinder. When swirl measurements are made in an operating engine, a Swirl Ratio, R_s as shown in Equation (2.1) is normally used to define the swirl (Heywood, 2018). It is defined as the angular velocity of a solid-body rotating flow ω_s , which has equal angular momentum to the actual flow, divided by the crankshaft's angular rotational speed.

$$R_s = \frac{\omega_s}{2\pi N} \quad (2.1)$$

where ω_s is the angular velocity of a solid-body rotating flow, and N is the crankshaft's angular rotational speed.

Assuming that the port and valve retain the same characteristics under transient conditions of the engine as on the steady-flow rig, the equivalent solid-body angular velocity, ω_s at the end of the intake process is given by Equation (2.2) below. θ_1 and θ_2 are crank angles at the start and end of the

intake process and the torque T and mass flow rate $\dot{m}$ are evaluated at the valve lift corresponding to the local crank angle.

$$\omega_s = \frac{8}{B^2} \left(\int_{\theta_1}^{\theta_2} T d\theta \right) / \left(\int_{\theta_1}^{\theta_2} \dot{m} d\theta \right) \quad (2.2)$$

where θ_1 is the crank angles at the start of the intake process, θ_2 is the crank angles at the end of the intake process, T is the torque, and $\dot{m}$ is the mass flow rate.

Another measure of swirling flow is known as Swirl Number, S_n (Boushaki, 2019; Qu et al., 2019; Syred & Bekr, 1974) and, according to Qu et al. (2019), it was first proposed by Beer and Chigier (1972), as

$$S = 2G_\phi / G_x D_e \quad (2.3)$$

where G_ϕ is the axial flux of angular momentum, and G_x is the axial flux of linear momentum.

The swirl number determines the intensity of the swirl; the more this number is higher, the more the swirl effect is strong (Boushaki, 2019). However, the studies of early and more recently made by using Equation (2.3) and related equations (G_ϕ and G_x) mainly focused on the swirl effect on flames, which considering the geometry of burner nozzle and swirl generators. Based on this swirl concept, several equations have been developed by researchers to study the swirl effect on different applications and locations.

An alternative for swirl number was defined by Rocklage-Marliani et al. (2003) as rotation number, R_o and later used by Andrew and Cui (2010) as shown in Equation (2.4). Andrew and Cui (2010) tested several values of rotation number from 0.25 to 1 that were generated by the swirl generator in a smooth straight pipe to gain an understanding of the flow with several turbulence models involved in the simulation.

$$R_o = \frac{\Omega D}{2U_{ave}} \quad (2.4)$$

where Ω is the angular velocity of the swirl generator (refers to the pipe radius), and U_{ave} is the mean axial velocity

To analyse different definitions of the swirl number and their application to control the mixing process in the cylinder of direct injection diesel engines, another possibility for determining the swirl number based on equivalent angular velocity and comparison of different techniques for measuring swirl in motor cylinders has been discussed by Crnojevic et al. (1999). The determination of the swirl number in the axisymmetric case requires the calculation of the moment of axial, M_x and tangential M_ϕ mean components. These are calculated theoretically for a steady axisymmetric flow, which is the most convenient case of flow for industrial applications (Tobal et al., 2011). The expression for the swirl number is given in the form shown by Equation (2.5).

$$S = \int_0^R \bar{U} \bar{W} r^2 dr / R \int_0^R \bar{U}^2 r dr \quad (2.5)$$

where U is the axial component of the velocity, and W is the tangential component of the velocity.

This equation considers an internal helical flow, or swirl, in a tube of radius R as illustrated in Figure 2.24. A fluid particle of mass is displaced with velocity, v as shown in Equation (2.6).

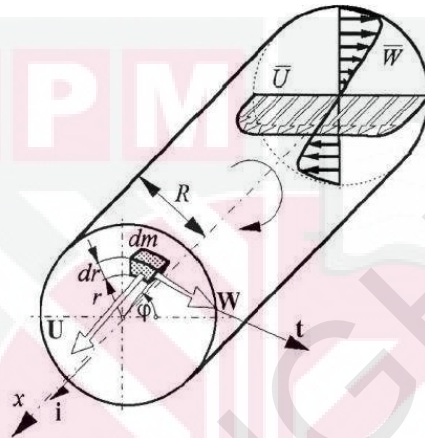


Figure 2.24: Flow with swirl in a tube
(Source: Crnojevic et al., 1999)

$$\mathbf{v} = U\mathbf{i} + W\mathbf{t} \quad (2.6)$$

where U is the axial component of the velocity, and W is the tangential component of the velocity.

The concept used by Crnojevic et al. (1999) was further extended and utilized by Tobal et al., (2011) for measuring the swirl flow in an engine cylinder as shown in Equation (2.7). The swirl number as defined by Tobal et al., (2011) is the ratio between the moment of axial momentum flux and the product of cylinder radius by axial momentum flux which are calculated from the velocity

components of U and W shown in the previous figure (Figure 2.24). The swirl number is a dimensionless number that characterizes the intensity or strength of the swirling flow (Tobal et al., 2011).

$$S = \int_A \bar{U} \bar{W} r dA / R \int_A \bar{U}^2 dA \quad (2.7)$$

where A is the cross-sectional area of the cylinder, U is the axial component of the velocity, and W is the tangential component of the velocity as can be seen in Figure 2.24 from Crnojevic et al. (1999).

2.12 Flow Strength Prediction

Swirling flow has many definitions as presented in the previous section. Two common measures of swirl strength are Swirl Number, S and Swirl Ratio, R_s , which depends on the application. Therefore, the comparison of swirl intensity cannot be simply made without considering the area of the swirl. For engine study, the swirl intensity of the charge motion in the cylinder refers to the Swirl Ratio, meanwhile, the airflow motion in the intake manifold may refer to the Swirl Number by referring to the previous definition of swirling flow in a cylinder (Crnojevic et al., 1999; Tobal et al., 2011).

For the swirl application in a tube or cylinder, the swirl strength or intensity can be generally divided into two main categories; high-swirl and low-swirl flow. In terms of the swirl number level, the flow is characterized by a low-swirl flow with $S < 0.4$ and a high-swirl flow with $S > 0.6$ (Keshtkar, 2018). Alternatively, the flow can also be characterized by a weak or moderate swirl if $S < 0.5$, while

a highly swirling flow can be obtained for swirl numbers of $S > 0.5$ (Haminudin & Zamberi, 2020). Then, the swirl intensity may be further categorized with the medium-swirl number of $S = 0.5$.

Based on the CFD simulation performed by Tobal et al. (2011), the reported swirl number range is between 0.18 to 0.45 along the cylinder (combustion chamber) centre axis of an engine having a spiral intake duct design. Vondal (2011) reported that the swirl number of a combustion chamber burner swirl generator has a range between 0.65 to 1.48 as generated by the axial guide vanes through the simulation and published experimental data. For combustion in a jet engine application, the swirl number can go up to 1.91 as reported (Keshtkar, 2018). These studies indicate that the swirl number varies according to a certain application and the area of analysis, thus for the verification of the swirl number, a general comparison against the typical range is more appropriate. Based on the results of numerous numerical and experimental tests, the available equations as presented in the previous section are reliable methods to predict the swirl according to each condition and definition.

In predicting the swirl strength produced by the intake manifold, the suitable equations to be employed are Equations (2.5) and (2.7). It is due to the fact that the generated swirling flow in the single intake manifold pipe will be calculated along the pipe's cylindrical axis without any relation to the engine's characteristics such as valve, port design and speed of the crankshaft. The swirl number calculated using Equations (2.5) and (2.7) will consider the

internal helical flow, or swirl, in the helical intake manifold of a single pipe of radius R .

As an alternative to Equations (2.5) and (2.7), a similar concept is adopted in the equations provided by Keshtkar (2018), Vaidya et al. (2011), and Zhou (2012). The following Equations (2.8-2.10) have different expressions in the axial flux integration by considering tangential and axial velocity at the corresponding radial position. Referring to other equations, Equation (2.1) is only relevant for the swirl measurement inside the engine cylinder together with the rotating crankshaft under similar characteristics as the steady-flow rig, while Equation (2.4) is used with the angular velocity of the swirl generator and thus not relevant to the study of swirl air in the intake manifold.

Swirl number, S_z expression given by Zhou (2012)

$$S_z = \frac{\int_0^R wu r dr}{\int_0^R uu r dr} \quad (2.8)$$

Swirl number, S_v expression given by Vaidya et al., (2011)

$$S_v = \frac{\int_0^R \rho W U r^2 dr}{R \cdot 2\pi \int_0^R \rho U^2 r dr} \quad (2.9)$$

Swirl number, S_K expression given by Keshtkar (2018)

$$S_K = \frac{\int_0^R \rho (Wr) U 2\pi r dr}{\int_0^R \rho U^2 2\pi r dr} \quad (2.10)$$

where, for Equations (2.8-2.10), R is the outer radius of the annulus or cylinder, ρ is the density of the fluid, u and w are the axial and tangential velocity at the corresponding radial position, r .

In the presence of air turbulence in the inlet manifold, the mixing rate will be increased as swirling flow becomes higher (increased in swirl number), thus the fuel-air mixing chemistry will be better. A proper swirl level produced by the inlet manifold before entering the intake port and combustion chamber will help to achieve homogeneous mixing. This swirl level can be predicted and measured by employing a proper swirl number formula such as Equation (2.5) and Equations (2.7-2.10) according to the objective of each application.

2.13 Summary

This chapter discloses some of the recent findings on the study of swirl air intake to improve fuel-air mixing in the CI engine. Numerous strategies have been adopted in the air intake system to maximize the potential of the engine to achieve optimum combustion. From the fuel injection to fluid flow strategies, the objective is mainly to achieve a better homogeneous mixture during the fuel-air premixed phase in the combustion chamber. Their findings show that the improvement to the engine combustion characteristics increases the engine performance and subsequently reduces exhaust emissions. Many researchers have also reported higher brake or indicated thermal efficiency with the combustion-controlled parameters based on their investigations.

In the fluid flow strategy, various designs of air intake systems have been implemented in the engine. The list of recent patented, published or commercial swirl air intake devices as presented in Table 2.1 has shown that the current motivation in improving the engine performance is undoubtedly through the air induction method. Through the design of a swirl air device or induction pipe, inlet port and piston bowl, better combustion characteristics can be achieved with the increase of turbulence, formation of swirl and tumble motions that provide adequate mixing of air-fuel and appropriate distribution of fluid in the chamber. The swirl intensity or strength is indicated by the Swirl Number or Swirl Ratio and can be estimated based on several swirl definitions. However, several researchers also reported on the limitations of this method which may result in poor engine performance and exhaust emissions. This indicates that the effectiveness of generated swirl air through various designs of intake mechanisms needs to be further studied and investigated.

CHAPTER 3

METHODOLOGY

3.1 Introduction

The important parameters that indicate the combustion process's effectiveness are the in-cylinder pressure, ignition timing or delay, combustion duration heat release and cumulative heat release rate. In-cylinder pressure can be directly measured from the engine and other combustion parameters can be calculated from the in-cylinder pressure. Meanwhile, the ignition delay can be estimated based on the time elapsed between the start of fuel injection and the start of combustion. In evaluating the 'ram' swirled effect in a diesel engine, the peak in-cylinder pressure is observed from the experiment to understand the trend in the ignition delay by analyzing the maximum pressure rise rate besides of the heat release rate calculation. CFD simulations presented in this chapter are used to compare with the experiment data and discussed by referring to the factor of air turbulence in the combustion. The measurement method for the important engine parameters is also discussed.

Figure 3.1 presents the general research flow process, while Figure 3.2 and 3.3 show the details of the simulation and experimental process flow. The design and simulation work started with the literature review and theoretical study of the swirl air intake. Recent prior art such as patents, publications and commercial products are searched and benchmarked to produce a swirl air intake design concept. The modelling is done in Solidworks and further simulated in Ansys Fluent software. There are two phases of simulation which

start with the base swirl followed by the parametric swirl model. The parametric study runs according to the general Full Factorial DOE method and results from the post-processing are further analysed. The experimental flow chart starts with the prototype device being tested under two stages with the first stage of the experimental run planned according to the Taguchi DOE method.

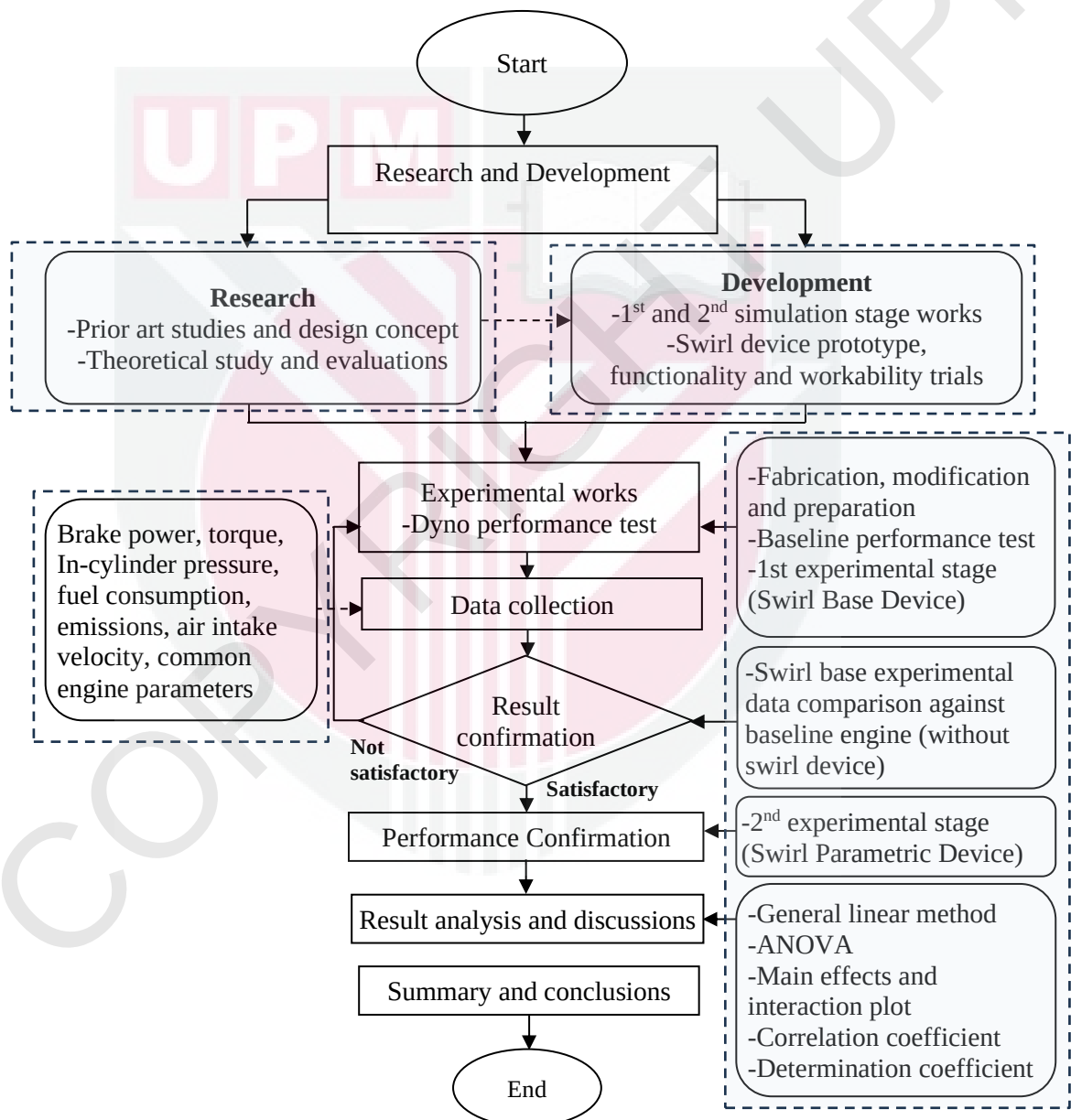


Figure 3.1: General research flow processes

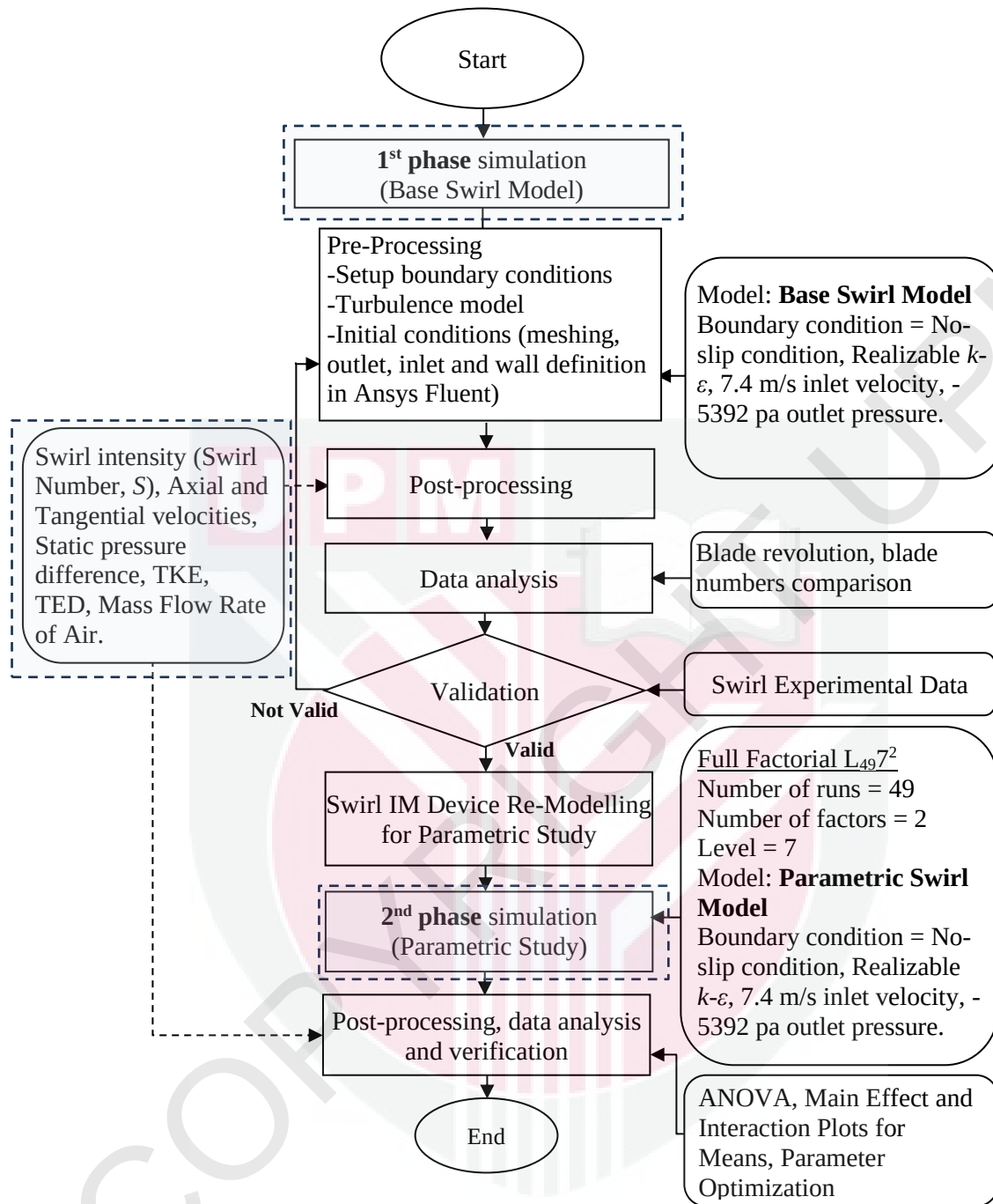


Figure 3.2: Design and Simulation Flow Chart

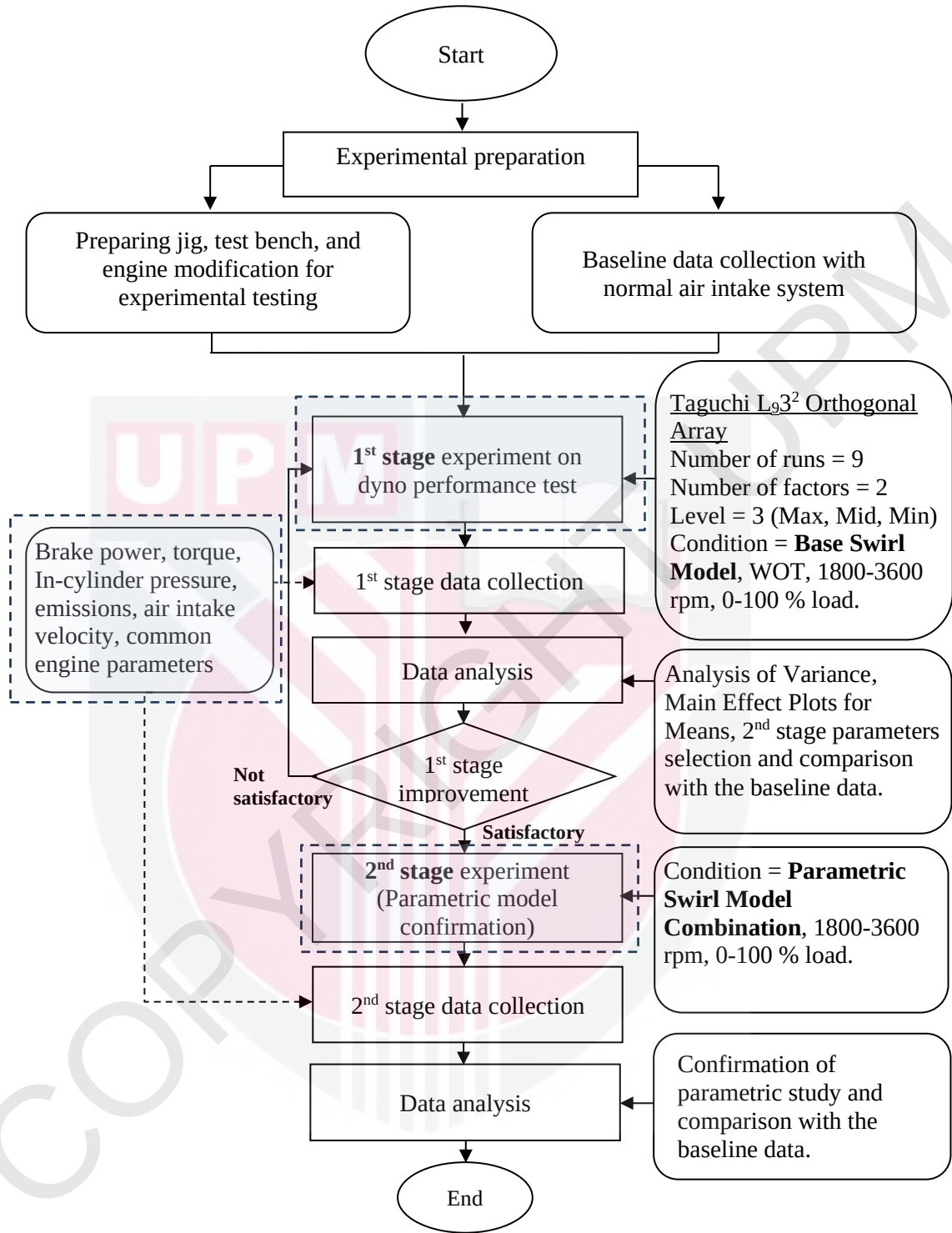


Figure 3.3: Experimental Methodology Flow Chart

3.2 Swirl Air Intake Device Design and Development

The development of a swirl air intake device consists of two phases. The first phase involves the modelling and CFD simulation of the base swirl air intake design. The second phase involves CFD simulation of the parametric study that implements the general 'full factorial' method. The parametric study aims to simulate the effect of the 'ram' together with the generated swirl produced from the base study to evaluate their combined effect on the swirl intensity and airflow characteristics.

The prototype model developed based on the first phase (base swirl) design is tested experimentally to compare with the performance of the baseline engine. For this purpose, the performance data of the baseline engine (normal air intake system) is collected for its performance level prior to the base swirl air intake testing. A range of the prototype model developed based on the second phase (parametric combinations) design is also tested to verify their effectiveness in improving the engine performance, combustion and emission due to the improved fuel-air mixing.

3.2.1 Swirl Air Intake Device Modeling

The modelling of the swirl air intake device involves the search of recent prior art such as patents, publications and other commercial products. The preliminary design concept considers the overall shape, air-guiding features, geometrical dimensions and type of swirling element. Based on the design benchmarking to the other available designs, the final swirl air intake device concept is decided for modelling and simulating works. The previous prior art

used for the design benchmarking has been discussed and described in Section 2.3 of Chapter 2. Figures 3.4 (a) and 3.4 (b) show the final design of the swirl air intake device for the base swirl model, meanwhile, Figure 3.4 (b) shows the modelling work on the blade geometry feature in the inner cylinder surface. The helix profile is chosen with a constant revolution pitch and a clearance of 2 mm is intentionally applied at both inlet and outlet sides to avoid any possible change in blade geometry or interference when it is assembled. Devices made from a polymer-based material are more prone to distortion compared to metal-based material.

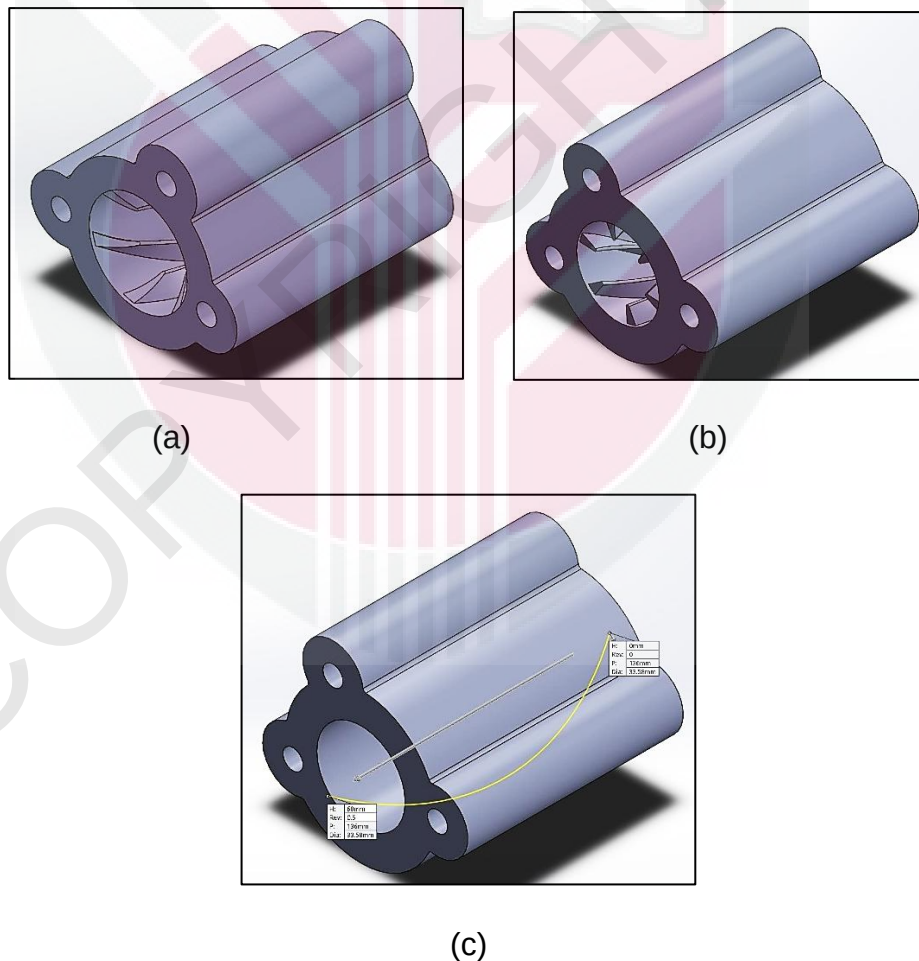


Figure 3.4: Isometric view of the swirl base 3D modelling work based on the 'Draft Angle 1.26°', 'Length 70 mm' and 'Blade Numbers 6'; (a) Inlet side, (b) Outlet side and (c) Helix profile with constant revolution pitch.

For the swirl parametric model, the modelling work is made after the engine performance test of the swirl base model with the aim of focusing on the specific design parameters that have proven to be effective in improving engine performance. Figure 3.5 below shows the model of one of the parametric designs that considers the draft angle and the length of the device as the parametric combination. The full range of the combined parameters is explained in the parametric model simulation section (Section 3.3.2). In total, there are 49 models created based on the combined parameters.

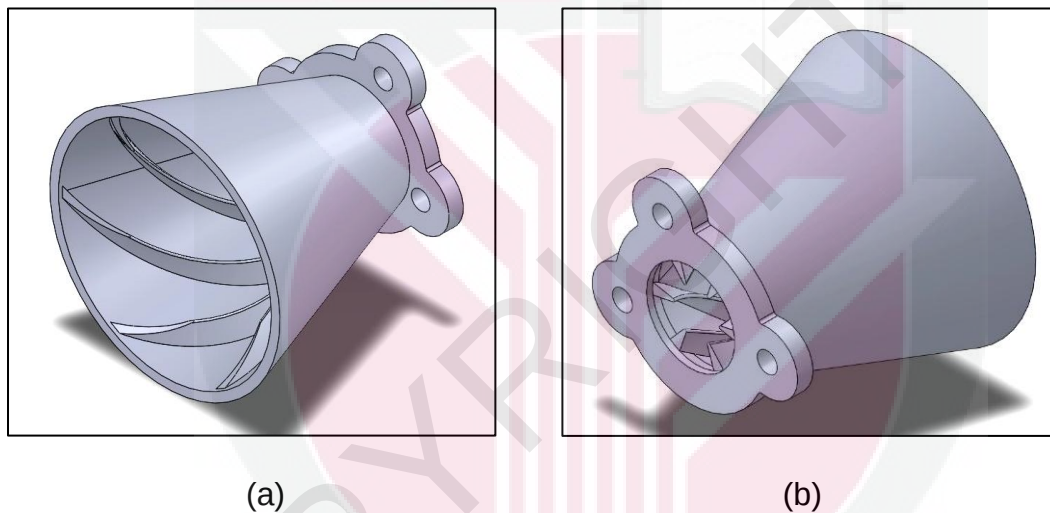


Figure 3.5: Isometric view of the swirl parametric model based on the ‘Draft Angle 18°’ and ‘Length 70 mm’; (a) Inlet side, (b) Outlet side.

CFD simulation work after the modelling stage involves the major study of the swirl formation, swirl intensity and flow velocity. Other flow characteristics such as Turbulence Kinetic Energy (TKE) and Turbulence Energy Dissipation (TED) are also observed to understand the effect of different design parameters.

The common airflow characteristics such as the mass flow rate of air and pressure difference based on the simulation are used to calculate the important engine parameters such as the volumetric efficiency that is simulated using the condition of an actual engine airflow. The details of the simulation work will be explained in the CFD simulation of swirl air intake sections (Section 3.3.1 and 3.3.2).

3.2.2 Design Characteristics

The swirl air intake device is a novel design used to guide the air taken from the air filter to the inlet port, which eventually forms the air swirl at the device's outlet through the internal blades. Unlike previous designs shown in Chapter 2, the internal cylinder of the device is designed with a draft angle with respect to the number of blades used to maintain the mass of air intake comparable with the original intake system, at the same time utilizing the air swirl to facilitate more complete fuel combustion in the engine.

The blade geometries are designed with an emphasis on promoting better air streamline in reducing the impact from the air resistance. The blade geometries are also effective with a larger positive draft angle to allow for considerably higher air flow velocity benefited by the ram effect while maintaining swirl intensity at the outlet.

Better engine operation not only requires swirl airflow to promote better air-fuel mixing, where the high air resistance causes more pressure drop which means reduced volumetric efficiency of the engine. The air resistance may also reduce the efficiency of the air swirl formed. Hence, the device is not only functioning to generate a swirl for more complete combustion. It is also made specifically to minimize air resistance due to internal obstruction.

In general, the focus of the swirl air intake manifold design is to minimize the effect of air restriction due to the inner geometry in the device to generate air swirl. With an optimized parameter of the draft angle and length, the device will further benefit from the 'ram' pressure effect that considerably increases the engine performance. Figures 3.6 to 3.11 below show the design and dimensions of the swirl air intake manifold device based on the swirl base model, while Table 3.1 shows the specification and range covered by the design.

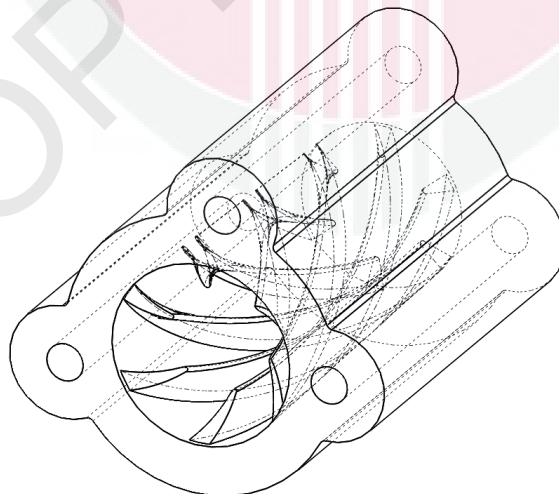


Figure 3.6: Isometric view of the Swirl Air Intake Manifold Device based on the swirl base model with nominal 0.5 revolution and 6 helical blades.

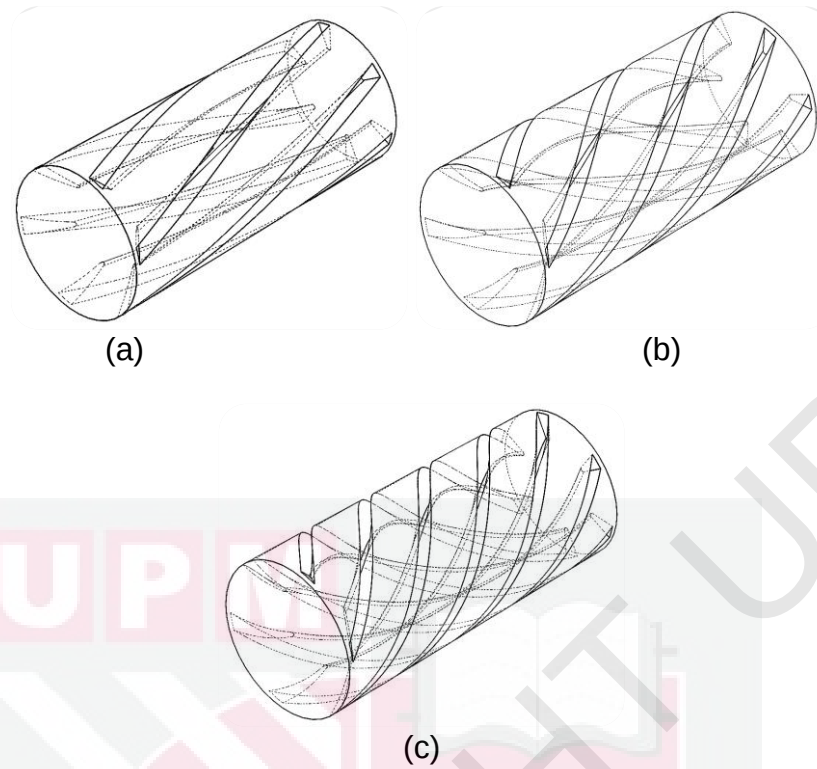


Figure 3.7: Range of Blade Revolutions; (a) 0.25 revolution, (b) 0.5 revolution and (c) 0.75 revolution.

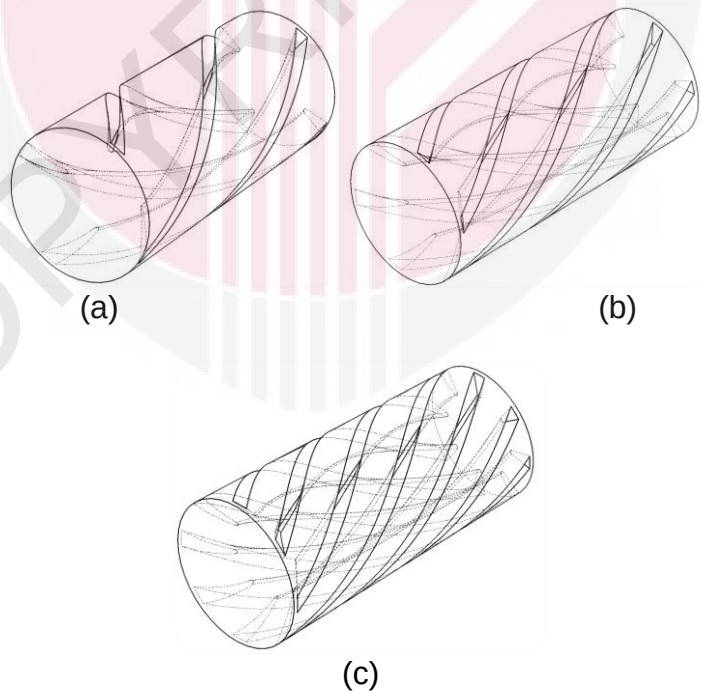


Figure 3.8: Range of Blade Numbers; (a) 4 blades, (b) 6 blades and (c) 8 blades.

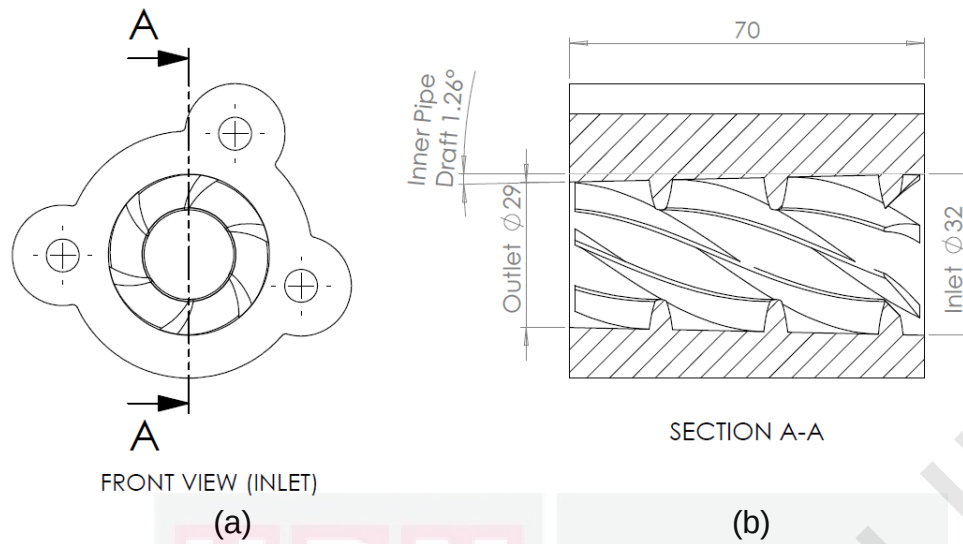


Figure 3.9: (a) Front view of the swirl base model with nominal 0.5 revolution and 6 helical blades, (b) Section A-A shows the dimensions and draft angle inside the cylinder.

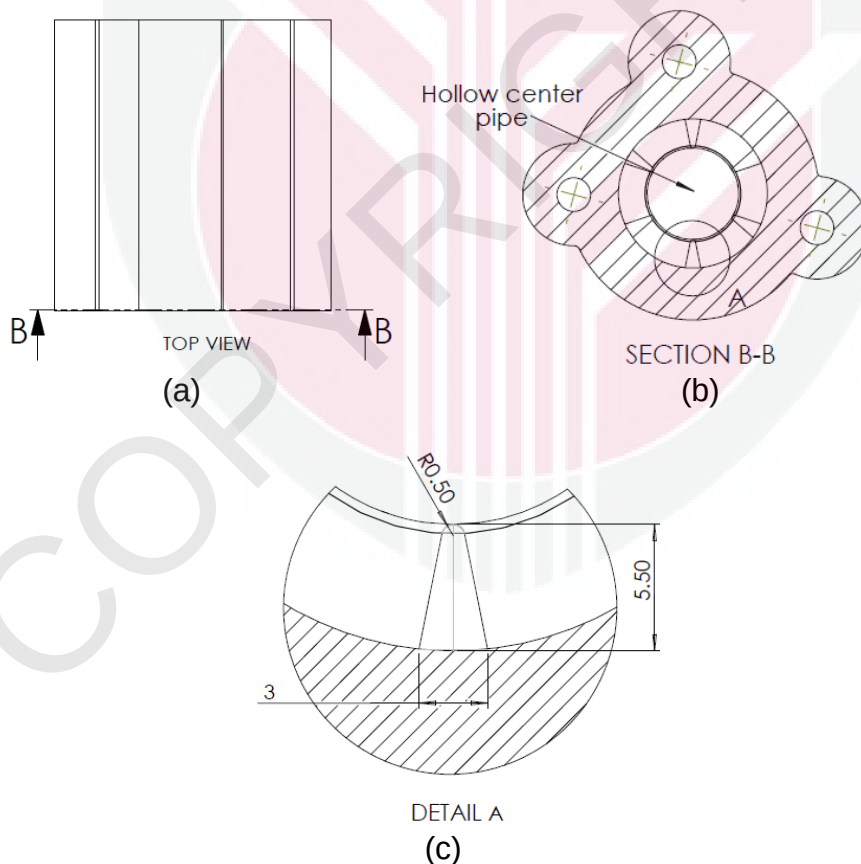


Figure 3.10: (a) Top view of the device with B-B cross-section, (b) Section B-B shows the hollow space at the centre of the cylinder and the highlight of the helical blade, and (c) shows the highlight on the dimensions of the helical blade.

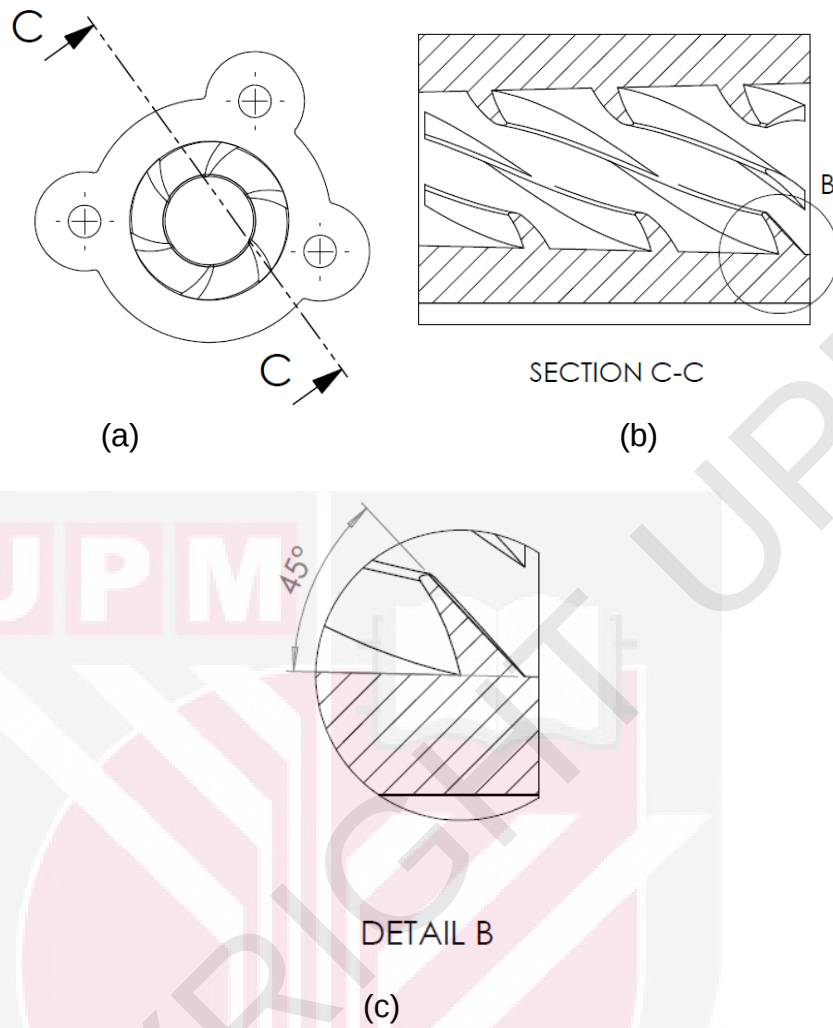


Figure 3.11: (a) Front view of the device with C-C cross-section, (b) Section C-C shows the highlight of the helical blade opening angle, and (c) shows the detail of the helical blade opening dimension.

Table 3.1: Specification and range of size and blade numbers of swirl air intake manifold device.

Parameters	Unit	Nominal value	Range value
Inner Cylinder Draft angle	degree	1.26	0.8-18
Outlet diameter (Engine intake port side)	mm	29 Follow engine size	No range (Value to be based on the engine intake port size)
Inner Cylinder width (length)	mm	70	50 - 120 [Aspect ratio (2.4137:1)]
Blade height	mm	5.5	3.9 - 9.4 [Aspect ratio (12.7273:1)]
Blade opening angle (inlet side)	degree	45	Constant
Blade peak radius	mm	0.5	0.35 - 0.85 [Aspect ratio (140:1)]
Blade base length	mm	3	2.14 – 5.14 [Aspect ratio (23.3333:1)]
Blade numbers	quantity	6	4 - 8
Blade revolution (helical)	revolution	0.5	0.25 - 0.50

The range of values as shown in Table 3.1 are decided mainly based on the aspect ratio calculation of the swirl base model and the result of simulation work. Some of the ranges are finalized after the experimental stage which verifies the effectiveness of the design parameters. Figure 3.12 shows one of the designs and dimensions of the swirl air intake manifold based on the swirl parametric model that implements a different dimension range as per Table 3.1.

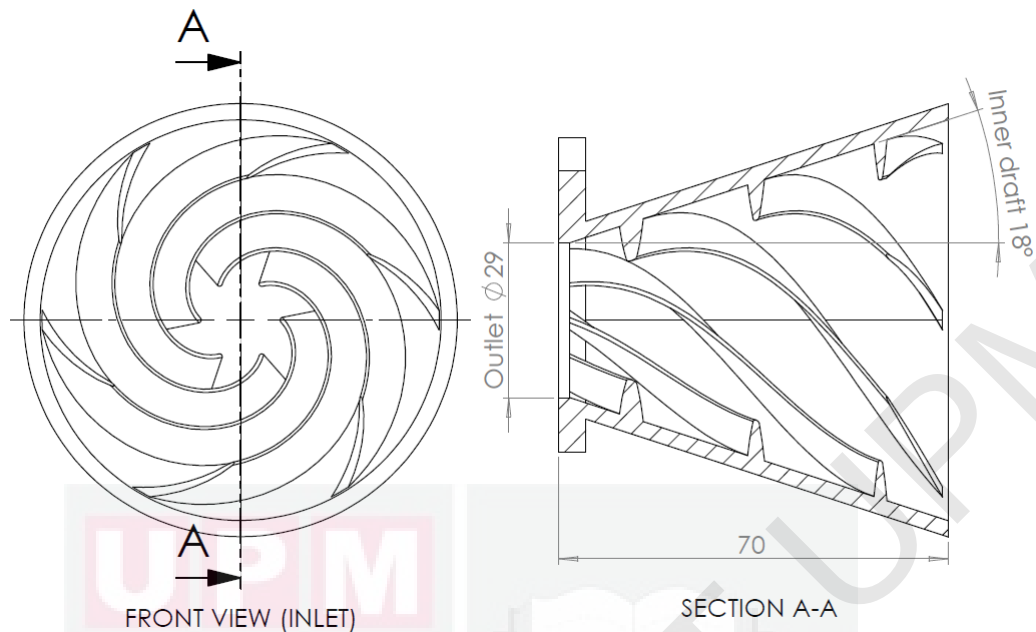


Figure 3.12: (a) Front view of a swirl parametric model with nominal 0.5 revolution and 6 helical blades, (b) Section A-A shows the dimensions and draft angle inside the model with the 'Draft Angle 18°' and 'Length 70 mm'.

3.2.3 Development of the Prototype Device

After the simulation stage, the prototype device based on the selected model is fabricated via a 3D printing facility and verified using the engine dynamometer for performance and emissions measurement. All these equipment and facilities are available in the Department of Mechanical and Manufacturing Engineering, Universiti Putra Malaysia. Figures 3.13 to 3.15 show the printing and finishing process of the prototype device using the available facilities.

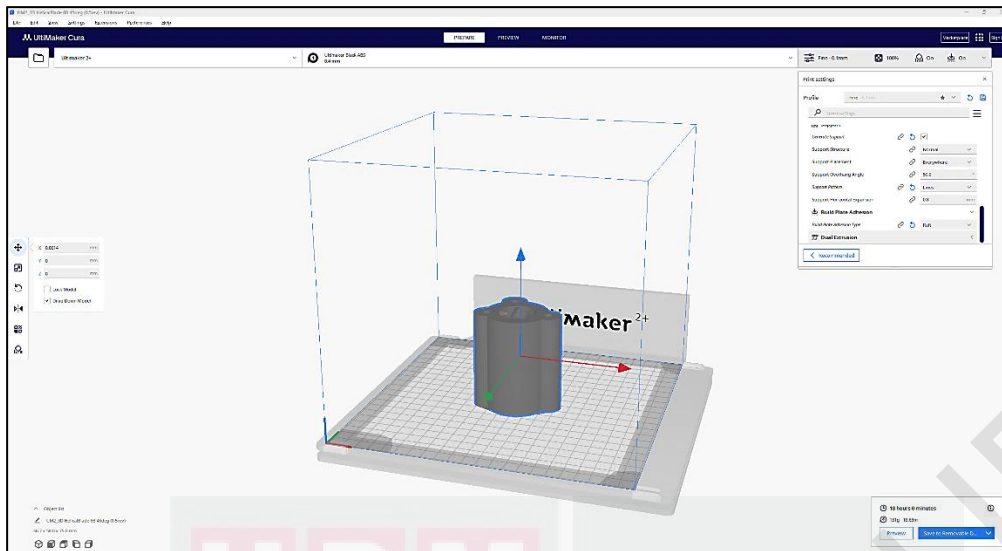
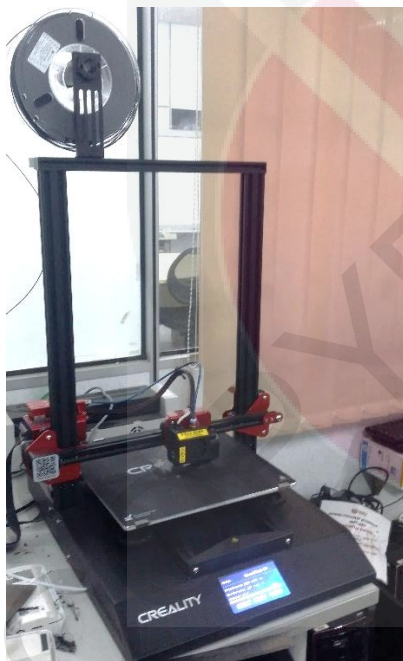


Figure 3.13: Creality Cura software GUI for 3D printing parameter setting



(a)



(b)

Figure 3.14: 3D printing process of the Swirl IM prototype device; (a) 3D printing machine setup, (b) Completed Swirl IM prototype.



Figure 3.15: Swirl IM prototype device finishing process using the ZMM SLIVEN CU 400M universal lathe machine.

The experiment runs with the prototype device made of a thermoplastic polymer-based material (ABS plastic) with a typical heat resistance of 103.9 °C (Polymaker, 2023). PolyLite™ ABS is supplied as a filament wire and works perfectly for the 3D model fabrication using a typical 3D printing instrument for research purposes. ABS plastic is the recommended material that is found to be tough, resilient and able to work within the normal engine operating temperature. Table 3.2 shows the thermo-physical properties of this material.

Table 3.2: Thermo-physical properties of PolyLite ABS plastic material

Property	Typical value
Density	1.12 g/cm ³ at 23 °C
Heat resistance	103.9 °C
Young's Modulus (X-Y)	2246.6 ± 58.2 MPa
Tensile strength (X-Y)	33.4 ± 0.6 MPa
Bending strength (X-Y)	2127.2 ± 29.9 MPa

(Source: Polymaker, 2023)

In a typical operation, when the coolant temperature reaches 80 to 90 °C, the thermostat will allow the cooling fluid to flow into the engine, and at the same time, the radiator fan will start to operate, thus maintaining the engine temperature at approximately 90 °C. Since this experiment used an air-cooled

type of engine cooling system, the engine temperature is fully monitored throughout the operation to avoid running under extreme temperatures.

The first experimental stage involved the engine performance test with the prototype of the swirl base model that implements different helical blade revolutions and a constant blade number. The second stage of the swirl air intake experiment commences when the test of the swirl base model is concluded, and the parametric model is started. The helical blade number and revolution are carried over from the first stage of the experiment with a few eliminations to the less effective swirl base model design depending on the result and evidence of both simulation and experiment works. The second stage prototype device is fabricated to verify the performance of the parametric model based on the results of the parametric study. Figures 3.16 to 3.18 show the prototype device from the first and second stages of experimental testing.



Figure 3.16: First experimental stage prototype Swirl IM device (0.25, 0.5 and 0.75 revolutions)



Figure 3.17: Second experimental stage Swirl IM prototype device (0.25 Revolution)



Figure 3.18: Second experimental stage Swirl IM prototype device (0.5 Revolution)

Table 3.3 shows the 3D printed component inspection results and measurement error between the measured and expected value using the vernier calliper. The maximum measurement error is 1 % which indicates that there is a high closeness of agreement between the measured and expected value of the 3D printed component.

Table 3.3: 3D printed component inspection and measurement error

Inspection items	Unit	Nominal	Measurement (Average)	Measurement Error (%)	Judgement
Inner Cylinder Draft angle	degree	1.26	1.25	0.79	OK
Outlet diameter (Engine intake port side)	mm	29	28.80	0.69	OK
Inlet diameter	mm	32	31.83	0.53	OK
Inner Cylinder width (length)	mm	70	70.04	0.06	OK
Blade height	mm	5.5	5.49	0.18	OK
Blade base length	mm	3	3.03	1.00	OK

3.3 Computational Fluid Dynamics (CFD) Simulation

This section presents the CFD works during the design and development stage which comprises two main simulation works. The CFD simulation is mainly conducted for swirl air intake (base study) and swirl with ram air intake (parametric study).

3.3.1 Swirl Air Intake Simulation

The base model simulation of the swirl air intake device is initially conducted to finalize the device's base geometry. Two important geometries, namely helical blade revolution and blade numbers are initially studied to identify their

effect on the swirl intensity and the airflow characteristics. The final blade parameters are decided in the base model simulation stage before further verification is done on the performance.

The simulation started with the generation of the 3D model geometry in the Ansys Workbench to initially define the inlet and outlet of the fluid flow, and the contact region between the fluid flow and the body. Figures 3.19 and 3.20 show one of the geometry models of the base swirl air intake device with 0.5 revolution in the Ansys Workbench environment. To reduce simulation time, the geometry is separately prepared in the Solidworks software and then imported to Ansys Fluent via the Ansys Workbench package for meshing.

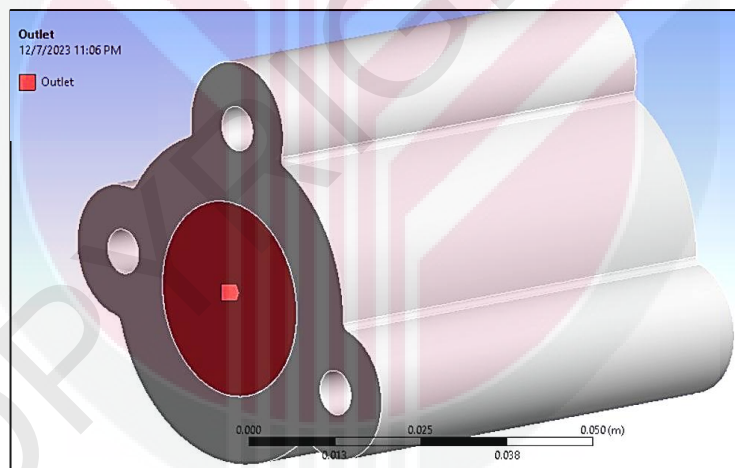


Figure 3.19: Pressure-outlet definition in the Ansys Workbench

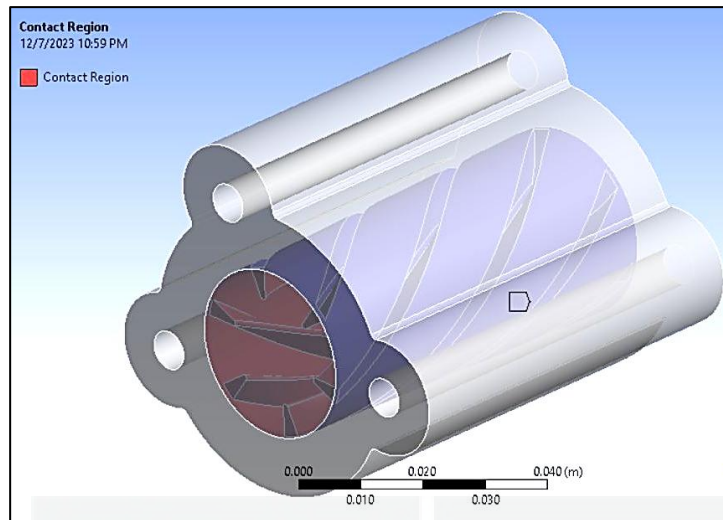


Figure 3.20: Fluid flow and wall contact region definition in the Ansys Workbench

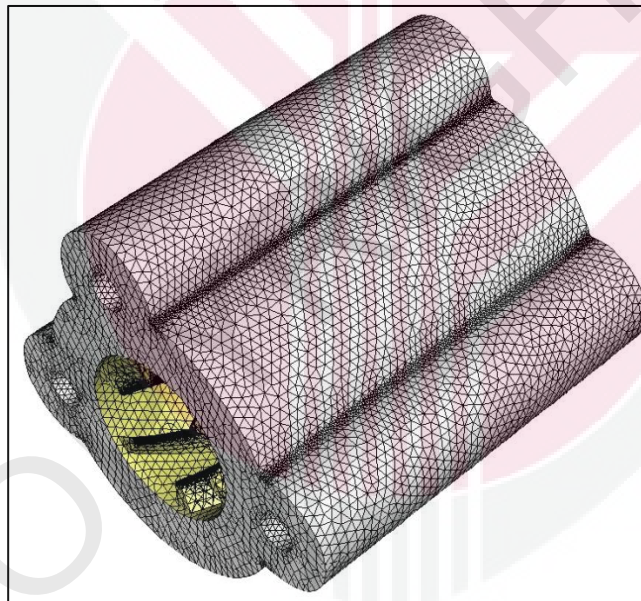


Figure 3.21: Meshing of the swirl (0.5 rev, 6 blades) base model's body region in the Ansys Fluent

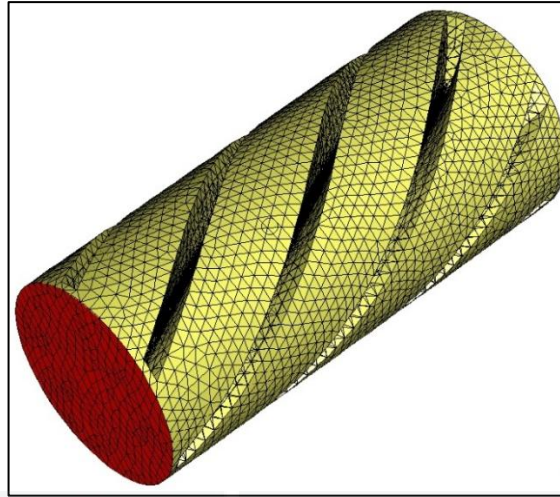


Figure 3.22: Meshing of the swirl (0.5 rev, 6 blades) base model's fluid flow region in the Ansys Fluent

As presented in Figures 3.21 and 3.22, the mesh is generated using the tetrahedral element and consists of 512,499 refined elements for the Swirl Base 2 – 0.5 revolution's fluid flow region as the mesh size is known to influence the accuracy of the simulation results. Tables 3.4 and 3.5 summarize the meshes and boundary conditions set for all the swirl base models involving the helical blade revolutions (0.25, 0.5, 0.75 revolutions) and blade numbers (4, 5, 6, 7, 8 blade numbers). The turbulence model employed in the entire simulation is the Realizable $k-\varepsilon$ with the no-slip condition for the walls. The inlet and outlet devices are set to fluid velocity and gage (vacuum) pressure respectively with the standard air properties set as the fluid medium.

Inlet air velocity is set to 7.4 m/s and represents the actual average speed of air intake velocity measured at 3600 rpm under zero load conditions as per the intended application; engine used for a generator set. The speed of air intake flow to the engine may change when it is put under different engine loads, which is not the focus part of this study. The -5391 Pa set for the gage pressure

at the outlet, where it is based on the difference between the standard atmospheric pressure and the theoretical absolute pressure calculated from the theoretical mass flow at the engine speed of 3600 rpm, engine cylinder volume (Yanmar engine capacity of 0.219 Litre), inlet air temperature assumption at room temperature (298 K) and gas constant R for dry air (287.058 J/kgK).

Table 3.4: Summary of the Swirl Base CFD model (different helical blade revolutions) boundary condition

Boundary Condition	Swirl Base 1 - 0.25 revolution	Swirl Base 2 - 0.5 revolution	Swirl Base 3 - 0.75 revolution
Mesh type	Tetrahedral	Tetrahedral	Tetrahedral
Nodes (Fluid flow region)	84,638	106,035	128,802
Elements (Fluid Flow region)	407,484	512,499	625,263
Turbulence Model	Realizable $k-\epsilon$	Realizable $k-\epsilon$	Realizable $k-\epsilon$
Velocity-Inlet (m/s)	7.4	7.4	7.4
Pressure-Outlet (Pa)	-5391	-5391	-5391
Walls	No-slip condition	No-slip condition	No-slip condition

Table 3.5: Summary of the Swirl Base CFD model (different blade numbers) boundary condition

Boundary Condition	Blade 4	Blade 5	Blade 6	Blade 7	Blade 8
Mesh type	Tetrahedral	Tetrahedral	Tetrahedral	Tetrahedral	Tetrahedral
Nodes (Fluid flow region)	73801	89942	106035	122084	137459
Elements (Fluid Flow region)	356371	434535	512499	590039	664731
Turbulence Model	Realizable $k-\epsilon$	Realizable $k-\epsilon$	Realizable $k-\epsilon$	Realizable $k-\epsilon$	Realizable $k-\epsilon$
Velocity-Inlet (m/s)	7.4	7.4	7.4	7.4	7.4
Pressure-Outlet (Pa)	-5391	-5391	-5391	-5391	-5391
Walls	No-slip condition	No-slip condition	No-slip condition	No-slip condition	No-slip condition

In the beginning stage of the simulation work, the tetrahedral mesh is initially used before transitioning to a polyhedral mesh later in a parametric study. This is done so that the fundamental behaviour of the flow using the swirl base model can be rapidly assessed. Tetrahedral meshes are generally easier to generate compared to polyhedral meshes since the swirl air intake device features a set of irregular blade geometries. The application of this mesh type in the 1st stage of simulation work is also due to the well-established meshing algorithm that will ensure the ease of mesh generation with fair mesh quality. The post-processing fluid flow involved the generation of the physics report with essential information from the flow vectors and streamlines. Important parameters such as the Swirl Number (S) are then calculated from the post-processing information.

3.3.2 Swirl with Ram Air Intake Simulation

The 2nd stage of the simulation work involved the parametric study of the swirl air intake device. The simulation for the swirl parametric model is conducted according to the General Full Factorial DOE method to simulate all possible combinations of factors and factor levels and observe the effects that may result from all ranges of the design parameters. This method offers a systematic and comprehensive approach to explore the parameter space, main effects and interaction identification. This method is capable of statistically determining the significance of factors and quantification of the effects to enhance the reliability and validity of the parametric study results. The General Full Factorial method also offers to explore the significance level of the factor effects on the response, thus enhancing the reliability and validity

of the parametric study results. For this purpose, ANOVA is used to statistically determine the significance of factors and quantification of the effects based on the established simulation plan according to the General Full Factorial method. For this section, the ANOVA analysis only focuses on the swirl intensity data since the prime objective of the swirl air intake design is to effectively form a swirl airflow through the air intake charged for better fuel-air mixing in the cylinder.

In ANOVA analysis, the P-value is calculated to measure how likely the sample results are, assuming whether the null hypothesis is true or false. The P-value ranges from '0' to '1' with a P-value less than the significant value (α -level ≤ 0.05) indicating that the factor studied has a statistically significant effect on the response observed. The α -level 0.05 is decided based on a 95% Confidence Interval (CI) following the standard industrial practice (Morato, 2012; Swanston, 2023) which means 5 out of 100 times in data estimation could be tolerable if wrong. It could also be understood differently, if the study is to be repeated 100 times, around 95 times would be expected to fall within this range.

In this DOE method, two (2) factors with seven (7) factor levels have been identified based on the geometrical design of the swirl IM device. The device 'Draft Angle' and 'Length' have been selected as the studied factors based on their impact on the airflow characteristic, while each of these factors has been set with three (3) parameter or factor levels. In the parametric simulation, one blade revolution is employed for the entire simulation which is 0.5 revolution. The decision to use one type of blade revolution in this 2nd stage of simulation

work is based on the expectation that a similar trend could be found with the other blade revolution under the similar combined parameters (draft angle and length). Table 3.6 shows the summary of the selected simulation run planning based on the designated parameters set with the general full factorial method ($L_{49}7^2$). Full simulation run planning consist of all 49 runs can be referred in the appendices.

Table 3.6: Selected simulation run of the parametric combination based on General full factorial design ($L_{49}7^2$)

Run	Draft Angle (degree)	Device Length (mm)	Responses
2	1.26	70	Studied responses (From CFD simulation post-processing data): 1. Velocity-Outlet 2. Swirl Number (Intensity) 3. Pressure-Inlet 4. Pressure Difference 5. Turbulence Kinetic Energy (TKE) 6. Turbulence Energy Dissipation (TED) 7. Mass Flow Rate of Air
7	1.26	120	
44	18	70	
49	18	120	

The simulation of the swirl parametric model is the subsequent simulation that employs the final blade parameters decided through the results of the base model simulation work as well as the model geometries based on the combination factors and factor level. Other than the mesh setting, all boundary conditions in the previous swirl base model's simulation are maintained for the swirl parametric model since the verification is still being made on the same engine conditions.

The simulation started with the generation of the 3D model geometry in the Ansys Workbench to initially define the inlet and outlet of the fluid flow, and the contact region between the fluid flow and the body. Figures 3.23 and 3.24 show one of the geometry models of the parametric study in the Ansys Workbench environment with the defined inlet and outlet sides of the fluid flow, and the contact region between the fluid flow and the body

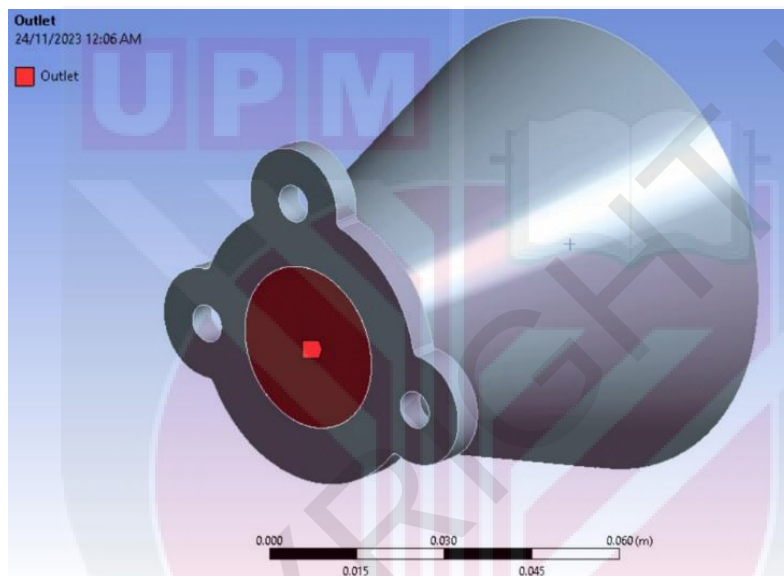


Figure 3.23: Pressure-outlet definition for parametric model (Simulation Run 44 – Draft Angle 18°, Length 70 mm) in the Ansys Workbench

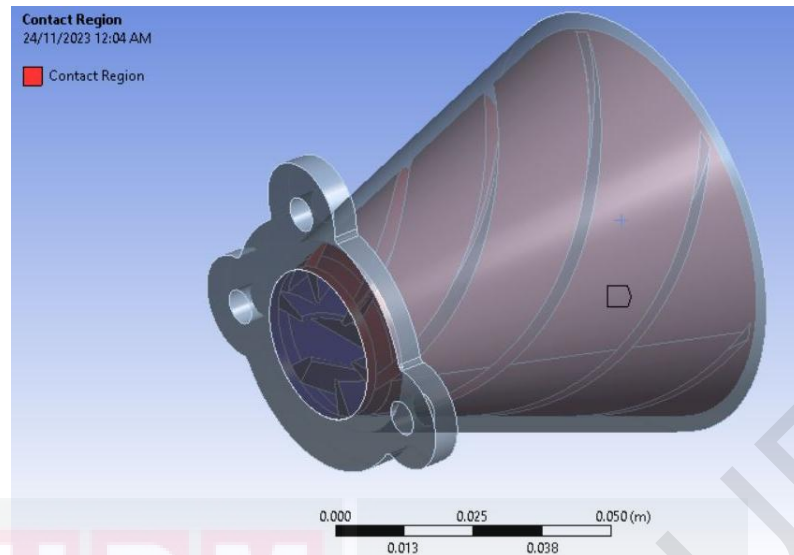


Figure 3.24: Fluid flow and wall contact region definition for parametric model (Simulation Run 44 – Draft Angle 18°, Length 70 mm) in the Ansys Workbench

In this simulation stage, the mesh is generated using the polyhedral element for the entire parametric model's simulation. Polyhedral meshes are capable of providing better element quality and representation of complex geometry shapes compared to tetrahedral meshes. In the parametric study, the blade and inner body geometry shapes are more irregular with 49 possible 'Draft Angle' and 'Length' combinations, thus, expected to produce a more complex flow pattern and behaviour. Therefore, to capture more accurate flow behaviour of the additional geometric complexities, the polyhedral element is chosen. This type of mesh also often requires fewer elements to represent the same geometry, resulting in faster simulation runtimes. Simulating 49 different combinations under the general full factorial method is obviously time-consuming, thus employing this type of mesh is highly recommended. Figures 3.25 and 3.26 show the polyhedral mesh element on the body and fluid flow region respectively with 836,956 refined elements generated under the

simulation run 44 fluid flow regions. Table 3.7 summarizes the meshes and boundary condition set for the selected range of the swirl parametric models involved (Simulation run 2, 7, 44 and 49) with 0.5 blade revolution.

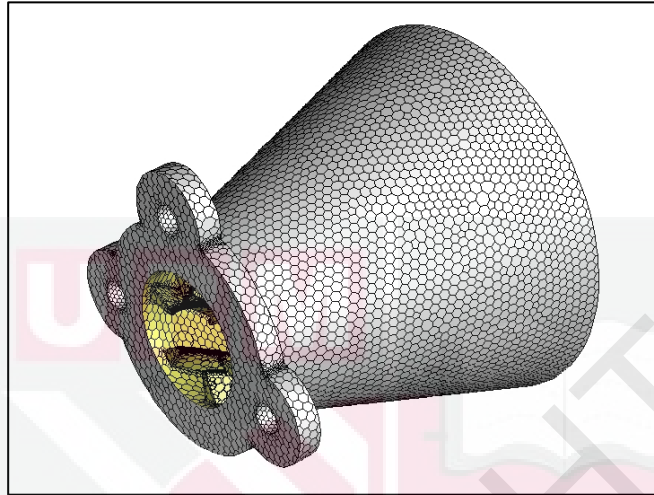


Figure 3.25: Meshing of the swirl parametric model's body region (Simulation Run 44 – Draft Angle 18°, Length 70 mm) in the Ansys Fluent

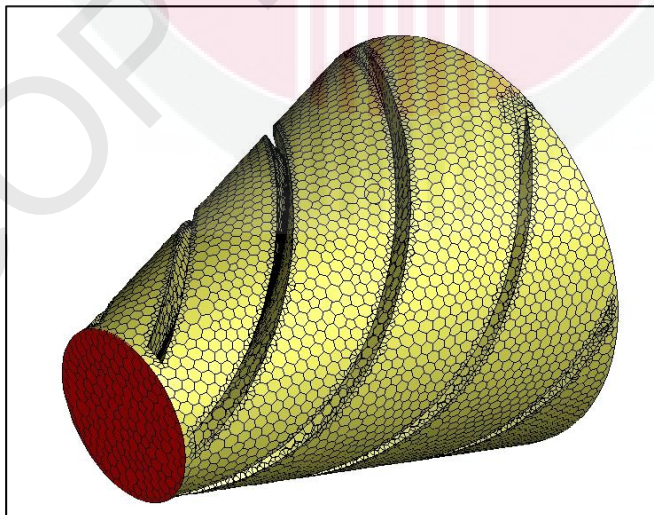


Figure 3.26: Meshing of the swirl parametric model's fluid flow region (Simulation Run 44 – Draft Angle 18°, Length 70 mm) in the Ansys Fluent

Table 3.7: Summary of the Swirl Parametric CFD model selected range boundary condition

Boundary Condition	Simulation 2 – D/Angle 1.26°C Length 70 mm	Simulation 7 - D/Angle 1.26°C Length 120 mm	Simulation 44 - D/Angle 18°C Length 70 mm	Simulation 49 – D/Angle 18°C Length 120 mm
Mesh type	Polyhedral	Polyhedral	Polyhedral	Polyhedral
Nodes (Fluid Flow region)	1,355,121	2,375,339	2,840,161	7,565,677
Elements (Fluid Flow region)	484,127	834,994	836,956	1,982,309
Turbulence Model	Realizable $k-\epsilon$	Realizable $k-\epsilon$	Realizable $k-\epsilon$	Realizable $k-\epsilon$
Velocity-Inlet	7.4 m/s	7.4 m/s	7.4 m/s	7.4
Pressure-Outlet	-5391 Pa	-5391 Pa	-5391 Pa	-5391 Pa
Walls	No-slip condition	No-slip condition	No-slip condition	No-slip condition

3.4 Experimental System and Techniques

In the beginning, the engine is modified at the cylinder block, cylinder head, intake port, flywheel and flywheel cover to be equipped with several important sensors for data measurement. The intake air pipe also is specially fabricated to assist with the intake air velocity measurement of the swirl air intake device. After the modification is completed, the engine setup will be tested for its readiness by confirming with the engine manufacturer data before the actual experimental procedure. Confirmation of the manufacturer data is done to ensure that the modified engine performance and current dynamometer setup can produce performance results comparable to the original engine design.

3.4.1 Engine Test Bench

The engine involved in the experimental study is an air-cooled, four-stroke, vertical single-cylinder, 0.219-litre direct injection engine. Yanmar L48N engine runs on a neat diesel fuel which provides the maximum rating power output of 3.5 kW and torque output of 9.7 Nm at 3600 rpm. A complete range of specifications of the engine is shown in Table 3.8 below. Meanwhile, Figure 3.27 shows the performance and specific fuel consumption curves of a new L48N Yanmar engine provided by the manufacturer.

Table 3.8: Test engine specification

Engine Specification	Details
Model	Yanmar L48N
Type	4-stroke, Vertical Cylinder, Air-Cooled Diesel Engine
Combustion System	Direct Injection
No. of Cylinders	1
Bore x Stroke	70 x 57 mm
Displacement	0.219 L
Compression Ratio	20.1 ± 0.5
Maximum Rated Output (Brake Power)	3.5 kW at 3600 rpm
Maximum Rated Output (Brake Torque)	11.3 Nm at 2800 rpm
Fuel Injection Timing	16.5 ° bTDC

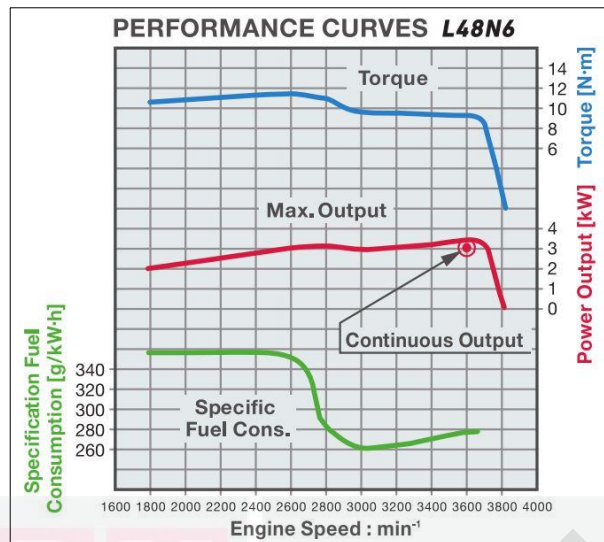


Figure 3.27: Performance and specific fuel consumption curves of L48N Yanmar diesel engine
(Source: Yanmar, 2017)

Figure 3.28 shows the actual engine fully equipped with relevant sensors, mounted on a test bed and secured by a direct coupling from the dynamometer to the shaft. Meanwhile, Figure 3.29 shows the full schematic diagram of the engine experimental setup.

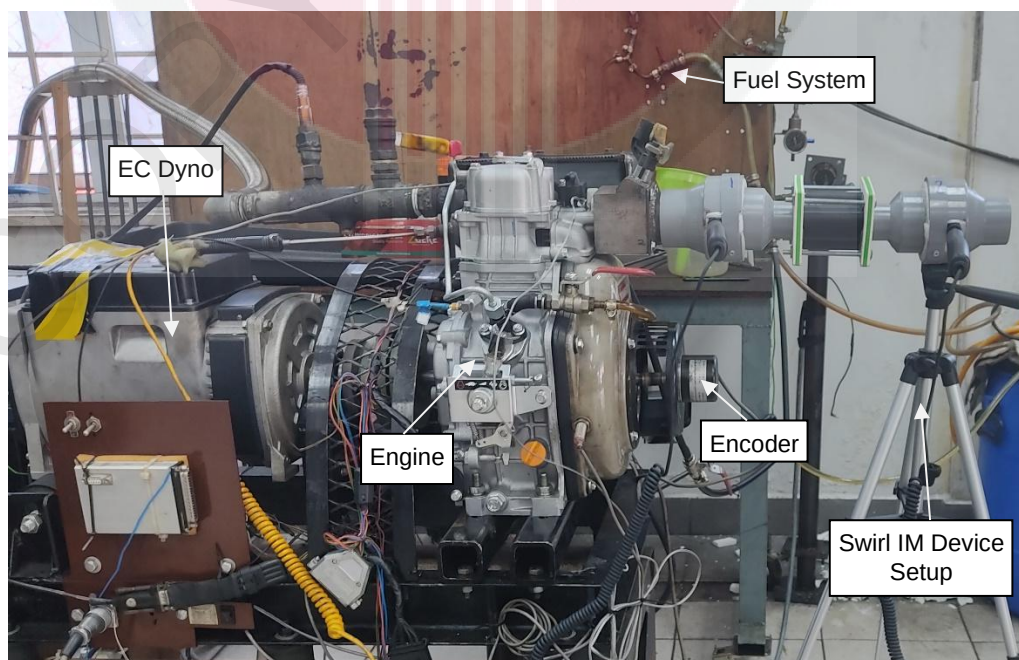


Figure 3.28: Actual test engine layout

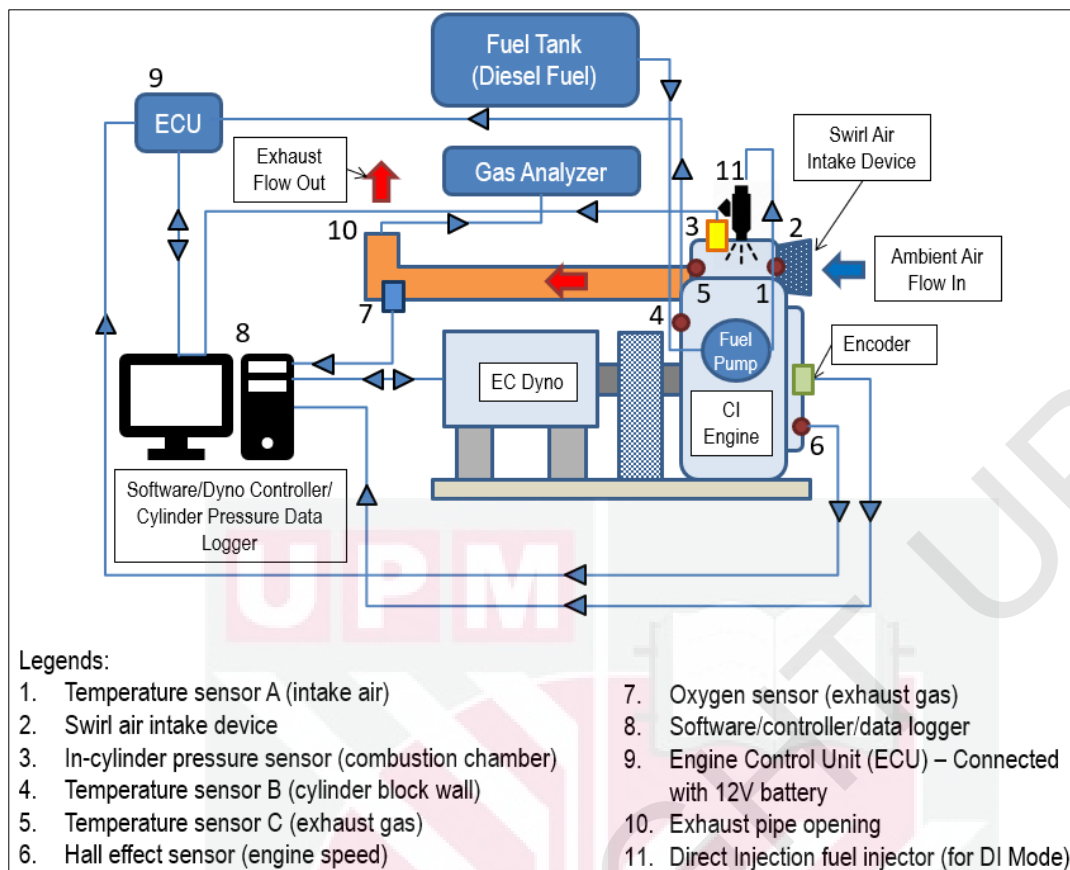


Figure 3.29: Schematic diagram of Yanmar engine test system

The experiment involved a single-cylinder 4-stroke Yanmar 0.2 litre engine running with pure or neat diesel and attached to an eddy current (EC) dynamometer. The performance test is conducted at the Thermodynamics Laboratory located at the Faculty of Engineering, UPM Serdang.

The dynamometer used in the experiment is a generator-type ‘eddy current’ dynamometer. It is suspended on bearings and the torque is measured by a load cell as shown in Figure 3.30 (a). The power from the engine rotates the rotor of the generator, while the rotor field is controlled by the dynamometer current. As more current is provided, the rotor magnetic field becomes stronger to provide more load to the engine. The Eddy Current BD-G10kW Bench Dynamometer can absorb a maximum power of 10 kW, a maximum torque of

22 Nm and a maximum speed of 4,500 revolutions per minute. For the dynamometer control and data recording, the dynamometer runs with DynoMon-5 Monitoring System by Focus Applied Technologies (FAT). Figure 3.30 (b) shows the dynamometer control system, while Table 3.9 summarises the specification of the dynamometer.

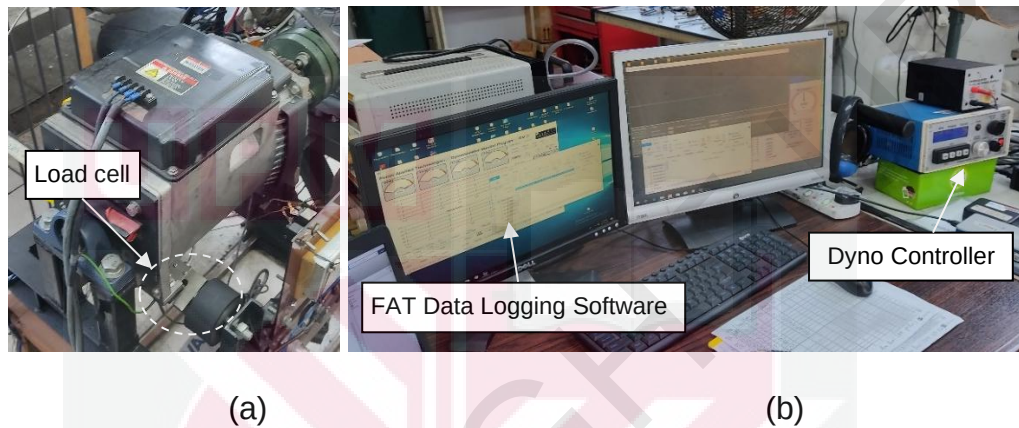


Figure 3.30: (a) Dynamometer BD-G10kW, (b) Controller system

Table 3.9: Dynamometer specification

Dynamometer Specification	Details
Mechanical Power	10 kW maximum
Torque	22 Nm maximum
Speed	4500 rpm maximum (direct coupling)
Electrical Power	5 kW maximum
Voltage output	300 Volt maximum
Current	25 Amp maximum
Dimension	1200 x 490 x 400 mm
Weight	65 kg

3.4.2 Engine Fuel and Cooling System

The engine test bench uses a manual fuel volumetric measurement method to measure fuel consumption. A stopwatch is used together with the burette to measure the time taken to drain a specific volume throughout the engine performance test. Figure 3.31 shows the schematic diagram of the test engine fuel system. For the cooling system, this engine is air-cooled utilizing a cooling fan. The cooling system consists of a fan on the flywheel, which means the engine uses the forced air by the flywheel fan without any coolant fluid.

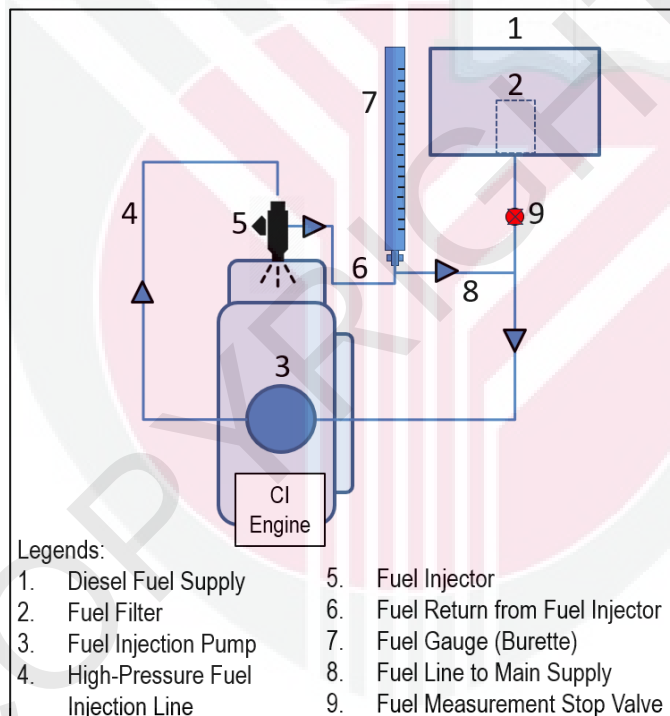


Figure 3.31: Schematic diagram of the test engine fuel system

3.4.3 Exhaust Emissions Measurement

The exhaust gas emissions are measured using the EMS 5002 Portable Exhaust Gas Analyzer. There are five emission elements measured by the equipment; Hydrocarbon (HC), Carbon Monoxide (CO), Carbon Dioxide (CO₂),

Oxygen (O₂) and Nitrogen Oxide (NO_x). CO and HC are measured by a non-dispersive infrared sensor (NDIR) while NO and O₂ are measured by an electrochemical sensor (Huang et al., 2019). The gas analyzer is equipped with a measurement probe that must be placed in the tailpipe for exhaust gas sampling. Figure 3.32 shows the interface of the portable gas analyzer that works in real-time for quick monitoring, while Table 3.10 summarises the range of specifications of the equipment.

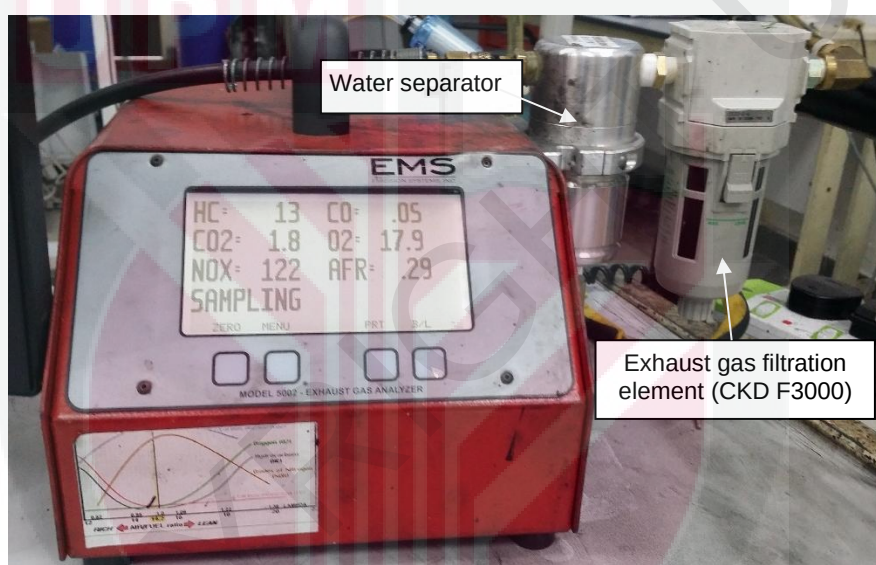


Figure 3.32: EMS portable exhaust gas analyzer used in the study

Table 3.10: The specification of the exhaust gas analyzer

Exhaust Gas Analyzer Specification	Details
Model	EMS-5002
Power	10 – 16 Volt DC
Warm-up	Less than 5 minutes
Ambient condition	2 °C - 45 °C
Range	HC: 0-2000ppm CO: 0-10% CO ₂ : 0-20% O ₂ : 0-25% NO: 0-5000ppm
Accuracy	HC: 4ppm CO: 0.06% CO ₂ : 0.3% O ₂ : 0.1% NO: 25ppm

3.4.4 Air/Fuel Ratio Measurement

The measurement of the air-fuel ratio is done by applying the lambda sensor that must be installed in the exhaust pipe. Bosch LSU 4.2 Wideband O₂ Sensor measures the air-fuel ratio (AFR) or Lambda value derived from the oxygen content of the exhaust gases. The measurement of lambda shown by the lambda display (display model MTX-L) is the actual air-fuel ratio over the stoichiometric ratio. This lambda measurement of '1' equates to the air-fuel ratio of 14.6 for a typical diesel engine. When lambda is less than 1 the engine runs 'rich', while greater than 1 means it runs on a lean mixture. This instrument provides a means to measure the actual air-fuel ratio or lambda in the engine in operation directly from the exhaust.

This lambda sensor is a wideband type sensor which has been developed to provide higher accuracy in measuring the air-fuel ratio over a broader range of operating conditions compared to a narrowband sensor which can measure only between approximately 14:1 to 15:1 of air-fuel ratio (Baker, 2019). According to Baker (2019), a wideband air-fuel ratio sensor is preferred by the industry due to the ability to reach the optimum operating temperature faster by the heater channel. Figure 3.33 shows the setup of the lambda sensor, while Table 3.11 summarises the range of specifications of the instrument. To properly use this sensor with an accurate reading, re-calibration must be done every time the sensor is dismantled from the exhaust pipe.

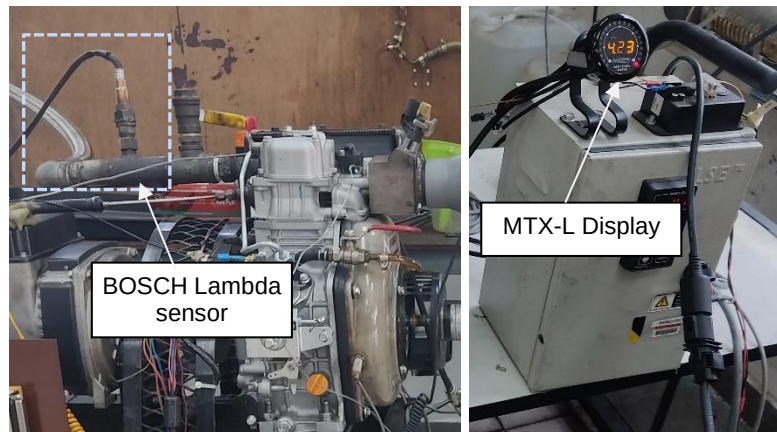


Figure 3.33: Bosch lambda sensor on the test engine and MTX-L display

Table 3.11: Range of specification for lambda sensor

Lambda Sensor Specification	Details
Model	Bosch
Type	LSU 4.2 Wideband
Temperature range	Maximum 930 °C
Mounting torque	Maximum 60 Nm
Accuracy (Lambda value)	1.016 ± 0.007 1.70 ± 0.05

3.4.5 Air Velocity Measurement

Actual air velocity is measured using an anemometer installed within the air intake system. Figures 3.34 and 3.35 show the installed condition of the anemometer in the engine setup, while Table 3.12 summarises the range of specifications of the anemometer. Figure 3.36 shows the vane anemometer set used for the air velocity measurement. For the measurement of air velocity with the swirl air intake device, a special apparatus is fabricated to properly mount the anemometer before and after the swirl air intake device to measure the air velocity at the inlet and outlet of the device.

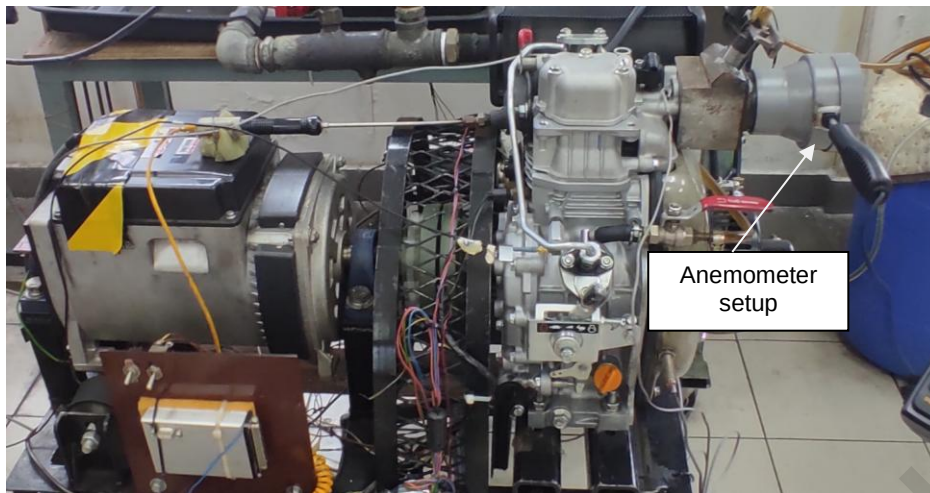


Figure 3.34: Anemometer setup for normal air intake

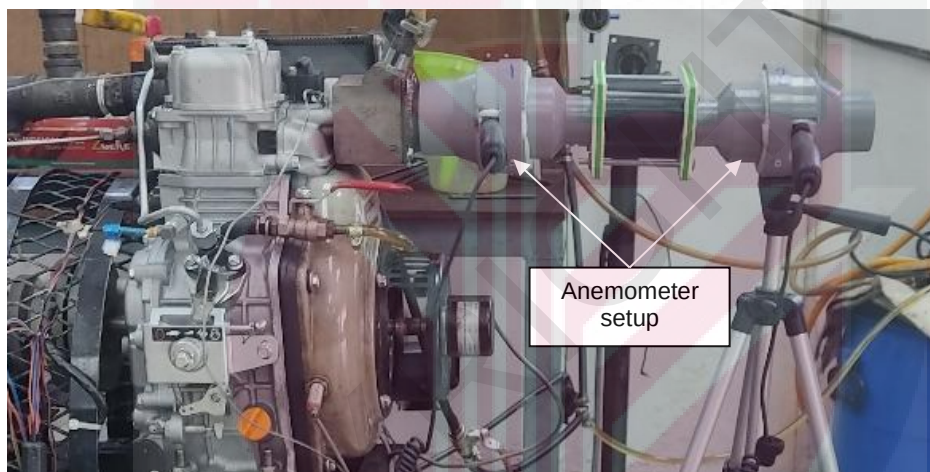


Figure 3.35: Anemometer setup for swirl air intake



Figure 3.36: Smart Sensor Vane Anemometer AS856 Set

Table 3.12: Range of specification for anemometer

Anemometer Specification	Details
Model	Smart Sensor
Type	AS856
Velocity range	0 to 45 m/s
Velocity permissible error	10 m/s $\pm$ 0.01

3.4.6 Data Acquisition System

The instrumentation of the Yanmar L48N diesel engine in the UPM Thermodynamic Laboratory is mainly equipped with the Focus Applied Technologies (FAT) data acquisition system (DAQ) and computer data logging software. In addition to the dynamometer and load cell connection, a K-type thermocouple that measures the engine temperature is also channelled to the FAT DAQ. The thermocouple used to measure the engine temperature is installed at the engine's external cylinder wall connected to the FAT DAQ channel, where the measurement is displayed in a FAT ECU software GUI. The engine temperature measurement works together with the FAT ECU module that will provide a 'MIL' indicator on the ECU GUI once the excessive engine temperature is detected. Figures 3.37 shows the FAT DynoMon-5 GUI that is used to control the engine operation, while Figure 3.38 shows the FAT ECU GUI for the monitoring of the engine speed and temperature, and Combustion Analyser for in-cylinder pressure and engine knocking monitoring.

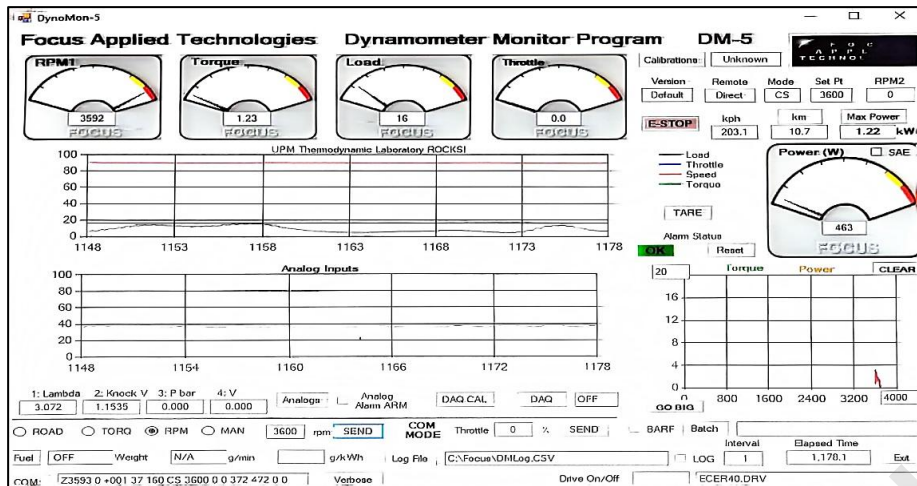


Figure 3.37: DynoMon-5 Data Logging Software GUI

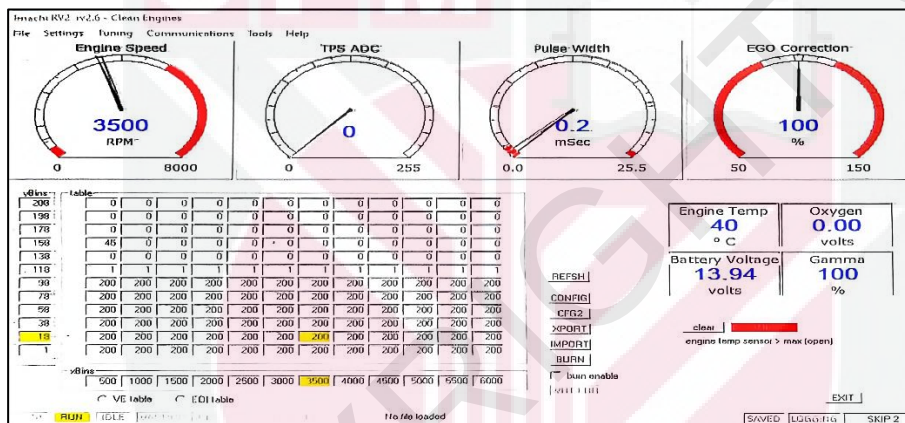


Figure 3.38: ECU Software GUI

The other thermocouples were used for the air heater controller box to measure the temperature at the intake port, and the UNI-T thermometer instrument for the temperature at the exhaust port. Since the air and gas temperature values are largely affected by the heat dissipation due to the length and material of the connecting pipes, the mounting position is set at the port area to ensure the accuracy of recorded temperatures. Table 3.13 summarises the typical range of specifications of the thermocouple used in the experiment. The provided range refers to the ASTM E230/E230M-17 standard which is the latest revision and an active ASTM document as of 2023.

Table 3.13: Typical range of specification for thermocouple

Thermocouple Specification	Details
Type	K
Temperature range	-270°C to 1372°C
Limit of error	Standard: ± 2.2°C or 0.75% above 0°C ± 2.2°C or 2.0% below 0°C
Material	Nickel-Chromium versus Nickel-Aluminum

(Source: ASTM International, 2017)

In order to measure the cylinder pressure, a pressure transducer is needed to assess the inner part of the combustion. Optrand Fiber Optic Pressure Transducer was used and a hole size of 5mm diameter was drilled through the cylinder head until reached the combustion chamber wall. The Optrand sensor can measure the pressure range of up to 345 bar (5000 psi) with an output voltage signal of up to 4.5 Volt. It is capable of withstand extreme and harsh conditions of higher temperature in combustion up to 450 °C (Optrand, 2017). Figure 3.39 shows the position of the pressure sensor on top of the combustion chamber through the cylinder head, while Table 3.14 summarises the range of specifications of the Optrand sensor.

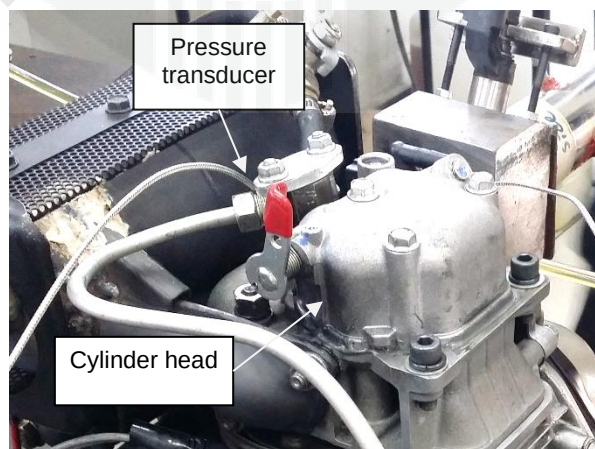


Figure 3.39: Pressure transducer location on top of the combustion chamber through the cylinder head.

Table 3.14: Range of specification for pressure sensor

Pressure Sensor Specification	Details
Model (Model No.)	Oprand (D52294-R41)
Thread Type	M5x0.5 Flange Face
Size range	Miniature size
Pressure range	0 – 345 bar (0 – 5000 psi)
Sensitivity	0.82 mV/psi @200 °C
Temperature range	Up to 450 °C

(Source: Oprand, 2001, 2017)

The measurement of a crank angle-based data result related to a combustion cycle needs a crank angle encoder. The pressure sensor must be set with the mapping of the crankshaft position or crank angle ($^{\circ}\text{CA}$), and this can be achieved by setting the crankshaft angular and TDC relationship with the encoder. A coupler is fabricated for this purpose to mount the encoder directly to the crankshaft flywheel free end position as shown in Figure 3.40.

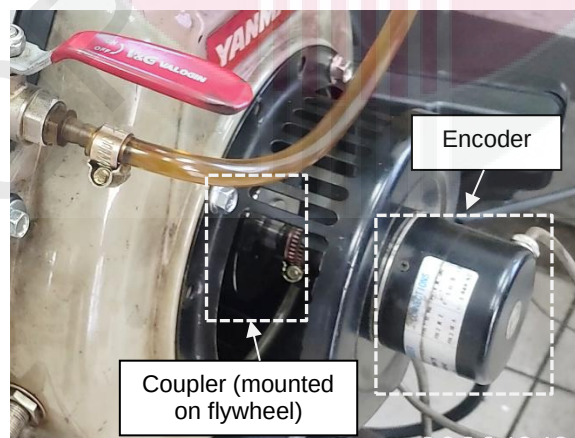


Figure 3.40: Coupler position to encoder mounted on the flywheel

For the initial setup, an oscilloscope is required to read the position signal since the encoder uses a voltage signal to indicate one complete cycle. During the

engine performance test, the mapped data from the pressure sensor and encoder is received by a dedicated DAQ system which is then connected to the Combustion Analyzer software for combustion analysis as shown in Figure 3.41.

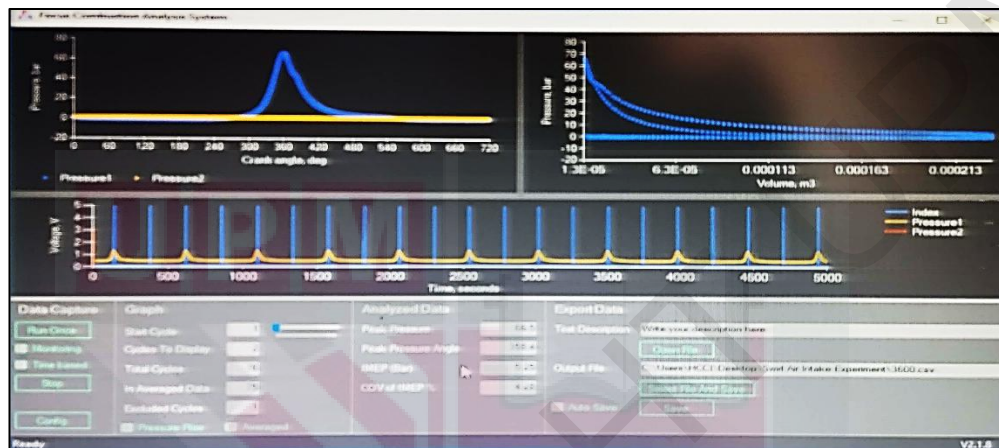


Figure 3.41: Combustion analyser software GUI

This engine is equipped with two types of vibration sensors. The first sensor is the Denso knock sensor which is installed at the nearest location close to the combustion chamber. The suitable position is at the top side of the cylinder block. The Dytran triaxial accelerometer is the second sensor which is positioned at the front side of the cylinder block close to the combustion chamber. Figure 3.42 shows the installed condition of both sensors on the cylinder block, while Table 3.15 summarises the range of specifications for both sensors.

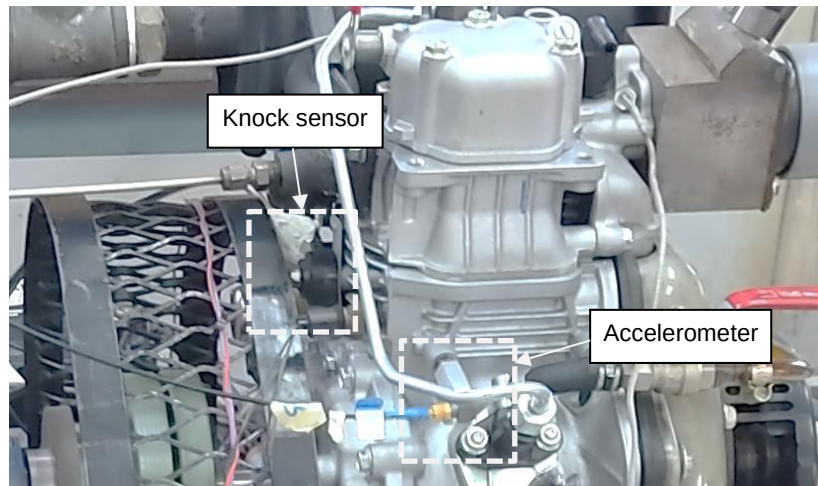


Figure 3.42: Knock sensor and accelerometer on the test engine

Table 3.15: Vibration sensor range of specification

Sensor Type	Items	Details
Knock Sensor (Monitoring only)	Model	Denso
	Part Number	89615-20090/ 171000-1052
	Application	Commercial engine (Perodua/Toyota)
Accelerometer (Data Collection)	Model	Dytran
	Type	3023A2 Triaxial Accelerometer
	Sensitivity	10mV/g Nominal

The main vibration analysis is taken from the Dytran triaxial accelerometer due to more accuracy in the measured vibrational acceleration at the x, y and z-axis directions. The signal from the sensor is processed by the NI-DAQ which is connected to the Matlab® software for data collection. Meanwhile, the knock sensor is a separate sensor used for monitoring purposes in the event of engine knocking. The sensor's voltage signal is processed by another DAQ system connected to the Combustion Analyzer software mapped with the crankshaft position (°CA) for accurate interpretation. The engine should stop

operating if the engine knocking event is detected by the knock sensor to avoid any damage.

3.4.7 Estimation of Measurement Uncertainty

The estimation of uncertainties for all the measurements is essential since the physical quantities in every measurement are subject to uncertainties. A measurement result is only complete if the measurement is accompanied by a statement of the uncertainty (Bell, 2001). Since real measurements are never made under perfect conditions, errors and uncertainties may come from the measuring instrument, the sample being measured, the process of measurement and other factors such as imported uncertainties from the previous calibration work.

According to the United Kingdom Accreditation Service (2022), there is no single standard to construct a measurement model to estimate the uncertainty. This is because the process is normally dictated by the nature of the information available in the measurement system, whereby some inputs are subjected to the knowledge of the person measuring in addition to the uncertainty from all the other physical quantities. Although there is no universally accepted model for a specific measurement process, however, to guide the current requirements within the ISO/IEC17025 standard, there are established rules for how to calculate an overall estimate of uncertainty from all the uncertainty sources.

In calculating the uncertainties of all the measurements during the engine experimental stage, the sources of uncertainty in the measurement are identified with the estimated uncertainty obtained from each source and are combined according to the established rules to give an overall combined uncertainty. Basically, two different types of uncertainty need to be measured; 'Type A' and 'Type B' evaluations. Type A estimation is taken from the repeatability of the actual evaluation of an instrument or of measurement results which needed at least 3 repeated measurements. For Type B estimate, the instrument or product tolerance, accuracy, calibration or uncertainty records from the manufacturer are used.

The individual uncertainties from Type A and Type B evaluations are calculated to form the combined standard uncertainty. For the measurement model that has a different unit of expression, relative standard uncertainty is used to estimate the combined uncertainty. Subject to the widely applied Confidence Level (CL) of 95% (Bell, 2001), the coverage factor k of 1.96 or 2 is used to get the expanded uncertainty as the final uncertainty result.

The calculation of the measurement uncertainty may employ the general formulas or expressions shown in Equations 3.1 to 3.6 below where the basis of defining uncertainty is by estimating the measurement's standard deviation.

The measurement uncertainties for this experiment are summarised in Chapter 5, meanwhile, the uncertainty formulas dedicated to each measurement model are shown in Appendix C.

Standard deviation, s

$$s = \sqrt{\frac{\sum_{i=1}^n (x_i - \bar{x})^2}{n - 1}} \quad (3.1)$$

Where,

x = Measurement value

$\bar{x}$ = Mean of the measurement values

n = Number of measurements

Standard uncertainty, u

$$u = \frac{s}{\sqrt{n}} \quad (3.2)$$

Combined standard uncertainty, u_c

$$u_c = \sqrt{u_1^2 + u_2^2 + \dots + u_n^2} \quad (3.3)$$

Where,

u_n = standard uncertainties of the same unit expression

For a measurement result that employs a certain formulation as shown in Equation 4, then Equation 5 is used for a combined uncertainty formula.

$$y = \frac{x_{2-n} \cdot x_{3-n}}{x_{4-n} \cdot x_{5-n}} \quad (3.4)$$

$$u_c(y) = y \cdot \sqrt{\left(\frac{u(x_1)}{x_1}\right)^2 + \left(\frac{u(x_2)}{x_2}\right)^2 + \dots + \left(\frac{u(x_n)}{x_n}\right)^2} \quad (3.5)$$

Where,

y = Mean of the repeated measurements

$\frac{u(x_n)}{x_n}$ = Relative standard uncertainty (unitless)

Expanded uncertainty, U_e

$$U_e = k u_c \quad (3.6)$$

Where,

k = Coverage factor at any Confidence Level (e.g. $k=1.96$ at 95% CL)

3.4.8 Engine Performance Characteristics

The level of diesel engine performance is mainly affected by the engine geometries, condition of air intake charging, properties of fuel supplied and the engine components requirement. The engine torque T (N.m.) is an output parameter from the engine measured with a dynamometer, hence also referred to as the engine Brake Torque. By using the engine torque output, the engine Brake Power BP (kW) is calculated by the formula given in Equation 3.7.

$$BP = \frac{2\pi NT}{60000} \quad (3.7)$$

where N is the engine speed in revolutions per minute, and T is the engine brake torque in N.m.

The average mean pressure in the cylinder that would produce the measured brake output is provided by the Brake Mean Effective Pressure, $BMEP$. This pressure is calculated as the uniform pressure in the cylinder as the piston rises from top to bottom of each power stroke. $BMEP$ is a useful calculation that theoretically compares the engine performance based on the brake power of any engine size. A direct judgment of the engine efficiency with different displacements is not a true comparison since the engine performance is largely affected by the different engine capacities. The engine $BMEP$ (bar) is calculated by the formula given in Equation 3.8.

$$BMEP = \frac{BP \times 60000}{LAnK} \quad (3.8)$$

where BP is the engine brake power in kW, and L is the cylinder stroke length in m, A is the cross-sectional area of the cylinder in m^2 , n is the engine speed divided by two (for four-stroke) in revolution per minute, and K is the number of cylinders.

Brake Thermal Efficiency, η_{th} is the ratio of heat of combustion from the fuel against the useful mechanical power developed by the engine. It is a dimensionless number to measure the performance of the engine. The heat of combustion is the energy released as heat when the fuel undergoes complete combustion with oxygen under standard conditions. The heat of combustion is obtained by multiplying the mass flow rate of fuel with the lower heating value,

while the mechanical power is taken from the engine brake power. The engine $\eta_{th}(\%)$ is calculated by the formula given in Equation 3.9.

$$\eta_{th} = \frac{BP}{\dot{m}_f \times LHV} \times 100 \quad (3.9)$$

Where BP is the engine brake power in kW (equivalent to kJ/s), $\dot{m}_f$ is the mass flow rate of fuel in kg/s, and LHV is the lower heating value of the fuel in kJ/kg.

The Brake Specific Fuel Consumption $BSFC$ is the mass fuel flow rate required to produce a unit of power output of the engine. The engine $BSFC$ (bar) is calculated by the formula given in Equation 3.10.

$$BSFC = \frac{\dot{m}_f}{BP} \quad (3.10)$$

where $\dot{m}_f$ is the mass flow rate of fuel in kg/h, and BP is the engine brake power in kW.

The Mass Flow Rate of Fuel, $\dot{m}_f$ is the measurement of the amount of fuel per unit time that flows through the fuel gauge. The $\dot{m}_f$ (kg/s or kg/h) is calculated by the formula given in Equation 3.11.

$$\dot{m}_f = \frac{\rho_a \times V_f}{t} \quad (3.11)$$

where ρ_a is the density of air in kg/m³, V_f is the volume of the fuel in m³, and t is the time for a specific fuel volume to drain in seconds.

The air-fuel ratio (AFR) is the mass ratio of air to fuel present in the engine combustion process. The *AFR* (unitless) is calculated by the formula given in Equation 3.12.

$$AFR = \frac{A}{F} = \frac{\dot{m}_a}{\dot{m}_f} \quad (3.12)$$

where $\dot{m}_a$ is the mass flow rate of air in kg/h, and $\dot{m}_f$ is the mass flow rate of fuel in kg/h.

Heat Release Rate, *HRR* is the rate at which the combustion releases energy or power. The *HRR* is the difference between chemical energy or heat released by the combustion of the fuel and the heat transfer from the system. The *HRR* (in J/°CA) is calculated by the formula given in Equation 3.13. *HRR* calculation employs the Pressure Rise Rate, *PRR* value which is calculated from the in-cylinder pressure data. *HRR* and *PRR* profiles can be used to estimate the timing and intensity of the combustion as an indication of the ignition delay.

$$HRR = \frac{dQ}{d\theta} = \frac{\gamma}{\gamma - 1} p \frac{dV}{d\theta} + \frac{1}{\gamma - 1} V \frac{dp}{d\theta} \quad (3.13)$$

where γ is the specific ratio of heat, p is the instantaneous cylinder pressure, and V is the instantaneous volume of the engine cylinder. $\frac{dV}{d\theta}$ and $\frac{dp}{d\theta}$ are both the rate of the volume (m³/°CA) and pressure (bar/°CA) rise respectively. These parameters rise with the change of the crank angle during the combustion cycle and can be calculated by taking the value difference at two different crank angles.

3.4.9 Experimental Techniques

In this section, several techniques related to the experimental execution are explained. The performance test is tested at full engine load to cover the higher range of engine capacity and capability. The test is executed with a structured experimental plan to cover all ranges of the tested parameters. Several control variables are set in performing the investigation runs at high to low engine load.

Taguchi Design of Experiment (DOE) methodology is utilized to investigate the effect of both parameters towards the engine performance, combustion behaviour and emissions level. The results of the experimental run are analysed using the Minitab statistical software (Version 22). A similar software is used to analyse the parametric study in the previous simulation stage. The description of how the Taguchi methodology is applied and its relevance to the investigation is also explained in the next section.

3.4.10 Performance test at full engine load

The performance test is conducted until full engine load conditions. This means that the engine is tested at the maximum or full range of capacity up to 100 % engine load within 1800 to 3600 rpm engine speed as specified by the engine specifications. Maximum fuel opening shall be applied for the engine to be tested at the maximum capacity. The performance test of this engine is expected to produce the maximum rated output of approximately 3.5 kW at 3600 rpm. Engine test runs with a maximum rated output lower than 3.4 kW at 3600 rpm shall be considered underpowered. Any test run that results on such brake power shall be considered abnormal and further inspection needs to be performed before the next test run.

3.4.11 Swirl air intake device experimental test array (L_93^2 Orthogonal Array)

The experimental execution of the engine performance measurement with the addition of the swirl air intake. The DOE method is used to plan a structured experimental design with the combination of the selected factors covering all the specified levels. The number of runs is strategically set based on the combination of at least two factors and a parameter level as suggested by the method. Rather than relying on trial-and-error, random and time-consuming experimental executions, the application of the DOE method considers how significant are the studied factors based on the effect of each factor on the response. The result findings are assisted with the statistical analysis for accurate decision-making that is highly relevant for any analysis.

DOE has many types of methodologies that work differently. However, the main objective is similar to the optimization of the studied parameters. The DOE includes Full Factorial, Fractional Factorial, Response Surface Methodology and Taguchi methodology. In the experiment execution, the focus is given only to the Taguchi methodology based on many successful research conducted with Taguchi methodology and the benefit to the experimental duration and cost-effectiveness. There are two arrays arranged orthogonally to each other as introduced by the Taguchi method; the inner and outer arrays. In this experiment, the inner array is used to create a replication for each experimental run and the outer array is specially to create the combination of the studied factors.

The combination of factor and level as suggested by the Taguchi method considers an orthogonal array which means the design is balanced so that factor levels are weighted equally and each factor can be assessed independently. Table 3.16 below shows the available combination of experimental design according to Taguchi's methodology.

Table 3.16: Available combination of Taguchi Orthogonal Array Design

Orthogonal Array	Number of Runs	Maximum Factors	Maximum of columns at these levels			
			2-level	3-level	4-level	5-level
L4	4	3	3			
L8	8	7	7			
L9	9	4		4		
L12	12	11	11			
L16	16	15	15			
L'16	16	5			5	
L18	18	8	1	7		
L25	25	6				6
L27	27	13		13		
L32	32	31	31			
L'32	32	10	1		9	
L36	36	23	11	12		
L'36	36	16	3	13		
L50	50	12	1			11
L54	54	26	1	25		
L64	64	63	63			
L'64	64	21			21	
L81	81	40		40		

(Source: Balisnomo, 2008)

This study involved two control factors with three parameter levels for each factor. The two control factors are the helical blade revolution and the device position. The revolution of the helical blade has been identified as the main factor in the experiment design since it is the main characteristic that forms the swirling airflow at the outlet. The blade is used to produce an air swirl flow at the outlet by guiding the inducted air to flow helically and constantly according

to the curvature. As the number of revolutions increases, there will be more curvature on the blade profile. More curvature causes more air deviation from being in a straight line. This creates a stronger swirl intensity as well as increasing the air velocity. Therefore, the helical blade revolution is selected as the first factor to understand the effect of a different blade revolution on the engine characteristics that are mainly affected by the fuel-air mixing especially the brake power, torque, in-cylinder pressure and emissions level.

Device position has been considered as the other factor alongside the helical blade revolution. The position in this experiment can also be identified as the distance of the device assembled within the air intake system from the engine intake port. The device position is studied to identify the effect of varying the distance of the swirl air intake device within the air intake system. As the swirl produced in the cylinder is decayed due to the cylinder wall friction and the energy dissipation of the charge (Abo-Elfadl & Abd El-Sabor Mohamed, 2018), theoretically, the swirl decay in the air intake system may also be possible due to the effect of the wall friction and energy dissipation by a different swirl device. Hence, it is practically important to study and conclude whether the swirl decay in this particular application is significant to the brake power and torque. The significance of this effect shall be concluded from the statistical results of the main and interaction effects between each factor via the Analysis of Variance (ANOVA).

Table 3.17 shows the completed experimental plan with the designed factor and level according to the Taguchi method. In total, there are 9 experiment runs with a balanced factor level that is equally weighted based on the $L_9 3^2$ Orthogonal Array.

Table 3.17: $L_9(3^2)$ Orthogonal Array (Inner and outer) for engine experimental design

INNER ARRAY (CONTROLLED FACTORS)		OUTER ARRAY (REPLICATIONS)			
		Replication	1 st	2 nd	3 rd
Run Number	IM Helical Design (rev.)	IM Location			
1	0.25	1	Studied responses (From data collection): Performance (Power, Torque, BSFC) Combustion (In-cylinder pressure) Exhaust emissions (HC, CO, CO ₂ , O ₂ , NO _x) Engine vibration Air intake flow velocity		
2	0.25	2			
3	0.25	3			
4	0.5	1			
5	0.5	2			
6	0.5	3			
7	0.75	1			
8	0.75	2			
9	0.75	3			

Note:

IM Location (distance from inlet port):

Location '1'=0 m, Location '2'= 0.07 m, Location '3'= 0.5 m

3.5 Summary

In the presented chapter, the research methodology is discussed in detail. The research consists of three (3) categories, swirl air intake device design and development, CFD simulation of swirl air intake, and experimental works on the swirl device-operated diesel engine operating with a pure diesel. The design and development of the swirl air intake model is conducted on Solidworks software, meanwhile, the CFD is conducted on an Ansys Workbench which includes the Ansys Fluent software for flow simulation work.

The prototypes of the swirl air intake device are 3-D printed and assembled into the engine's air intake system for the data measurement in the experimental stage. In order to work with the current air intake system, an additional apparatus is fabricated to properly mount the device and measurement instruments in a straight flow directly to the inlet port. The experimental work is conducted on a single-cylinder, four-stroke diesel engine to investigate the effect of swirl air intake variables on the engine performance and emissions.

CHAPTER 4

COMPUTATIONAL FLUID DYNAMICS (CFD) ANALYSIS OF THE SWIRL AIR INTAKE MANIFOLD DEVICE

4.1 Introduction

The objective of the swirl air intake flow simulation is to independently predict and analyse the swirled flow generated by the swirl air intake device. This chapter presents the findings of the CFD simulation work by varying the parameters of the swirl air intake device to observe the effect on the flow velocity, swirl intensity and other characteristics at the device outlet where the velocity-inlet and pressure-outlet are set according to the air intake condition of the engine at the speed of 3600 rpm. The CFD results are then validated against the experimental measurement data of the engine test bench.

4.2 CFD Modelling Verification and Validation

The CFD result of the velocity at the outlet has been validated against the experimental result using the swirl base model of 6 blades with 0.25, 0.5 and 0.75 revolutions. The body dimension of the model follows the parametric combination of 70mm length and 1.26° draft angle. Figure 4.1 shows the comparison between the CFD velocity-outlet (average) with the experimental velocity-outlet measurement using a standard vane anemometer. The comparison shows a good agreement between them with a 2.17 % difference for Base 2 – 0.5 revolution. Table 4.1 summarizes the difference for all the verified models.

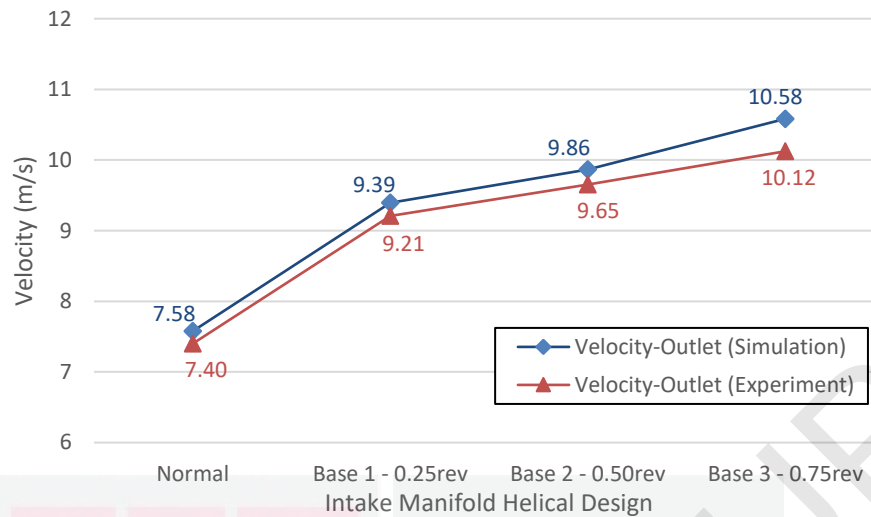


Figure 4.1: Average Velocity-Outlet Intake Manifold Helical Design with 6 Blades (Simulation versus Experiment)

Table 4.1: Summary of the differences in the velocity-outlet of CFD simulation against experiment.

Model	Velocity-Outlet (Average) – Simulation	Velocity-Outlet – Experiment	Difference (%)
Normal	7.578	7.401	2.39
Base 1 – 0.25 rev	9.395	9.207	2.04
Base 2 – 0.5 rev	9.864	9.654	2.17
Base 3 – 0.75 rev	10.583	10.123	4.54

More comparison is conducted to verify the swirl intensity range and outlet velocity effects in a different swirl air intake model. Figure 4.2 shows the general comparison of the swirl intensity for four parametric combinations, base and normal intake model. The chart may represent two observations that can be concluded. First, the observation of two different stages that employs

different mesh types. Second, the observation of the swirl intensity range across the air intake model.

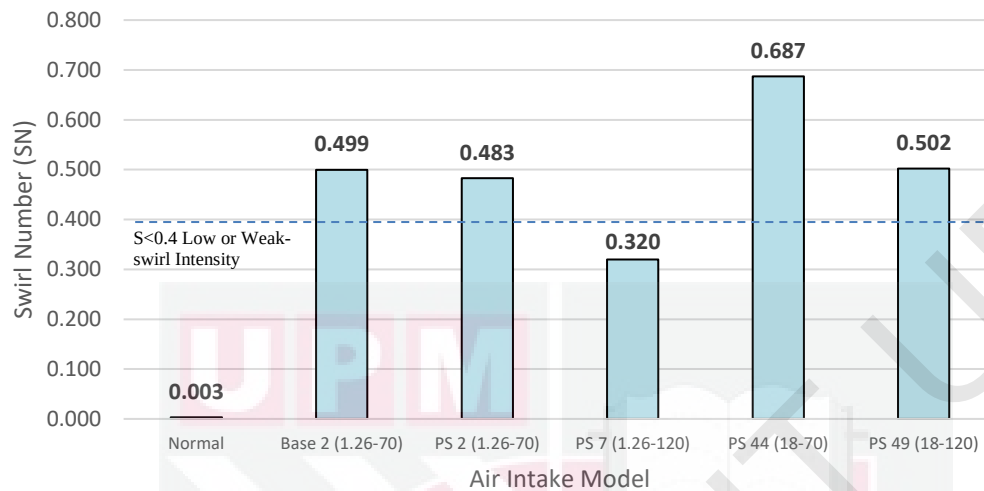


Figure 4.2: Comparison of Swirl Number for parametric, base and normal intake simulation at 0.5 revolution

The first observation concludes that the swirl air intake model performs similarly well in two different stages, where the swirl airflow formed by the blade geometries can maintain a good swirl intensity. The change of mesh from tetrahedral (1st stage simulation) to polyhedral (2nd stage simulation) does provide two variations in the swirl number resulting from the variation in the flow velocity magnitudes as shown in Figure 4.2. This result is consistent with the earlier prediction that anticipates the change of results due to the increase in accuracy and sensitivity in the modelling. But overall, the swirl number is almost identical with a 3.2% difference (similar parameter conditions for Base 2 and Parametric Simulation 2 (PS2) resulted in the Swirl Number of 0.499 and 0.483 respectively. This indicates the two simulations are reaching similar conclusions which may provide confidence in the reliability of the results.

For the second observation, compared to the normal intake, it can be concluded that all swirl models (parametric and base) are capable of producing the swirl airflow at the outlet. The swirl parametric model simulation produces a swirl number of at least 0.320 for 1.26°-120mm and a maximum of 0.775 for an 18°-50mm combination. Meanwhile, swirl base model simulation produces a swirl number of at least 0.303 for a 1.26°-70mm combination (0.25 revolution) and 0.701 for a 1.26°-70mm combination (0.75 revolution). This indicates the effectiveness of the helical blade geometries to the overall design of the swirl air intake device. A complete discussion of these findings is provided in the proceeding sections.

Further down to the range of swirl intensity, the low or weak-swirl intensity ($S < 0.4$) is recorded for only models with a device length of 100mm to 120mm that combines with the draft angle of 1.26° to 3°. It means any combination with a lower than 100 mm length and over 3° draft angle is capable of producing at least medium-swirl intensity ($S \geq 0.4$). This also indicates that the better design for producing medium to high-swirl airflow is the larger draft angle but shorter device length. In this case, a longer device is not preferable in a device that generates a swirl as it increases the energy dissipation that may further decays the swirl intensity when approaching the outlet. A larger draft angle also significantly increases the energy dissipation, however, due to the domination of the 'ram' effect by the relative bulk motion of air intake charged, the effect from the energy dissipation may not be significant to the swirl decay.

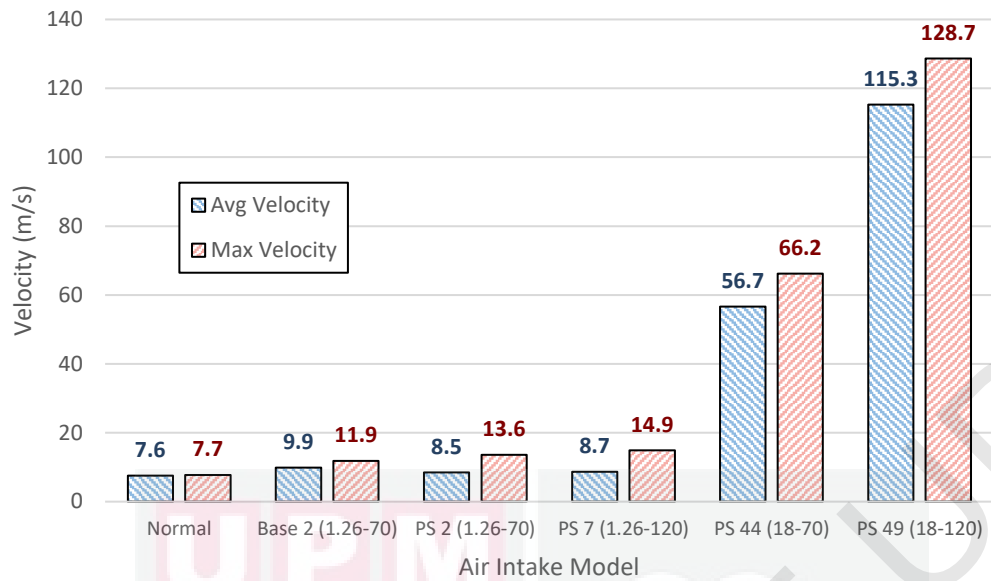


Figure 4.3: Comparison of Velocity-Outlet for parametric, base and normal intake at 0.5 revolution

Figure 4.3 shows the general comparison of the velocity-outlet for four parametric combinations, base and normal intake. It is observed that the change of mesh in the 2nd stage simulation does change the velocity magnitude at the outlet where there are two variations exist in the maximum and average velocity magnitude. Comparing Base 2 and PS 2 model, the maximum velocity magnitude is recorded higher than the 1st stage but instead of a higher average velocity magnitude, it is actually lower than the 1st stage. In terms of percentage difference, Base 2 recorded approximately a 16.8% difference between the maximum and average velocities, while PS 2, it is recorded approximately 60% difference. Explaining the possibilities of the large differences in these two models, the maximum velocity and average velocity in the fluid flow simulations can exhibit different behaviours between polyhedral and tetrahedral meshes which is mainly due to the variations in flow distribution within the mesh elements.

Typically, polyhedral meshes have a larger element compared to tetrahedral meshes for the same volume causing the polyhedral to have lower resolutions or density. This is also the reason why the number of elements in the polyhedral is lower than tetrahedral in the current simulation. Further justification on the number of elements, the model used for the current simulation work has a number of elements that are consistent with these assumptions. To clarify, the number of elements in the fluid region for the PS 2 model is recorded at 484,127 which is lower than the number of elements for the Base 2 model with 512, 499. From the shape of the element, there is a larger number of neighbors for each node in the polyhedral than that of the tetrahedral mesh causing the polyhedral to have a slightly larger number of mesh nodes. This is also consistent with the number of nodes for the Base 2 and PS 2 models. The larger number of mesh nodes can significantly improve the prediction of high-gradient areas (H. Zhang et al., 2020).

Base 2 and PS 2 are the two kinds of swirl models from the 1st and 2nd simulation stages that represent the same parametric levels with similar draft angles and device length (Draft Angle = 1.26° and Length = 70 mm), but they are simulated using a different type of meshes. This is done to compare their differences when using tetrahedral and polyhedral meshes. For the mesh element shape, polyhedral meshes have irregular element shapes and aspect ratios where this element can adapt to the shape of the complex geometry more easily than tetrahedral, which typically require more elements to capture complex shapes accurately (Y. Li et al., 2023; W. Wang et al., 2021). The flow near these elements will be further influenced by this irregular shape and

affecting the accuracy of the airflow, leading to higher sensitivity to changes in geometrical features.

Back to the maximum velocity, the magnitude represents the highest velocity within the selected flow region. It is possible that polyhedral meshes can capture localized flow features more accurately than tetrahedral meshes due to polyhedral sensitivity and the ability to represent complex shapes with fewer elements on the same geometry as tetrahedral. The ability of the polyhedral in complex geometries results in better capturing of the flow characteristics, which then leads to a higher maximum velocity magnitude. For average velocity at the outlet, this magnitude represents the mean velocity over the selected flow region. The mean velocity can be affected by the irregular shapes and varying element sizes where the uneven distribution of flow properties due to this irregularity can lead to a lower average velocity value in polyhedral meshes.

4.3 Boundary Conditions

A detailed description of the boundary conditions used in this simulation study has been discussed in the previous chapter (Section 3.3, CFD Simulation). In brief, the entire simulations for both 1st and 2nd stages of simulation work use the velocity-inlet, pressure-outlet and standard wall boundary conditions. The velocity-inlet boundary condition is used where the static pressure at the inlet will vary with the change of the device geometries. A pressure-outlet boundary condition is used to define the static pressure at the outlet of the swirl air intake device. The velocity-inlet and pressure-outlet values are 7.4 m/s and -5391 Pa

respectively and consistent with the experimental condition of an actual engine run at 3600 rpm. The velocity and the vacuum pressure are chosen to study the effect of a diesel air intake flow on the swirl air intake device of different geometries, creating a certain level of swirl intensity with the 'ram' pressure. The static pressure is assumed to be the condition of the pressure that occurs in the flow domain since the simulation is involved with a subsonic flow, where the subsonic flow occurs when the Mach number is less than 1. Mach number is defined as the ratio of the speed of a flow to the speed of sound in the surrounding medium. A standard wall condition (no slip) is used to define the interface between the fluid and the solid body where the viscous effect is considered.

The simulations are conducted under two different parts of the study. The first part involved the air intake with the base model of the device to simulate the effect of different helical blade parameters and numbers with a compromised draft angle in maintaining the normal intake flow volume. This part focuses only on establishing the effective swirl intensity formed by the swirl base model. The velocity and the swirl intensity demonstrated by the CFD study represent the visualization of flow characteristics that will enter the cylinder during intake stroke. The first part of the study is important as a proof-of-concept, establishing ground understanding and feasibility evaluation before developing and exploring other possible parameters as per the research objective.

The second part of the study focuses on the establishment of swirl intensity with the 'ram' effect that exists in the system due to the greater pressure difference created by the larger draft angle in the swirl parametric model. The relative bulk motion of the air through the device will increase the fuel-air mixing rate (due to air swirl) and mass of airflow (due to ram pressure) which also means an increase in the inducted air volume entering the engine cylinder. The pressure difference across the device is expected to provide greater flow velocity in conjunction with the swirl formation compared to the normal intake system. This is because the outlet of the device is designed to be in size with engine inlet port geometry, thus should not cause a reduction of airflow, but instead, induce more air volume to the engine.

The turbulence model utilised in the entire simulation is Realizable k -epsilon (k - ϵ). The decision to use this model is based on the findings of the previous studies relating to a complicated flow structure and pressure gradient, where numerous investigations that cover various swirl flow strengths or intensities have specifically employed the Realizable k -epsilon (k - ϵ) turbulence model in their simulation works. This justification has been explained in the previous chapter (Section 2.10.2 Turbulence Models). The swirl air intake model is designed with the unique features of the helical blades and inner draft angle to form the swirl airflow with the 'ram' effect, thus the complexity of the flow structure within the flow domain has become the major concern in the simulation. The realizable k - ϵ model is used in this study since this model has been found to be more accurate than the other k - ϵ models in the fluid flow prediction involving high-pressure gradients and complex flow structures.

The other fluid parameters such as density and dynamic viscosity are kept consistent throughout the simulations. Table 4.2 shows the summary of the boundary conditions assumptions used in the simulation work.

Table 4.2: Boundary conditions assumptions at different simulation stages

Boundary Condition	Swirl Base Study (Stage 1)	Swirl Parametric Study (Stage 2)
Mesh type	Tetrahedral	Polyhedral
Turbulence Model	Realizable $k-\epsilon$	Realizable $k-\epsilon$
Velocity-Inlet (m/s)	7.4	7.4
Pressure-Outlet (Pa)	-5391	-5391
Walls	Standard (No-slip condition)	Standard (No-slip condition)
Material	Fluid (Air)	Fluid (Air)
Density (kg/m ³)	1.225	1.225
Dynamic Viscosity (kg/m.s)	0.000017894	0.000017894
<i>Note:</i> 8 simulation runs were conducted under Stage 1 49 simulation runs conducted under Stage 2		

4.4 Swirl Base Model Simulation Findings and Analysis

The flow field is analysed in the inner volume of the swirl base model. Figure 4.4 shows the path lines of the air stream that flow smoothly over the blades for 0.5 revolution. This is consistent for all three cases of different revolutions (0.25, 0.5 and 0.75 revolutions). The air streams are guided through the helical blades which eventually form the swirl airflow inside the flow field. The velocity

of the air stream increases dramatically as it approaches the outlet of the device. Similar flow characteristics can also be seen for all five cases of different blade numbers (4,5,6,7 and 8 blades).

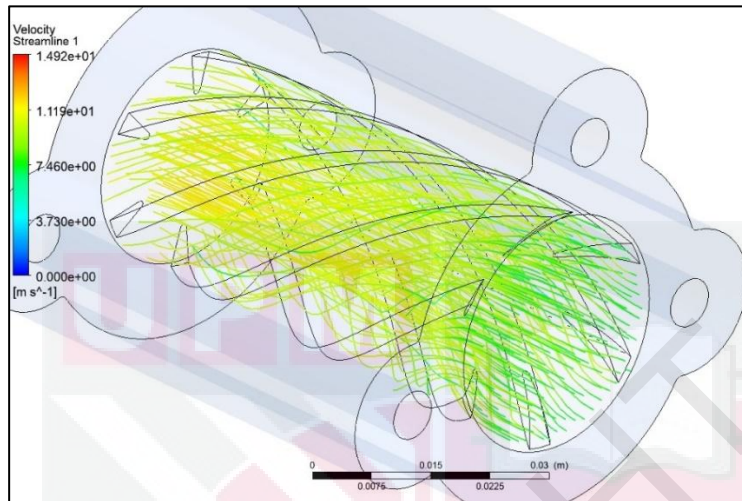


Figure 4.4: Smooth air streams flowing through the blades starting from the inlet of the device (0.5 revolution)

Figure 4.5 shows the velocity vectors observed at the outlet position where the swirling airflow is expected to form before being inducted into the engine cylinder through the inlet port. From the velocity vector results, the flow moves tangentially in a specific direction after being guided by the helical blade geometry. This is consistent for all three cases of different revolutions. The changes in the flow direction that initially moves in the axial direction at the device upstream indicate the formation of a strong and robust swirl airflow. Similar velocity vector characteristics can also be seen for all five cases of different blade numbers.

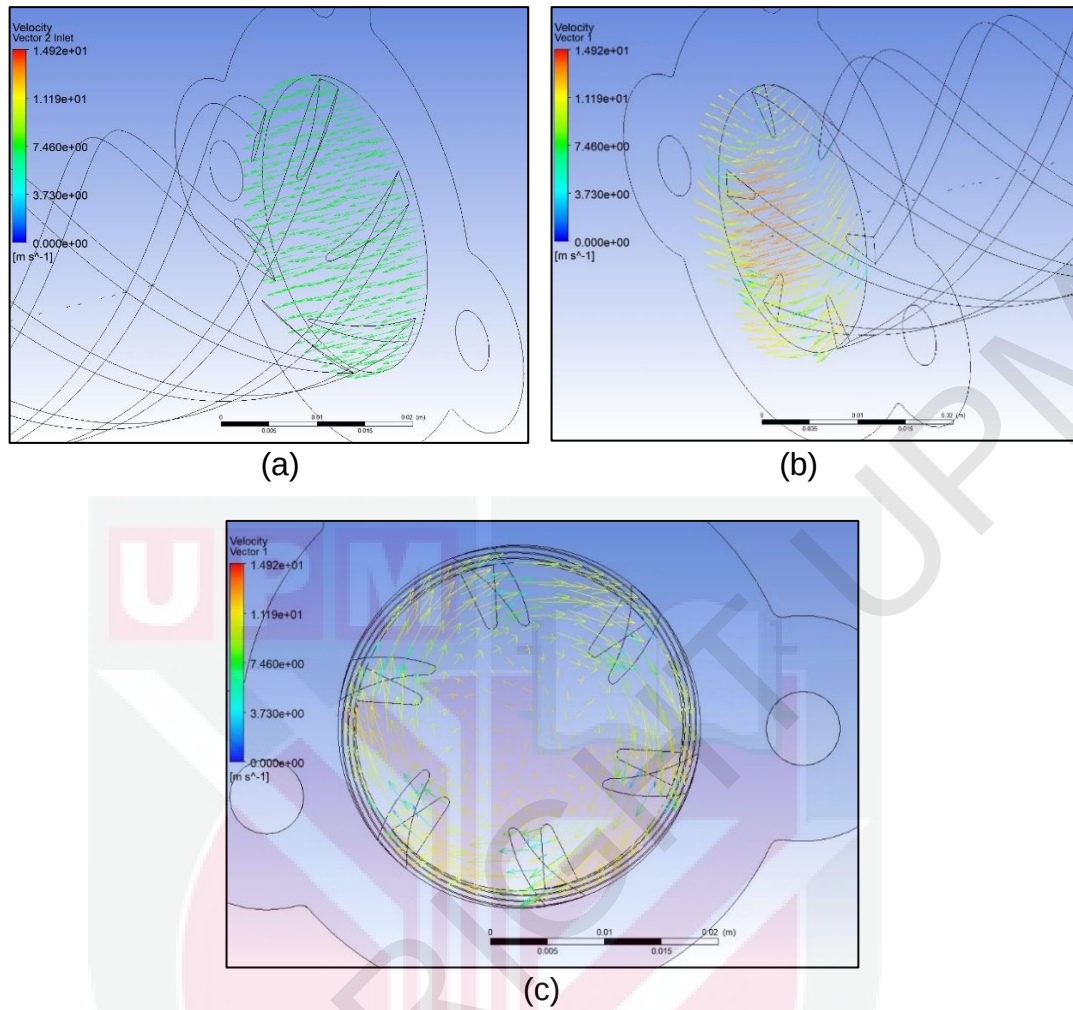


Figure 4.5: Swirl airflow formation at the outlet of the device and the change of the velocity vector direction (0.5 revolution); (a) Velocity vector at the inlet, (b) Velocity vector at the outlet (c) Front view of the velocity vector at the outlet.

Figure 4.6 shows the contours of static pressure at the wall section along the device for 0.5 revolution. The static pressure analysis taken in this section is to show the change of pressure mainly near the helical blades that guide the air stream. It can be seen that the static pressure changes dramatically through the device area. The static pressure is found to be minimum at the outlet location and maximum at the upstream location where there is an opening blade that receives the incoming air.

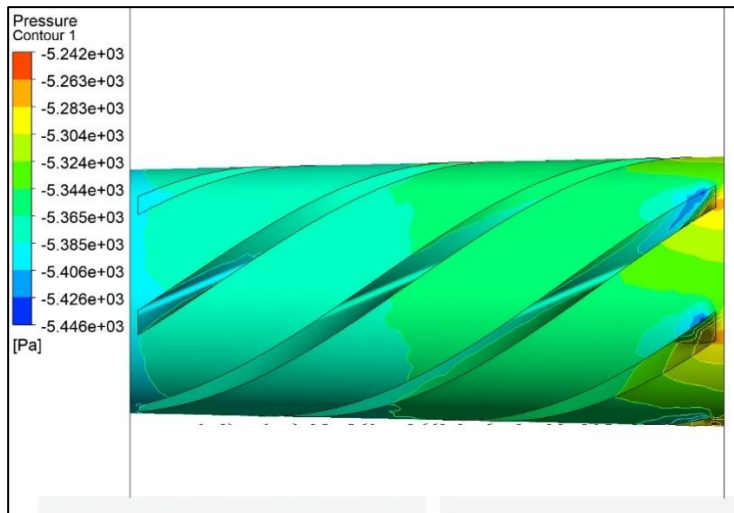


Figure 4.6: Contour of the static pressure on the y-z plane inside the device (0.5 revolution)

This result shows that there is a pressure drop across the device as expected due to the existence of the helical blade geometry that provides some amount of restriction to the airflow. This is also contributed by the draft angle imposed on the inner cylinder design. The effect from the pressure drops, however, has been considered in the helical blade design to minimize the impact of air restriction. Since the outlet size of this device is designed to be in size with the inlet port of the intended engine, the pressure drops also provide a positive impact by increasing the air intake velocity inducted to the engine. The inducted air intake may further benefit from the small amount of the 'ram' effect due to the draft angle design. A similar pattern of pressure contour change can be seen for all cases under different revolutions and blade numbers. A detailed comparison of all cases is provided in the next sections.

4.4.1 Simulation of different helix revolutions

Table 4.3 shows the comparison of the air stream path lines for the swirl base model of different helical blade revolutions. In general, the simulation of the

swirl base model provides three different maximum velocity magnitudes that occur in the flow domain at 11.1854 m/s, 11.8697 m/s and 13.133 m/s for 0.25, 0.5 and 0.75 revolutions respectively. Based on these results, there is a clear indication of the maximum velocity increase at the outlet with increasing blade revolutions.

Table 4.3: Swirl Base Model Streamline Flow Results (Different Blade Revolutions)

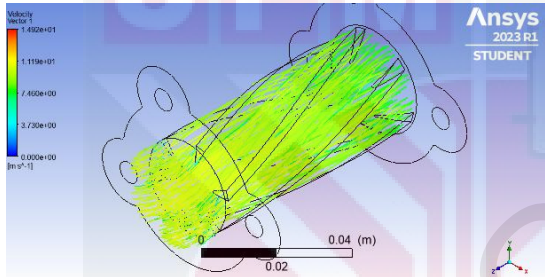
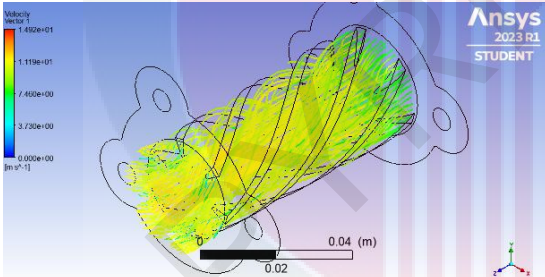
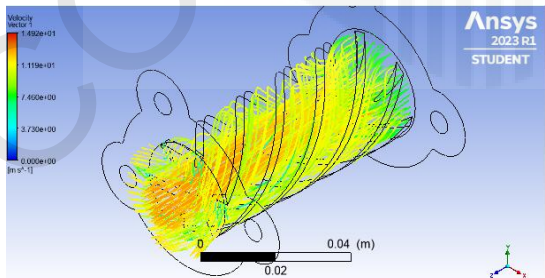
Streamline Flow Results	Details
	Model: Swirl Base Draft Angle: 1.26° Length: 70 mm Blade Numbers: 6 Blade Revolutions: <u>0.25</u> Max. Velocity-Outlet: 11.1854 m/s
	Model: Swirl Base Draft Angle: 1.26° Length: 70 mm Blade Numbers: 6 Blade Revolutions: <u>0.5</u> Max. Velocity-Outlet: 11.8697 m/s
	Model: Swirl Base Draft Angle: 1.26° Length: 70 mm Blade Numbers: 6 Blade Revolutions: <u>0.75</u> Max. Velocity-Outlet: 13.133 m/s

Table 4.4 shows the comparison of the velocity vector (outlet) for the swirl base model of different helical blade revolutions. In general, the simulation of the

swirl base model provides three different average velocity magnitude and swirl intensity at 9.395 m/s ($S=0.303$), 9.864 m/s ($S=0.499$) and 10.583 m/s ($S=0.701$) for 0.25, 0.5 and 0.75 revolutions respectively. These results are validated against the experimental data and discussed in Section 4.2 (CFD modelling validation).

Table 4.4: Swirl Base Model Vector (Outlet) Flow Results (Different Blade Revolutions)

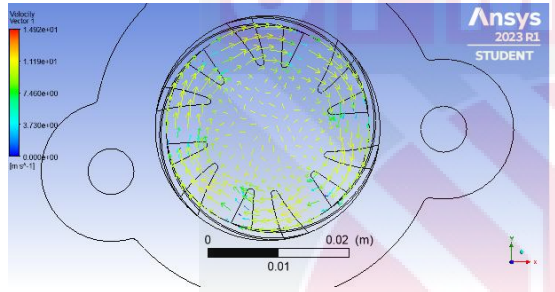
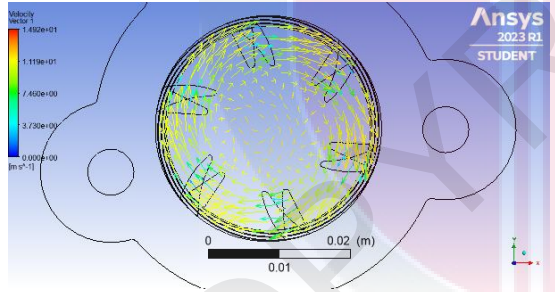
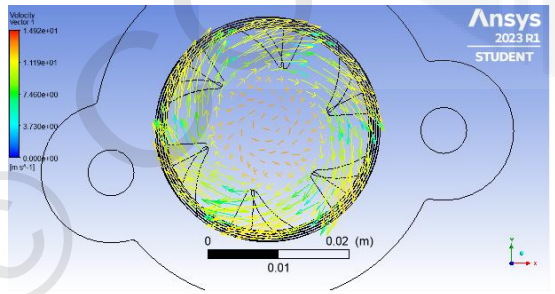
Vector (Outlet) Flow Results	Details
	Model: Swirl Base Draft Angle: 1.26° Length: 70 mm Blade Numbers: 6 Blade Revolutions: 0.25 Avg. Velocity-Outlet: 9.39451 m/s Swirl Number: 0.303
	Model: Swirl Base Draft Angle: 1.26° Length: 70 mm Blade Numbers: 6 Blade Revolutions: 0.5 Avg. Velocity-Outlet: 9.8639 m/s Swirl Number: 0.499
	Model: Swirl Base Draft Angle: 1.26° Length: 70 mm Blade Numbers: 6 Blade Revolutions: 0.75 Avg. Velocity-Outlet: 10.5829 m/s Swirl Number: 0.701

Figure 4.7 shows the comparison of the average velocity for each revolution. There is an increasing trend in the average velocity that is proportional to the increase of the blade revolution. Overall, compared to the normal intake, the swirl base model shows a higher velocity magnitude for all revolutions.

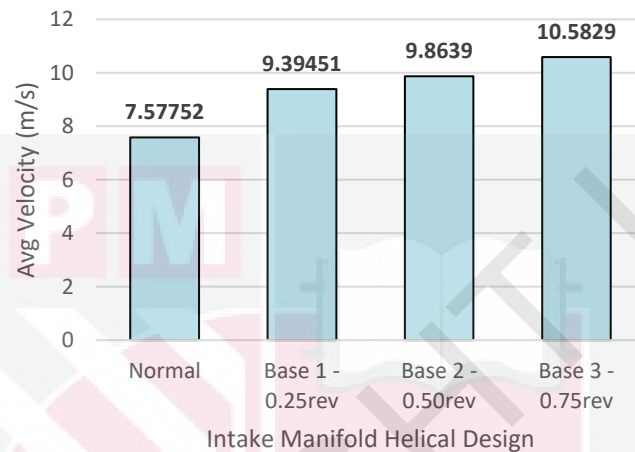


Figure 4.7: Comparison of the average velocity (outlet) for each revolution (6 blades) with normal intake

For the analysis of the swirl intensity, Figure 4.8 shows the comparison of the swirl number at the outlet with the normal intake. There is an increasing trend in the swirl number that is proportional to the blade revolution. The swirl number increases with an increase in blade revolutions. Overall, compared to the normal intake, there is a clear indication that all the swirl base models are capable of producing the swirl airflow as per the requirement.

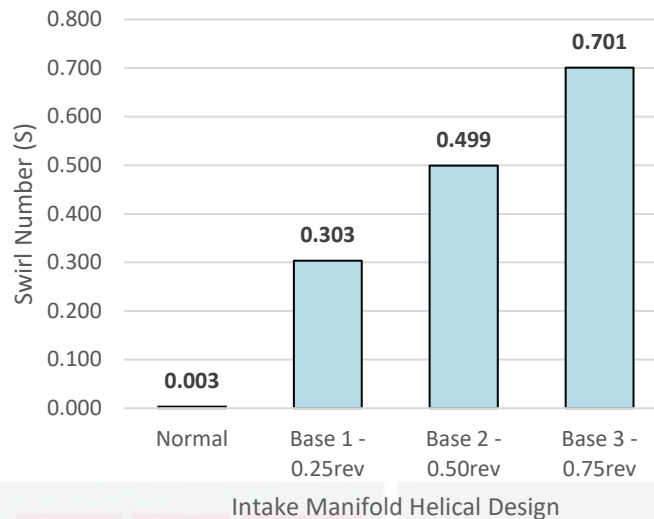


Figure 4.8: Comparison of the Swirl Number (outlet) for each revolution (6 blades) with normal intake

As outlined by Haminudin & Zamberi (2020) and Keshtkar (2018) on swirl intensity characterisation, Base 2 ($S=0.701$) and Base 3 ($S=0.499$) produce an intensity that can be considered as strong or high-swirl, while Base 1 ($S=0.303$) produces weak or low-swirl intensity. For normal intake, there is no indication of any swirl formation with the Swirl Number of 0.003. The swirl intensity in the 'normal' and 'swirl' intakes are expected to be different. This includes the variation in the helical blade revolution. The revolution of the blade forms a curvature. Swirl airflow will be produced at the outlet when the air is constantly guided to flow helically according to the curvature. As the number of revolutions increases, there will be more curvature on the blade profile. More curvature causes more air deviation from being in a straight line. This creates a stronger swirl intensity as well as increasing the air velocity.

However, for engine application, as the number of revolutions increases, there will be more obstruction to the direct airflow. Thus, more air restriction could be happened through the device and this may further increase when the

engine speed increases. Due to this reason, the base-swirl model with the least blade revolution but still having the necessary swirl intensity could also perform better than the other when the airflow efficiency becomes significant to the engine operation. This effect will be more apparent during the actual performance test and verification. In consideration of the effect of air resistance due to the inner features and blade helical revolution, the average Turbulence Energy Dissipation (TED) is used to provide a more representative measure of the overall impact of air resistance throughout the swirl device.

Figures 4.9 and 4.10 show the comparison of the maximum Turbulence Kinetic Energy (TKE) produced and the average Turbulence Energy Dissipation (TED) at the outlet. It can be observed that the Base 3 design (0.75 rev) has a lower maximum TKE compared to Base 2. Although the higher swirl intensity is preferable, however, the maximum TKE magnitude has dropped by 6.5% to 14.16 J/kg. Base 3 also has the highest average TED which indicates the greater impact of the air resistance in the model. High energy dissipation may cause swirl decay as highlighted by Abo-Elfadl & Abd El-Sabor Mohamed (2018). Base 2 may produce a greater performance with comparatively better air resistance compared to Base 3. It is also expected that the maximum TKE in the combustion chamber will be higher too as contributed by the highest maximum TKE in the Base 2 model. The contribution of a higher TKE value in increasing the engine's performance is also observed by Wu et al. (2022) in studying the effect of the swirl chamber turbulence combustion system in an air-cooled diesel engine. Thus, considering all these factors, the better design should be Base 2 followed by Base 1, then Base 3.

However, it may not be surprising if the Base 1 model can outperform Base 2 and 3 in the actual engine run. It is worth mentioning that Base 1 has the lowest average TED compared to the other base-swirl model, indicating a minimum impact by the air resistance in the model. Although Base 1 has a lower maximum TKE and low swirl intensity relative to the other swirl-base model, this model can perform sufficiently better than that of the normal intake. When the effect from the air resistance becomes significant in the engine operation, Base 1 could perform better than others with the lower air resistance as an advantage.

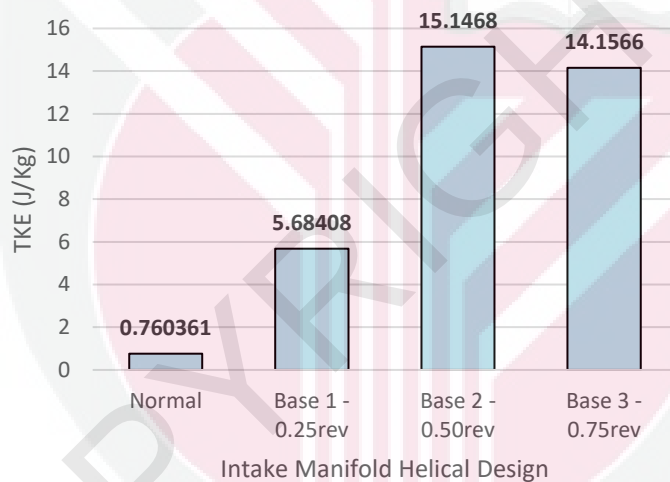


Figure 4.9: Comparison of the maximum TKE (outlet) for each revolution (6 blades) with normal intake

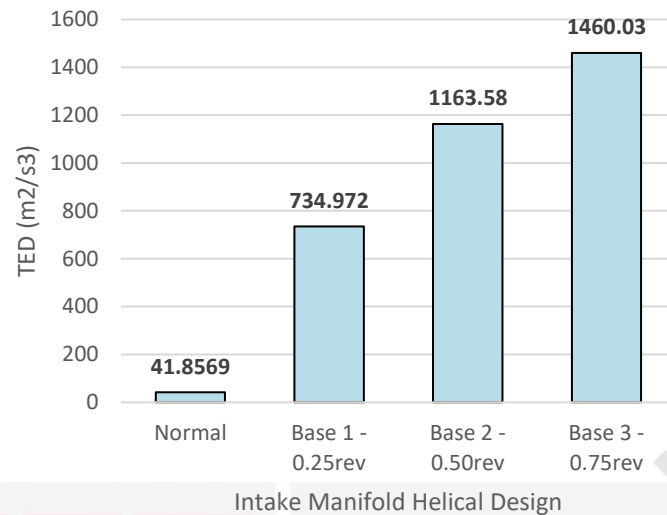
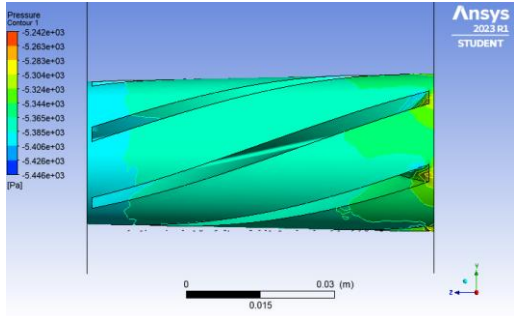
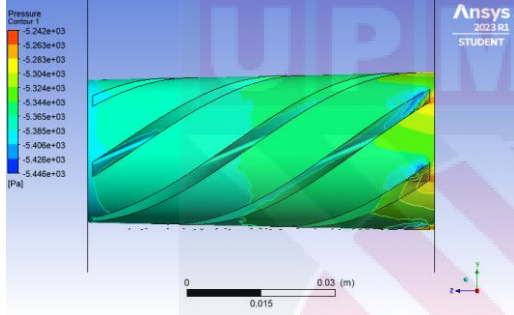
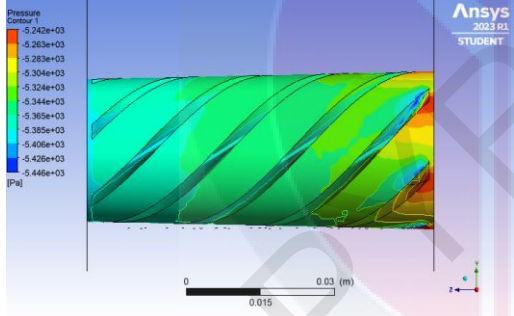


Figure 4.10: Comparison of the average TED (outlet) for each revolution (6 blades) with normal intake

Table 4.5 shows the comparison of the pressure contour result for the swirl base model of different blade revolutions. In general, the simulation of the swirl base model provides three different maximum pressures at the inlet with the pressure difference calculated at 98.01 Pa, 116.39 Pa and 152.84 Pa for 0.25, 0.5 and 0.75 revolutions respectively. The pressure difference is also referred to as pressure drop in a system, and it is directly proportional to the fluid velocity (Mamat, 2009). Based on the calculated value, it is observed that the pressure drops in the device increase linearly against the blade revolutions.

Table 4.5: Swirl Base Model Pressure Contour Results (Different Blade Revolutions)

Pressure Contour Results	Details
	Model: Swirl Base Draft Angle: 1.26° Length: 70 mm Blade Numbers: 6 Blade Revolutions: 0.25 Max. Pressure-Inlet: -5292.99 Pa Max. Pressure difference: 98.01 Pa
	Model: Swirl Base Draft Angle: 1.26° Length: 70 mm Blade Numbers: 6 Blade Revolutions: 0.5 Max. Pressure-Inlet: -5274.61 Pa Max. Pressure difference: 116.39 Pa
	Model: Swirl Base Draft Angle: 1.26° Length: 70 mm Blade Numbers: 6 Blade Revolutions: 0.75 Max. Pressure-Inlet: -5248.16 Pa Max. Pressure difference: 142.84 Pa

Compared to the CFD study conducted by Gocmen & Soyhan (2020) on a typical intake manifold geometry of a four-cylinder diesel engine, the pressure drops in these three models are at the lower range which is below the normal range of a typical air intake system. These results are expected since the design of the swirl base model focuses on maintaining a normal flow volume and minimising the effect of additional air restriction by the inner swirl features. Therefore, the use of this device alone or as an additional assembly part

should not reduce the engine performance. Instead, the use of the device may increase the performance as benefited by the overall swirl design.

4.4.2 Simulation of different blade numbers

Simulation on the effect of the blade number is one of the initial studies to understand the swirl and flow behaviour based on the number of blades. The blade is used to produce a swirling airflow at the outlet by guiding the inducted air to flow through the blade curvature. However, as the number of blades increases, there will be more obstruction to the airflow. Thus, more air restriction could be happened through the device and this may increase when the engine speed increases. The parameter of the blade used in this simulation is a helical shape with a nominal value of revolution. Based on the rule-of-thumb for one complete revolution, the nominal to be used for this simulation is the middle value which is 0.5 revolution.

Table 4.6 shows the comparison of the air stream path lines for the swirl base model of different blade numbers. In general, the simulation of the swirl base model provides five (5) different maximum velocity magnitudes occur in the flow domain at 10.9121 m/s, 11.3871 m/s, 11.8697 m/s, 12.5121 m/s, and 13.3154 m/s for 4, 5, 6, 7 and 8 blades respectively. Based on these results, there is a clear indication of the maximum velocity increase at the outlet with increasing blade numbers.

Table 4.6: Swirl Base Model Streamline Flow Results (Different Blade Numbers)

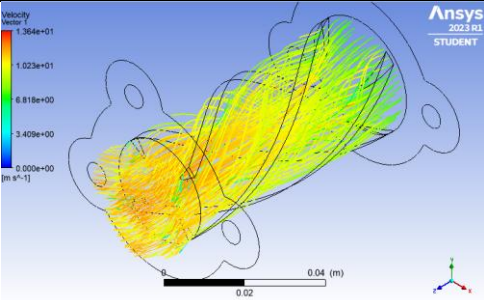
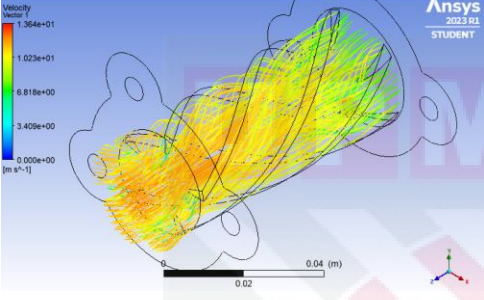
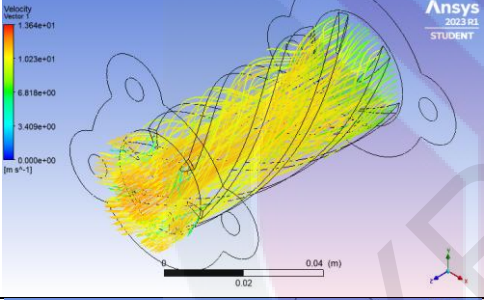
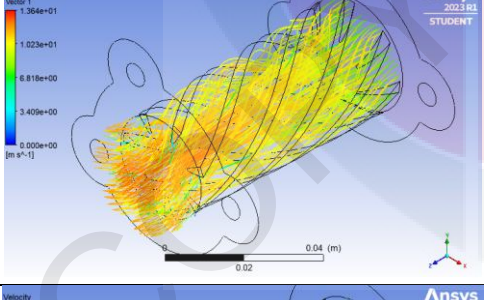
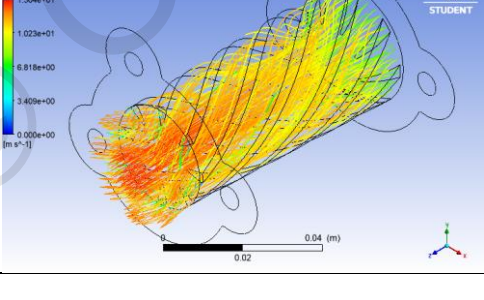
Streamline Flow Results	Details
	Model: Swirl Base Draft Angle: 1.26° Length: 70 mm Blade Numbers: <u>4</u> Blade Revolutions: 0.5 Max. Velocity-Outlet: 10.9121 m/s
	Model: Swirl Base Draft Angle: 1.26° Length: 70 mm Blade Numbers: <u>5</u> Blade Revolutions: 0.5 Max. Velocity-Outlet: 11.3871 m/s
	Model: Swirl Base Draft Angle: 1.26° Length: 70 mm Blade Numbers: <u>6</u> Blade Revolutions: 0.5 Max. Velocity-Outlet: 11.8697 m/s
	Model: Swirl Base Draft Angle: 1.26° Length: 70 mm Blade Numbers: <u>7</u> Blade Revolutions: 0.5 Max. Velocity-Outlet: 12.5121 m/s
	Model: Swirl Base Draft Angle: 1.26° Length: 70 mm Blade Numbers: <u>8</u> Blade Revolutions: 0.5 Max. Velocity-Outlet: 13.3154 m/s

Table 4.7: Swirl Base Model Vector (Outlet) Flow Results (Different Blade Numbers)

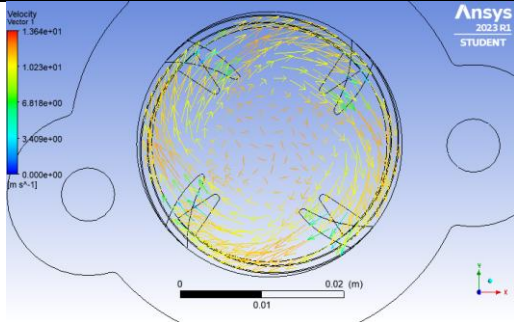
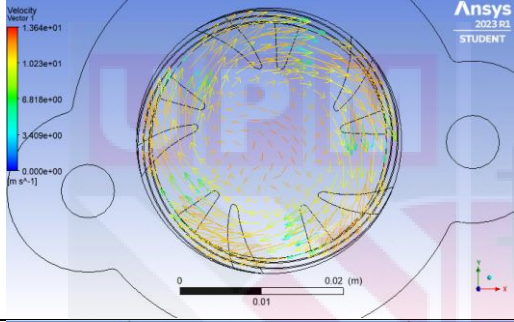
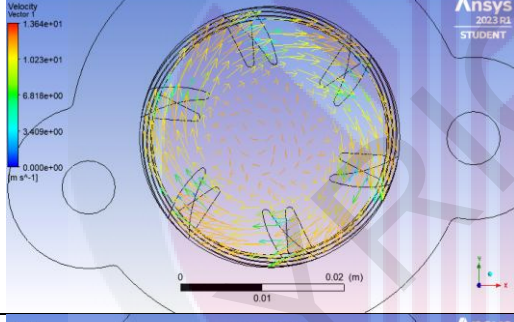
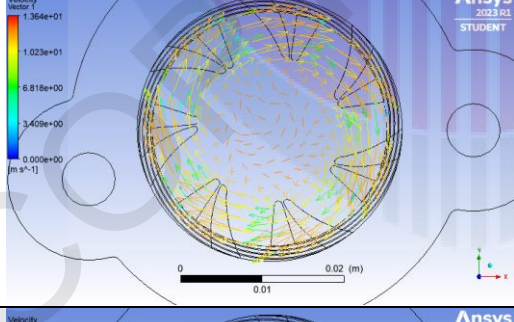
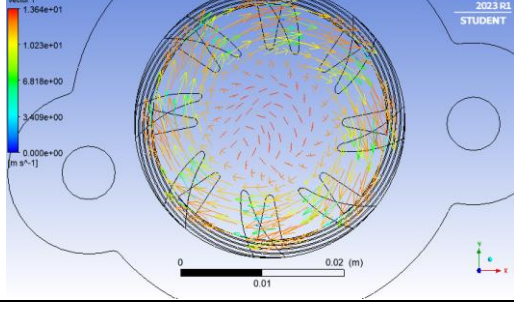
Vector (Outlet) Flow Results	Details
	Model: Swirl Base Draft Angle: 1.26° Length: 70 mm Blade Numbers: 4 Blade Revolutions: 0.5 Avg. Velocity-Outlet: 9.26791 m/s Swirl Number: 0.573
	Model: Swirl Base Draft Angle: 1.26° Length: 70 mm Blade Numbers: 5 Blade Revolutions: 0.5 Avg. Velocity-Outlet: 9.58331 m/s Swirl Number: 0.535
	Model: Swirl Base Draft Angle: 1.26° Length: 70 mm Blade Numbers: 6 Blade Revolutions: 0.5 Avg. Velocity-Outlet: 9.8639 m/s Swirl Number: 0.499
	Model: Swirl Base Draft Angle: 1.26° Length: 70 mm Blade Numbers: 7 Blade Revolutions: 0.5 Avg. Velocity-Outlet: 10.2592 m/s Swirl Number: 0.570
	Model: Swirl Base Draft Angle: 1.26° Length: 70 mm Blade Numbers: 8 Blade Revolutions: 0.5 Avg. Velocity-Outlet: 10.8099 m/s Swirl Number: 0.530

Table 4.7 above shows the comparison of the velocity vector (outlet) for the swirl base model of different blade numbers; all with 0.5 revolutions. In general, the simulation of the swirl base model provides five different average velocity magnitude and swirl intensity at 9.26791 m/s ($S=0.573$), 9.58331 m/s ($S=0.535$), 9.8639 m/s ($S=0.499$), 10.2592 m/s ($S=0.570$) and 10.8099 m/s ($S=0.530$) for 4, 5, 6, 7 and 8 blades respectively. Figure 4.11 below shows the comparison of the average velocity for each revolution. The average velocity magnitude is directly proportional to the increase of the blade revolution. Overall, compared to the previous normal intake average velocity, the swirl base model shows a higher velocity magnitude for all blade numbers.

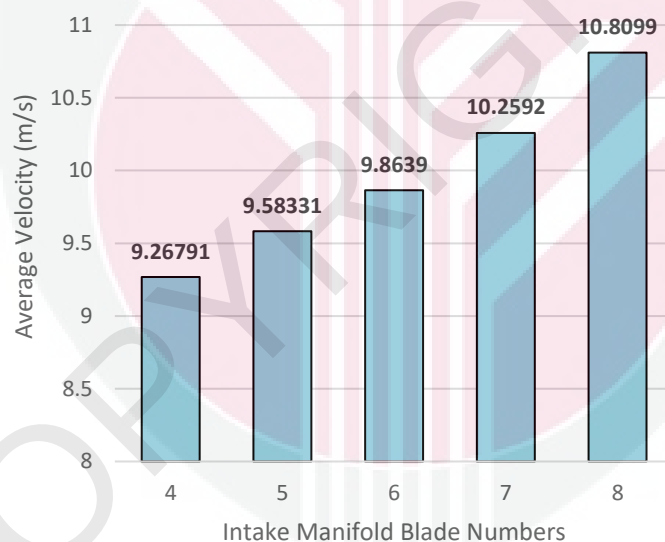


Figure 4.11: Comparison of the average velocity (outlet) for different blade numbers (0.5 revolution)

For the analysis of the swirl intensity, Figure 4.12 shows the comparison of the swirl number at the outlet for different blade numbers. The range of swirl intensity observed is between 0.5 to 0.573 which can be considered strong or high-swirl intensity ($S \geq 0.5$). None of these models produces weak or low-swirl

intensity ($S < 0.4$). However, there is no clear trend on the swirl number against the blade numbers. Based on the earlier findings, the swirl intensity is mainly dominated by the helical blade revolution.

However, the comparison of the swirl intensity involving similar blade revolutions may be dominated by a different factor such as the behaviour of the turbulence contributed by the number of blades and complex geometries. This could be further studied using much more quality or refined meshes to accurately quantify and understand the impact of blades and the complexity of shape to the turbulence behaviour. Overall, compared to the normal intake, all the swirl base models from 4 to 8 blades are capable of producing the swirl airflow as per the requirement.

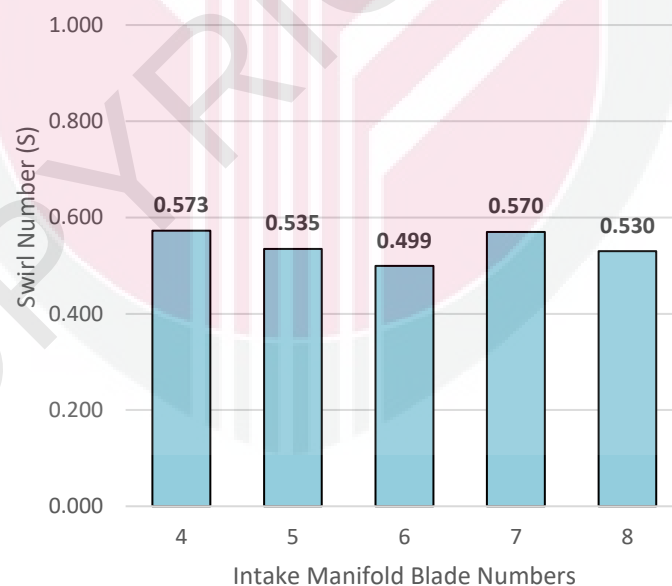


Figure 4.12: Comparison of the Swirl Number (outlet) for different blade numbers (0.5 revolution)

Figures 4.13 and 4.14 show the comparison of the maximum Turbulence Kinetic Energy (TKE) produced and the average Turbulence Energy Dissipation (TED) at the outlet. It can be observed that the 6-blade design has the highest maximum TKE with moderate average TED at the outlet. The average TED generally increases with the blade numbers. Taking the assumption based on Abo-Elfadl & Abd El-Sabor Mohamed (2018) findings on the energy dissipation of charge which causes the swirl to decay, TED can be part of the consideration when choosing the suitable blade numbers since the energy dissipation of the device may also affect the swirl decay. In brief, considering all these factors, 6 blades seem to be a better design compared to the other blade numbers.

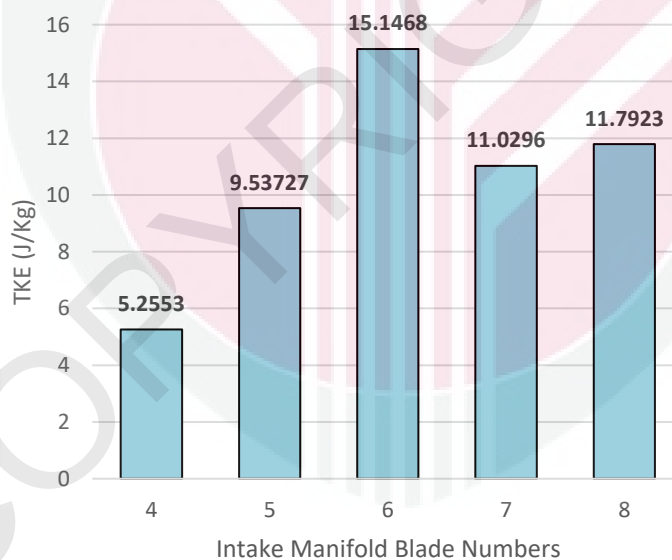


Figure 4.13: Comparison of the maximum TKE (outlet) against the blade numbers (0.5 revolution)

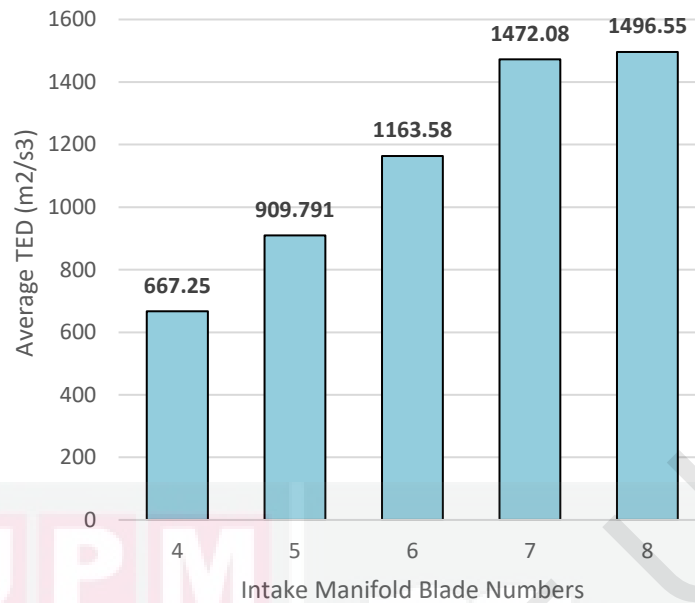
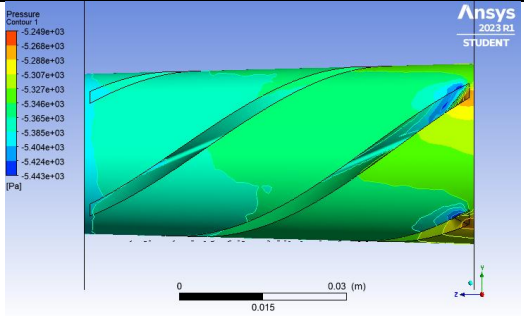
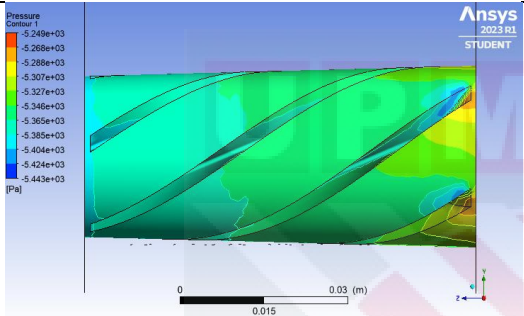
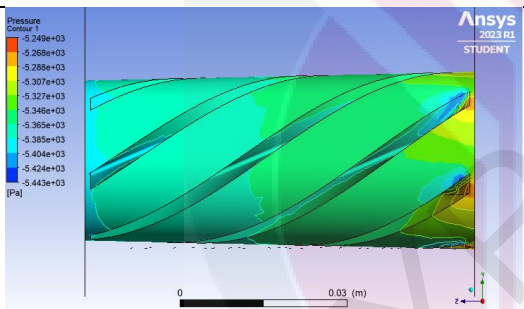
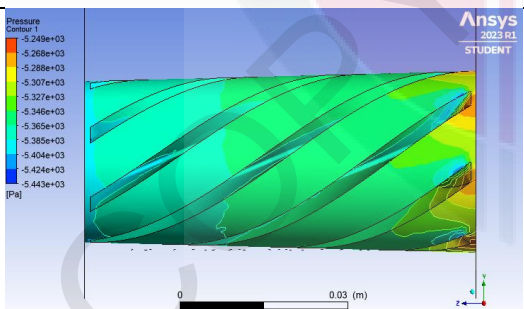
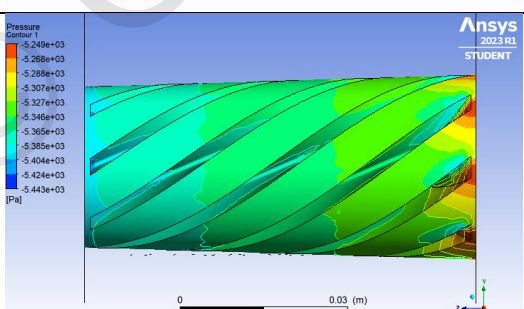


Figure 4.14: Comparison of the average TED (outlet) against the blade numbers (0.5 revolution)

Table 4.8 shows the comparison of the pressure contour result against the blade numbers. In general, the simulation of the swirl base model provides five different maximum pressures at the inlet with the pressure difference calculated at 93.40 Pa, 104.90 Pa, 116.39 Pa, 134.48 Pa and 145.31 Pa for 4, 5, 6, 7 and 8 blades respectively. Based on the calculated value, it is observed that the pressure drops in the device increase linearly against the blade numbers.

Compared to the CFD study conducted by Gocmen & Soyhan (2020), the pressure drops in these five models are still at the lower range which is below the normal range of a typical air intake system. Changing the blade number within 4 to 8 blades may have a small increase in the flow disturbance within the device but it does not greatly affect the normal intake charge. The use of this device with 4 to 8 blades should not reduce the engine performance. Instead, with the overall swirl design, it may increase the performance.

Table 4.8: Swirl Base Model Pressure Contour Results (Different Blade Numbers)

Pressure Contour Results	Details
	Model: Swirl Base Draft Angle: 1.26° Length: 70 mm Blade Numbers: 4 Blade Revolutions: 0.5 Max. Pressure-Inlet: -5297.6 Pa Max. Pressure difference: 93.40 Pa
	Model: Swirl Base Draft Angle: 1.26° Length: 70 mm Blade Numbers: 5 Blade Revolutions: 0.5 Max. Pressure-Inlet: -5286.1 Pa Max. Pressure difference: 104.90 Pa
	Model: Swirl Base Draft Angle: 1.26° Length: 70 mm Blade Numbers: 6 Blade Revolutions: 0.5 Max. Pressure-Inlet: -5274.61 Pa Max. Pressure difference: 116.39 Pa
	Model: Swirl Base Draft Angle: 1.26° Length: 70 mm Blade Numbers: 7 Blade Revolutions: 0.5 Max. Pressure-Inlet: -5256.52 Pa Max. Pressure difference: 134.48 Pa
	Model: Swirl Base Draft Angle: 1.26° Length: 70 mm Blade Numbers: 8 Blade Revolutions: 0.5 Max. Pressure-Inlet: -5245.69 Pa Max. Pressure difference: 145.31 Pa

4.5 Parametric Study Findings and Analysis

In the swirl parametric study, the CFD analysis conducted is mainly similar to the swirl base study except for the General Full Factorial analysis which provides more emphasis on the quantification of the parametric effects through statistical estimation. CFD analysis is done for all 49 parametric combinations, but the highlight is only focused on four chosen combinations. The four major combinations are the simulation run 2, 7, 44 and 49. These combinations are being considered based on the highest and smallest possible geometries that are suitable for the experimental engine size.

This is required for the performance verification during the experimental stage. The shortest device length is maintained at 70 mm as per the swirl base model design following the aspect ratio calculation for length to suit the engine inlet port size. In fact, the simulation run 2 combinations is purposely taken from one of the swirl base models selected based on the early-stage results.

4.5.1 Simulation of Parametric Study Combinations

Table 4.9 shows the comparison of the air stream path lines for the selected parametric model. In general, the simulation of the swirl parametric model provides five different maximum velocity magnitudes that occur in the flow domain at 13.6107 m/s, 14.8814 m/s, 66.2019 m/s, and 128.701 m/s for simulation runs 2, 7, 44 and 49 blades respectively. Based on these results, increasing the draft angle tremendously affects the outlet velocity due to the relative bulk motion of the air intake charge, causing the 'ram' effect to dominate the flow field.

Table 4.9: Swirl Parametric Model Streamline Flow Results

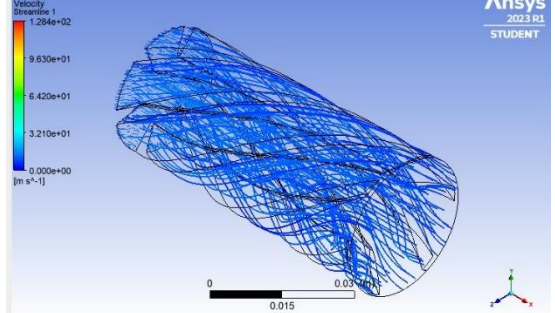
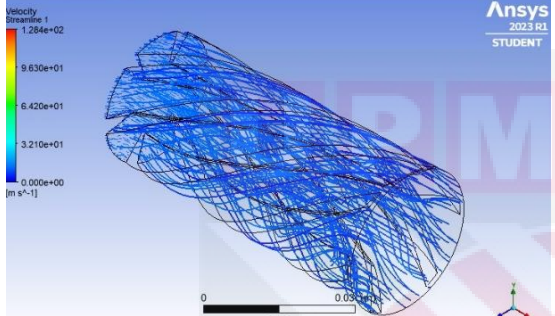
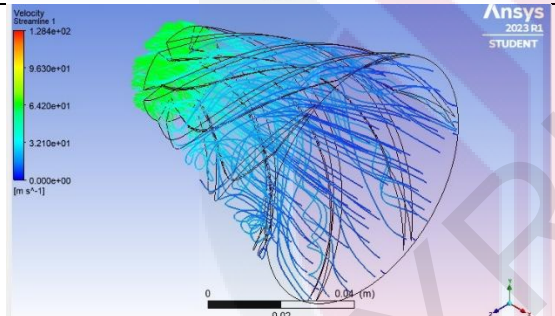
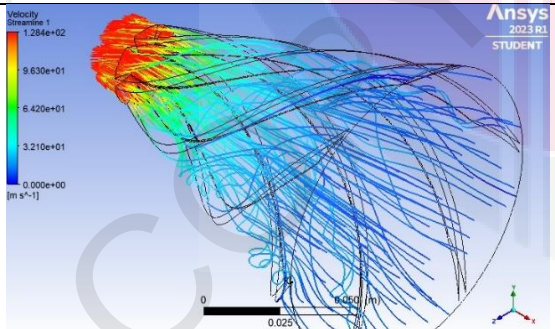
Streamline Flow Results		Details
		Model: Swirl Parametric Draft Angle: 1.26° Length: 70 mm Parametric Simulation Run: 2 Max. Velocity-Outlet: 13.6107 m/s
		Model: Swirl Parametric Draft Angle: 1.26° Length: 120 mm Parametric Simulation Run: 7 Max. Velocity-Outlet: 14.8814 m/s
		Model: Swirl Parametric Draft Angle: 18° Length: 70 mm Parametric Simulation Run: 44 Max. Velocity-Outlet: 66.2019 m/s
		Model: Swirl Parametric Draft Angle: 18° Length: 120 mm Parametric Simulation Run: 49 Max. Velocity-Outlet: 128.701 m/s

Table 4.10 shows the comparison of the velocity vector (outlet) for the selected parametric model. In general, the simulation of the swirl parametric model provides a different average velocity magnitude and swirl intensity at 8.488 m/s

($S=0.483$), 8.671 m/s ($S=0.32$), 56.665 m/s ($S=0.687$) and 115.323 m/s ($S=0.502$) for simulation run 2, 7, 44 and 49 respectively.

Table 4.10: Swirl Parametric Model Vector (Outlet) Flow Results

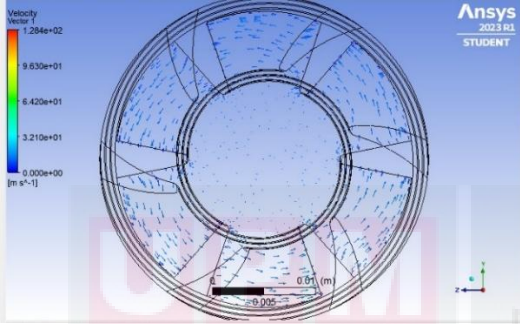
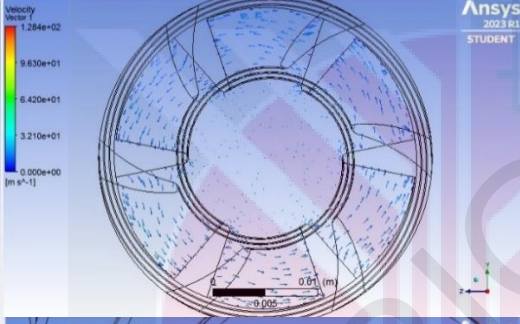
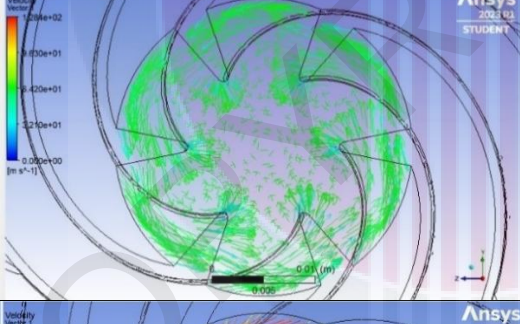
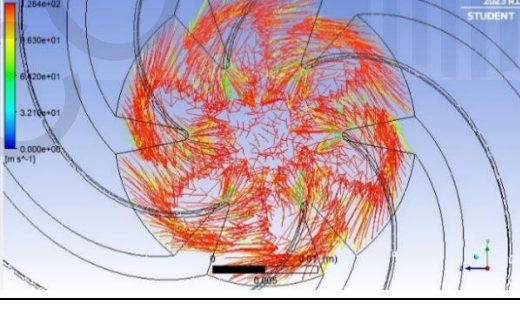
Vector (Outlet-Blade Exit) Flow Results	Details
	Model: Swirl Parametric Draft Angle: 1.26° Length: 70 mm Parametric Simulation Run: 2 Avg. Velocity-Outlet: 8.488 m/s Swirl Number: 0.483
	Model: Swirl Parametric Draft Angle: 1.26° Length: 120 mm Parametric Simulation Run: 7 Avg. Velocity-Outlet: 8.671 m/s Swirl Number: 0.32
	Model: Swirl Parametric Draft Angle: 18° Length: 70 mm Parametric Simulation Run: 44 Avg. Velocity-Outlet: 56.665 m/s Swirl Number: 0.687
	Model: Swirl Parametric Draft Angle: 18° Length: 120 mm Parametric Simulation Run: 49 Avg. Velocity-Outlet: 115.323 m/s Swirl Number: 0.502

Figure 4.15 below shows the trendline of the swirl velocity against the studied parameters involving the data for all 49 parametric combinations. Figure 4.15 (a) represents the trendline against the device length, while Figure 4.15 (b) represents the trendline against the device draft angle. Based on both trendlines, the average velocity magnitude is directly proportional to the increase in the device length and draft angle. From the coefficient of determination R^2 value predicted from the trendline and the distribution of data, it is also observed that the data for the draft angle tend to be closer to the trendline ($R^2 = 0.7655$) compared to the data for length ($R^2 = 0.1183$).

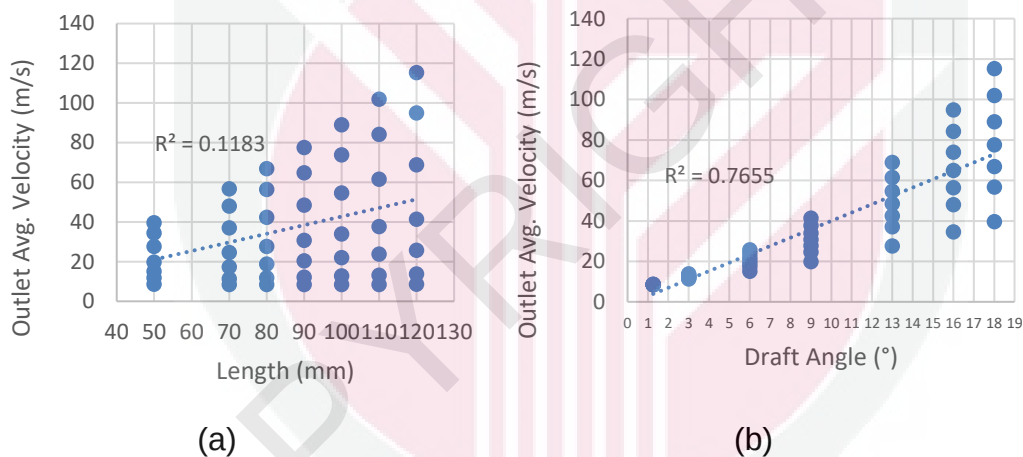


Figure 4.15: Trendline of the average velocity (outlet); a) Avg. velocity against length, b) Avg. velocity against draft angle

In essence, when the R^2 value is close to 1, it indicates that the trendline is a good fit for the data, suggesting that a large proportion of the variance in the dependent variable on the Y-axis can be explained by the independent variable on the X-axis. The R^2 value can be used to find the coefficient of correlation R which the value of zero indicates no correlation between two variables. This means the value does not seem to be linked at all. The value

of 1 or -1 indicates either a perfect positive correlation or a perfect negative correlation, respectively. It can also be classified as very highly correlated when the R whose magnitude is between 0.9 to 1.0, while 0.7 to 0.9 indicate a highly correlated, 0.5 to 0.7 indicates a moderately correlated, 0.3 to 0.5 indicates a low correlation, and less than 0.3 indicate a little correlation if any (Calkins, 2005). For this case, the data points tend to be closer to the trendline, indicating a stronger relationship with a high correlation between the average velocity and the draft angle ($R = 0.87449$), compared to the device length ($R = 0.3439$) which has a low correlation. This means that the velocity vector magnitude at the outlet is very much affected by the change in the draft angle.

Figure 4.16 shows the trendline of the swirl number (intensity) against the studied parameters involving the data for all 49 parametric combinations. Figure 4.16 (a) represents the trendline against the device length, while Figure 4.16 (b) represents the trendline against the device draft angle.

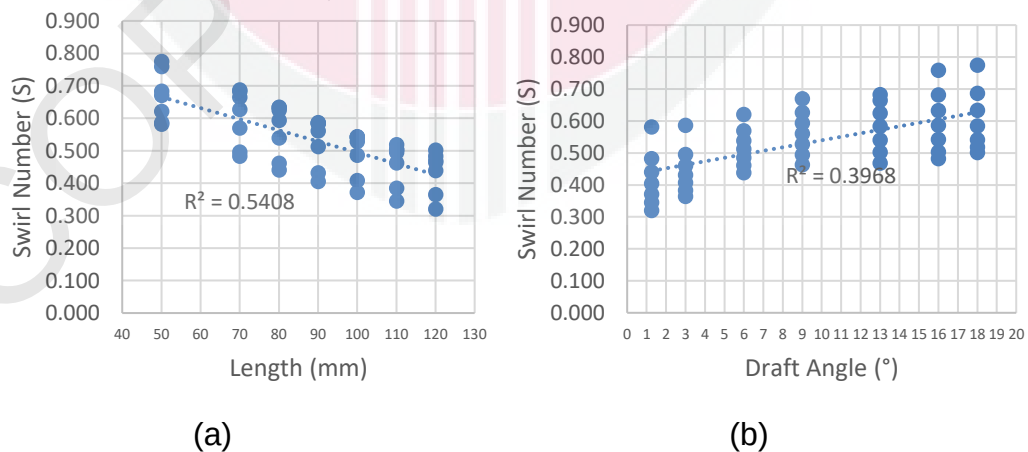


Figure 4.16: Trendline of the swirl intensity (outlet); a) Swirl intensity against length, b) Swirl intensity against draft angle

Based on the trendline against the length, the swirl number is inversely proportional to the increase in the device length. This means, that with an increase of length, the swirl intensity at the outlet becomes weaker (reduced swirl number). Meanwhile, the trendline against the draft angle shows the swirl number is directly proportional to the increase in the draft angle. From the coefficient of determination R^2 value predicted from the trendline, it is also observed that the data for length ($R^2 = 0.5408$, $R = 0.7354$) has a stronger relationship (high correlation) with the swirl intensity compared to the data for draft angle ($R^2 = 0.3968$, $R = 0.6299$) which has a moderate correlation.

Figures 4.17 and 4.18 show the trendline of the maximum TKE and average TED at the outlet against the studied parameters involving the data for all 49 parametric combinations. Figures 4.14 (a) and 4.15 (a) represent the trendline against the device length, while Figures 4.14 (b) and 4.15 (b) represent the trendline against the device draft angle.

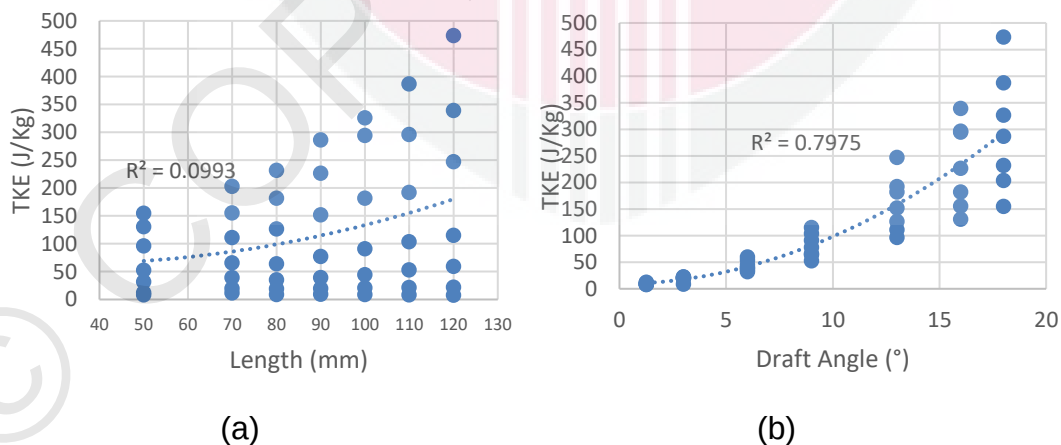


Figure 4.17: Trendline of the Maximum TKE (outlet); a) Maximum TKE against length, b) Maximum TKE against draft angle

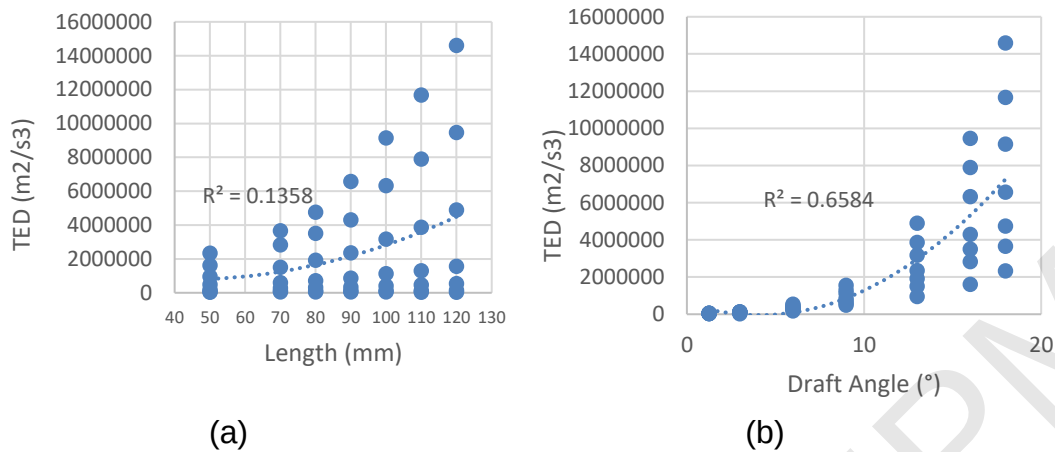
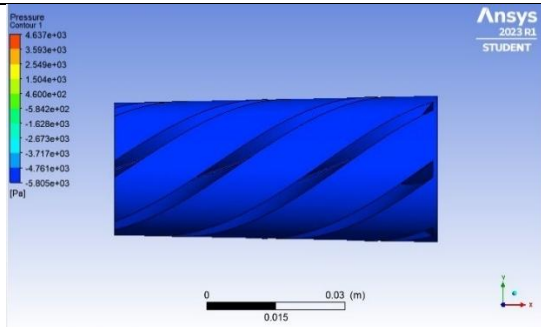
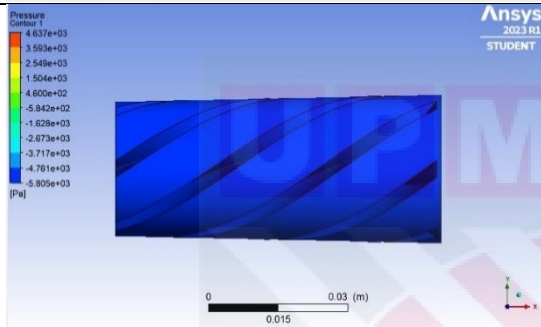
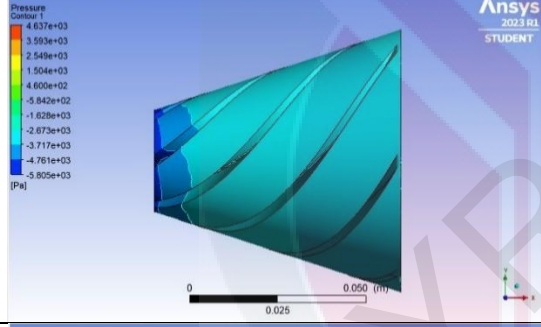
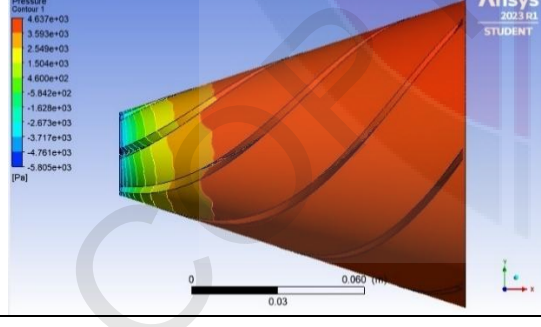


Figure 4.18: Trendline of the Average TED (outlet); a) Average TED against length, b) Average TED against draft angle

Based on the trendlines, both maximum TKE and average TED are proportional to the increase of the device length and draft angle. This means, that with an increase in length and draft angle, the maximum TKE and average TED are also increased. However, the magnitude of TKE and TED at the outlet is very much affected by the change in the draft angle rather than the length. From the coefficient of determination R^2 value, it is observed that the data of the draft angle for maximum TKE ($R^2 = 0.7975$, $R = 0.8930$) and average TED ($R^2 = 0.6584$, $R = 0.8114$) are highly correlated compared to the data of the length for maximum TKE ($R^2 = 0.0993$, $R = 0.3151$) and average TED ($R^2 = 0.1358$, $R = 0.3685$) which have a low correlation.

Table 4.11 shows the comparison of the pressure contour result against the selected parametric combinations. In general, the simulation of the swirl parametric model provides a large difference in maximum pressure at the inlet with the pressure difference calculated at 165.78 Pa, 184.33 Pa, 2737.46 Pa, and 10028.17 Pa for simulation runs 2, 7, 44 and 49 respectively.

Table 4.11: Swirl Parametric Model Pressure Contour Results

Pressure Contour Results	Details
	Model: Swirl Parametric Draft Angle: 1.26° Length: 70 mm Parametric Simulation Run: 2 Max. Pressure-Inlet: -5225.22 Pa Max. Pressure difference: 165.78 Pa
	Model: Swirl Parametric Draft Angle: 1.26° Length: 120 mm Parametric Simulation Run: 7 Max. Pressure-Inlet: -5206.67 Pa Max. Pressure difference: 184.33 Pa
	Model: Swirl Parametric Draft Angle: 18° Length: 70 mm Parametric Simulation Run: 44 Max. Pressure-Inlet: -2653.54 Pa Max. Pressure difference: 2737.46 Pa
	Model: Swirl Parametric Draft Angle: 18° Length: 120 mm Parametric Simulation Run: 49 Max. Pressure-Inlet: 4637.17 Pa Max. Pressure difference: 10028.17 Pa

Clearly, from the pressure contour results shown in Table 4.11, the ‘ram’ effect due to the geometry of the swirl air intake device for the draft angle of 18° significantly contributes to the high-pressure gradient in the flow domain, creating a large pressure difference. Figure 4.19 below shows the trendline of the pressure difference against the studied parameters involving the data for all 49 parametric combinations. Figure 4.19 (a) represents the trendline against the device length, while Figure 4.19 (b) represents the trendline against the device draft angle.

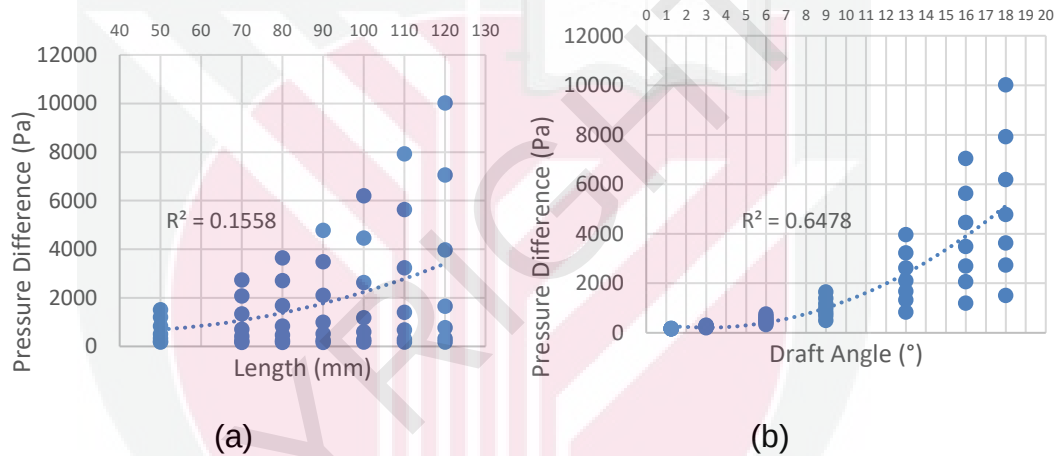


Figure 4.19: Trendline of the pressure difference (outlet); a) Pressure difference against length, b) Pressure difference against draft angle

Based on the trendline against the length and draft angle, the pressure difference is proportional to the increase of both variables. This means, that with an increase in length or draft angle, the pressure difference increases. From the coefficient of determination R^2 value, it is also observed that the data for draft angle ($R^2 = 0.6478$, $R = 0.8049$) has a stronger relationship (high correlation) with the pressure difference compared to the data for draft angle ($R^2 = 0.1558$, $R = 0.3947$) which has a low correlation.

The effect of the 'ram' pressure can also be seen in the simulated mass flow rate at the outlet and is very much affected by the change in the draft angle. Figure 4.20 below shows the trendline of the mass flow rate against all studied parameters. Based on the trendline against the length and draft angle, the mass flow rate is proportional to the increase of both variables. This means, that with an increase in length or draft angle, the mass flow rate increases. From the coefficient of determination R^2 value, it is also observed that the data for draft angle ($R^2 = 0.7446$, $R = 0.8629$) has a stronger relationship (high correlation) with the pressure difference compared to the data for draft angle ($R^2=0.1484$, $R = 0.3852$) which has a low correlation.

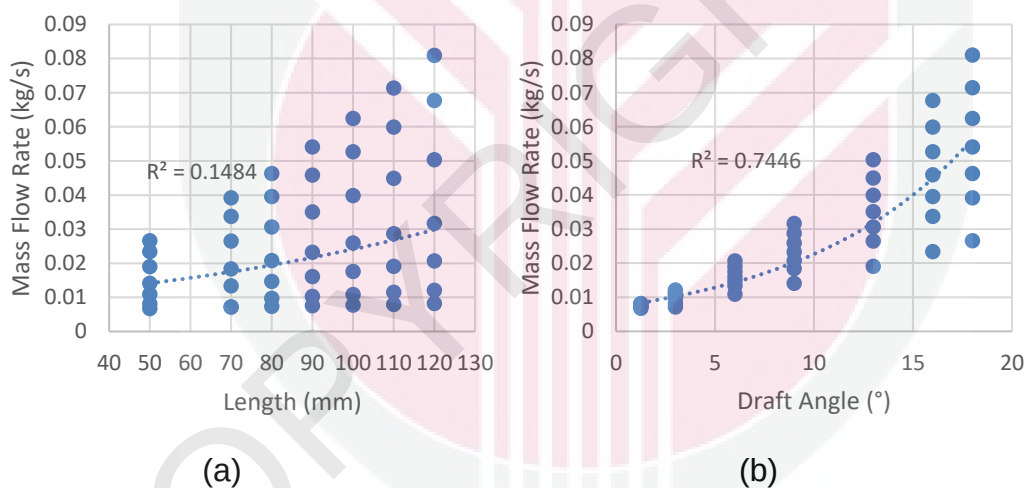


Figure 4.20: Trendline of the mass flow rate (outlet); a) Mass flow rate against length, b) Mass flow rate against draft angle

The 'ram' effect refers to the increase in air pressure and density that occurs during the intake stroke of the engine due to the dynamic action of the moving air. In this case, the draft angle has a primary effect on the increase of air pressure in the air intake charge. Consequently, the 'ram' pressure led to a

higher mass flow rate of air into the combustion chamber which can be seen in the above trendline presented in Figure 4.20 (b).

In summary, in terms of the flow behaviour, all the parametric models have a similar trait to the fluid flow of the previous swirl base models. It is observed that the path lines of the air stream flow smoothly over the blades and this is consistent for all combinations. The guided air streams form the swirl airflow inside the flow field and the velocity increases tremendously as it approaches the outlet of the device. A larger draft angle creates a greater impact on the relative bulk motion of the air charge, causing a greater amount of air received by the engine in bulk. The examples of the four parametric combinations streamline flow are presented in Table 4.9. For the velocity vector results, all parametric combinations show the increase of the average velocity at the outlet with the vector flow moving tangentially at the exit of the helical blade geometry as shown in Table 4.10. This indicates that there is a formation of swirl airflow with at least low or weak-swirl intensity.

Table 4.11 shows the contours of static pressure at the wall section along the device for the selected parametric model. It can be seen that the static pressure is greatly changed through the device area as the draft angle changes. This tremendous change of the static pressure indicates the 'ram' effect has been dominating the flow field due to the relative bulk motion of air intake charge by the larger draft angle design. Similar to the previous results of the swirl base model, the static pressure is found to be minimum at the outlet and maximum at the upstream location.

4.5.2 General Full Factorial Statistical Analysis

In General Full Factorial analysis, ANOVA analysis is done according to the two-competing hypothesis with the significance level as the decision to any true or false conclusion. To study the significance of the mean difference for draft angle, the analysis is done according to the following hypothesis:

H_0 : There is no mean difference in the studied response for the selected factor

H_1 : There is a mean difference in the studied response for the selected factor

In the full hypothesis, when the P-value is less than or equal to the significance level ($P\text{-value} \leq 0.05$), it is in the acceptance region of the H_1 hypothesis and the rejection of the H_0 (null) hypothesis. Hence, the selected factor is concluded as significant. On the contrary, when the P-value is more than the significance level ($P\text{-value} > 0.05$), it is in the acceptance region of the H_0 (null) hypothesis and the rejection of the H_1 hypothesis. Hence, it is concluded that the selected factor is not significant.

4.5.3 Swirl Number Response Analysis

The results of ANOVA analysis for the CFD result on the swirl number against the controlled factors are summarised in Table 4.12. The ANOVA table specifies the P-value for all the studied factors. Overall, the 'Length' factor has the largest change and greatest effect on the means response with a significant P-value of 0.000 based on the ANOVA model. The 'Draft Angle' factor is also significant with a P-value of 0.000. For the interaction effect between these two factors, the P-value could not be calculated due to the

saturated model. No error term could be estimated from a saturated model, while no model reduction can be performed with the existing factors.

Table 4.12: ANOVA analysis results for the swirl intensity response

Source	Degrees of Freedom	Sum of Squares	Mean Square	P-value
Draft Angle	6	0.23052	0.038420	0.000
Length	6	0.28678	0.047797	0.000
Error	36	0.01085	0.000301	
Total	48	0.52815		

In interpreting the behaviour of the combined parameters to the response, the Main Effects Plot and Interaction Plot are analysed. As shown in Figure 4.21, the plot shows the main effect across the parameter level for two factors specified in the study. Both factors have a great effect on the strength of the swirling airflow based on how they respond to the change in the factor's level. The effect can be concluded as significant since the P-value is less than the Significance Level ($P\text{-value} < 0.05$). In terms of the factor's interaction, Figure 4.22 shows that no interaction effect on the swirl intensity between these factors as the slopes of each line are parallel to each other, showing no interaction exists. In addition, there is no crossing between the lines. This concludes that there is no interaction between 'Length' and 'Draft Angle' that can affect the swirl intensity.

It is important to note that, 'no interaction' between these factors does not rule out the factors from the study. The individual factor effect is significant based on their P-value, which means any effect on the swirl intensity has no influence by the other factor because there is no possible interaction between them.

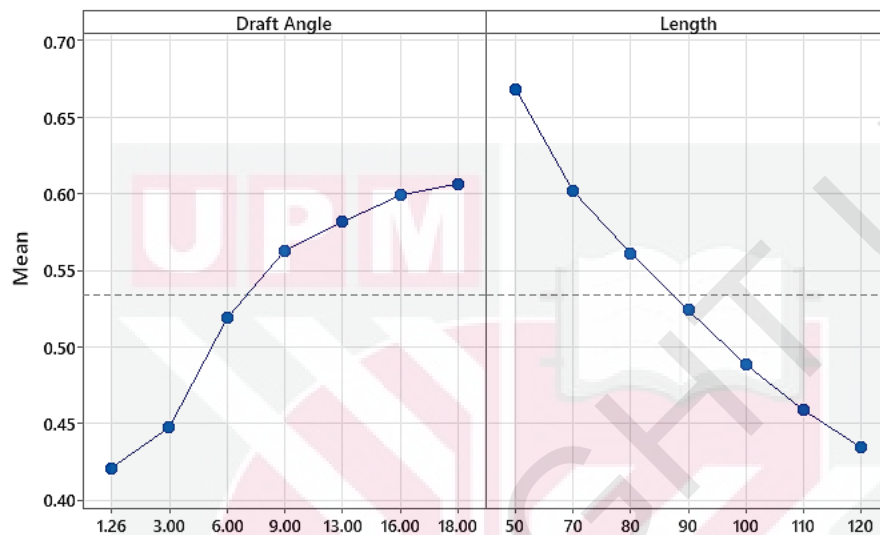


Figure 4.21: Main effect plot for Swirl Number (SN) – Data means

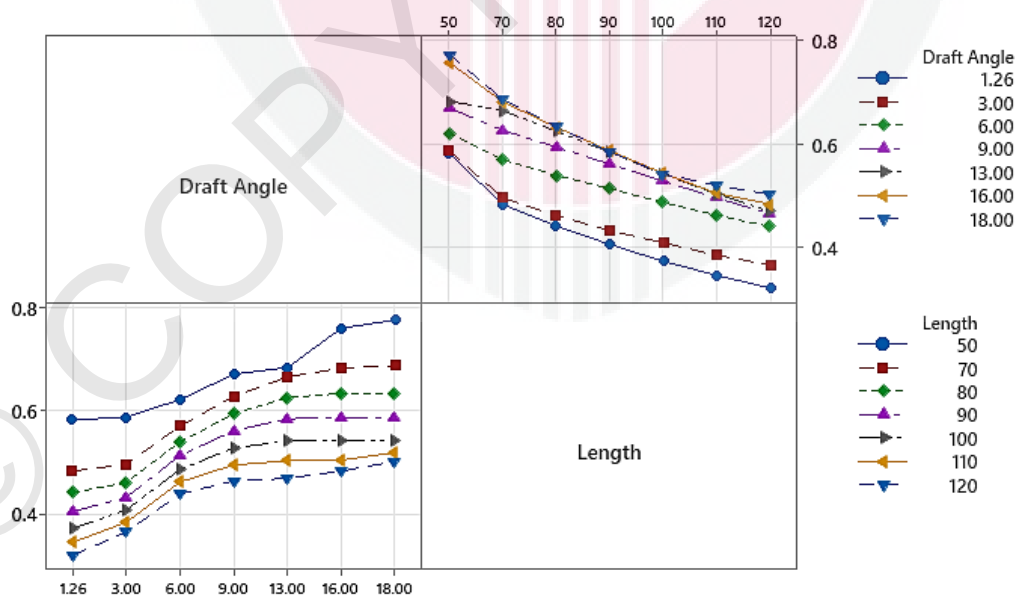


Figure 4.22: Interaction plot for Swirl Number (SN) – Data means

4.6 Summary

The flow behaviour of the swirl air intake manifold device is investigated through two stages of CFD simulation work. The simulation result is validated against the swirl device prototypes in actual engine running conditions. It can be concluded that the computed outlet velocity is in good agreement with the experimental result in actual engine conditions. This includes the simulation results from two different stages employing a different mesh type, which are comparable to the actual measurement.

In the 1st stage simulation, it is found that the outlet velocity and swirl intensity increased proportionally with the increase of blade revolutions. Meanwhile, increasing the blade numbers proportionally increases the outlet velocity but not the swirl intensity. In addition to the maximum TKE and the average TED results, the 1st stage simulation further suggests the recommended parameters level to be focused on in the next simulation stage. In the 2nd stage simulation, it is found that the swirl intensity behaves differently with the change of draft angle and length of the model, where the swirl intensity is directly proportional to the draft angle, but inversely proportional to the device length.

The ANOVA test further concludes that the effects from both factors are statistically significant to the strength of the swirl flow. This analysis has been verified in the experimental stage, where the parametric combinations contribute significantly to the engine performance, combustion and emissions due to the formation of the ram-swirled air intake. The findings from the experimental stage are thoroughly discussed in Chapter 5.

CHAPTER 5

EXPERIMENTAL PERFORMANCE AND EMISSIONS ANALYSIS OF A DIESEL ENGINE OPERATING WITH A SWIRL AIR INTAKE DEVICE

5.1 Introduction

This chapter presents the engine performance and emissions of a diesel engine operating with the swirl air intake device. is made in three parts. The first part discussed the result and analysis of the experimental run according to the base-swirl device, while the second part described the verification of the parametric-swirl device. The final part of this section summarises the comparison of vibration characteristics of all swirl air intake devices against the normal intake system.

The experiment of the swirl base device has been conducted according to the Taguchi robust methodology, where the factors and studied response are analysed for consideration in the subsequent tests. In the analysis of the vibration characteristics, the main focus is given to the severity and intensity of the engine vibration by analysing the RMS level of each swirl air intake device. The final section in this chapter presented the estimated measurement uncertainty calculated according to each measurement parameter.

5.2 Engine Test Conditions

The objective of the test is to investigate the performance and emissions of a single-cylinder diesel engine operating with the Swirl Air Intake Device. The comparisons have been made with the baseline engine operated with a normal

air intake without the swirl device. The engine operating conditions are summarized in Table 5.1, while the test fuel properties are presented in Table 5.2. The fuel properties have been re-tested at one of the accredited fuel test laboratories in Selangor, Malaysia to confirm the actual properties of the tested sample.

Table 5.1: Engine operating conditions

Test Mode	Air Intake System	Engine mode	Engine test speed (rpm)	Load (%)	Fuel type (Supplier)
Baseline engine	Normal	CI-DI	1800	Full (0 ~ 100)	Pure/Neat Diesel (KLK Oleo)
			2000		
			2200		
			2400		
Base-swirl engine	Base-Swirl Air Intake		2600		
			2800		
			3000		
			3200		
Parametric-swirl engine	Parametric-Swirl Air Intake		3400		
			3600		

Table 5.2: Test fuel (diesel) properties

Diesel property	Unit	Re-tested Value (ASTM Method)	Specification (No.2 ASTM 2D)
Density at 15 °C	kg/m ³	841	Min. 820 ² Max. 845 ²
Kinematic Viscosity at 40 °C	mm ² /s	3.650	Min. 1.9 ¹ Max. 4.1 ¹
Heating Value (HHV) / Gross Calorific Value	MJ/kg	44.022	Typ. 45.766 ³
Heating Value (LHV) / Net Calorific Value	MJ/kg	-	Typ. 42.791 ³
Cetane Number	-	55	Min. 40 ¹

Note:

¹ASTM-D975-23

²EN 590:2009

³Essom Co., Ltd

(Source: ASTM International, 2023; Essom, 2023; European Committee of Standardization (CEN), 2009)

5.3 Performance of diesel engine with the swirl base device

The analysis of the diesel engine performance with the swirl base device is divided into two parts. The first part of this section presents the comparison of the swirl-base performance against the normal intake. Three sets of combined parameters of 0.25, 0.5 and 0.75 revolutions which are represented by Helical-Base 1-D2, Helical-Base 2-D2 and Helical-Base 3-D2 respectively are used for comparison against the normal intake based on their overall performance level. The second part in Section 5.4 presents the statistical analysis based on the implemented Taguchi DOE methodology. For better visibility and readability, the y-axis range of the graphs presented in this chapter has been deliberately constrained, focusing on the relevant data points within the specified range.

5.3.1 Engine performance of the baseline against swirl base engine

The swirl airflow generated by the swirl device is expected to increase the fuel-air mixing rate in the cylinder compared to a normal airflow, thus further improving the fuel-air mixture for better combustion, leading to an increase in engine performance. This effect can be observed from the comparison graph for brake power and torque as shown in Figure 5.1 and 5.2, where all the swirl-base air intake device of the range 0.25 to 0.75 revolutions generally performs better or at least on par with the normal intake. The test at 3600 rpm resulted in the highest maximum brake power recorded by the 0.25 revolution at 3.62 kW with an approximately 2.9% increase compared to the normal intake. At higher load conditions, the largest increase provided by this device is

approximately 5.3% at the speed of 2000 rpm. This shows the ability of this device to improve the engine performance at high and low conditions.

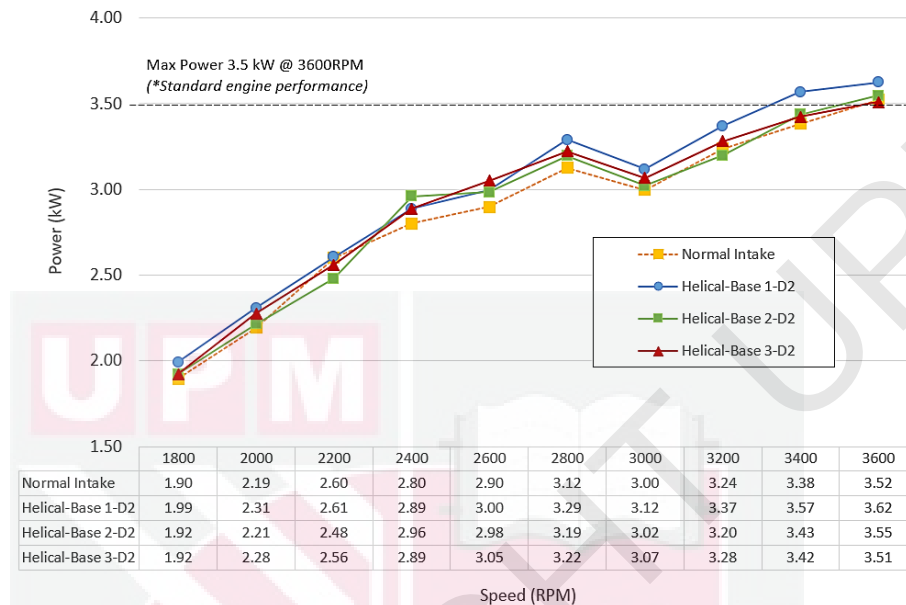


Figure 5.1: Comparison of engine performance - Brake Power for Base-Swirl Air Intake Against Speed

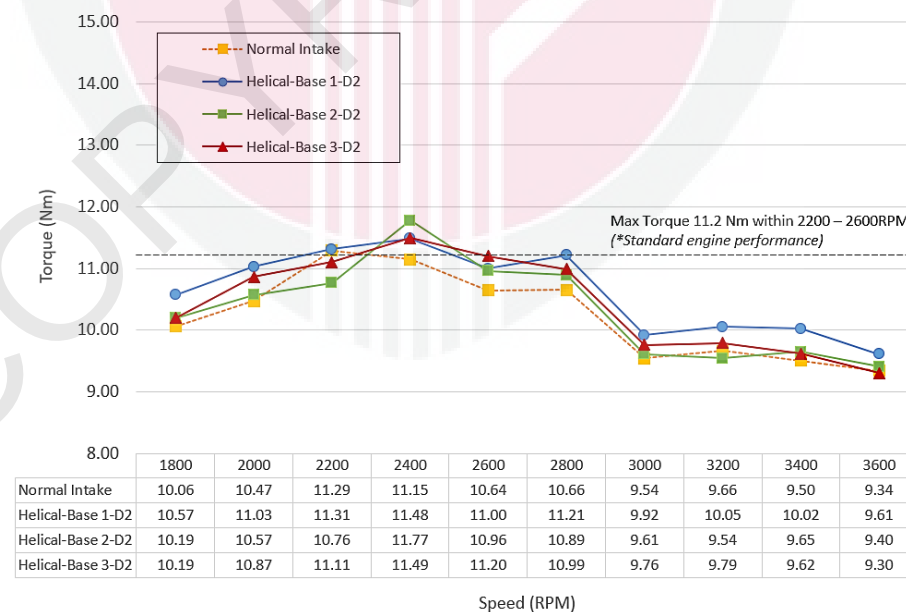


Figure 5.2: Comparison of engine performance - Brake Torque for Base-Swirl Air Intake Against Speed

Higher swirl intensity is assumed to increase the fuel-air mixing in the combustion chamber due to the increase of the fuel-air mixing rate. However, in the engine application, higher swirl intensity does not necessarily increase the actual performance due to the limitation of the effective airflow. The effective airflow in the intake manifold is very much affected by the increasing air resistance caused by an internal object, in this case, involving the inner blades. The more the blade revolves, the more air flows to be blocked by the blade direction which is almost normal to the flow direction. Higher air resistance will further reduce the airflow efficiency to the engine cylinder which becomes more significant at higher engine speed.

This can be seen in the comparison graph shown in Figures 5.3 and 5.4, where there is a decreasing trend in the brake power and torque at 3600 rpm with the increase of the blade revolutions. For example, the 0.75 revolution (Helical-Base 3) results in the lowest performance among the swirl device about 0.4% lower than normal intake after the 0.5 revolution (Helical-Base 2) which performs slightly better than normal intake about 1% higher. It shows that the greater air resistance due to the higher revolution in Helical-Base 3 is limiting the airflow efficiency even with the highest swirl intensity. The trend at the higher load conditions, however, is not consistent and conclusive where the performance of this engine could be more affected by various parameters such as operating temperatures.

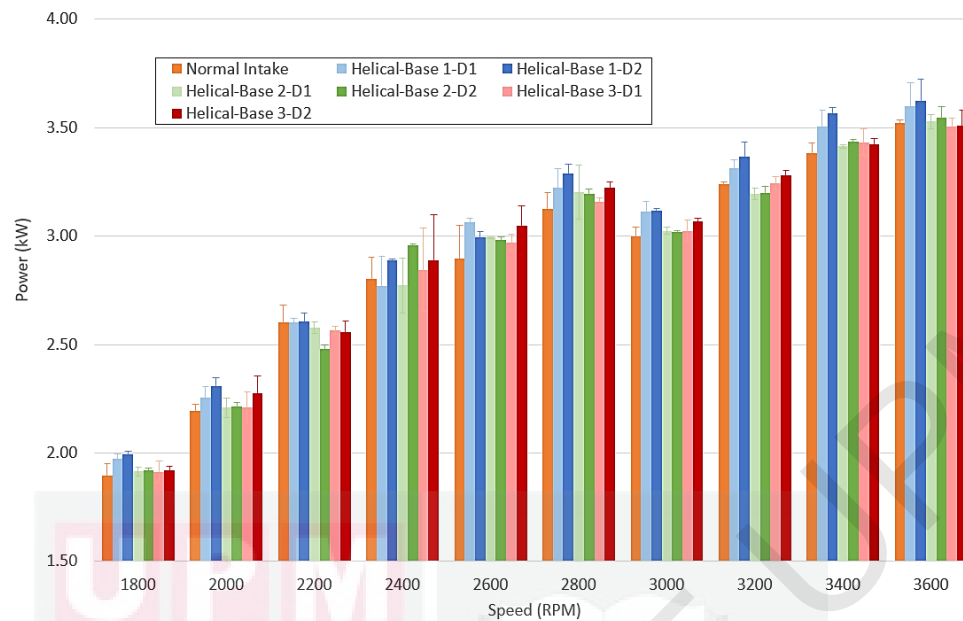


Figure 5.3: Comparison of engine performance - Brake Power for Base-Swirl Air Intake at Different Distance Against Speed

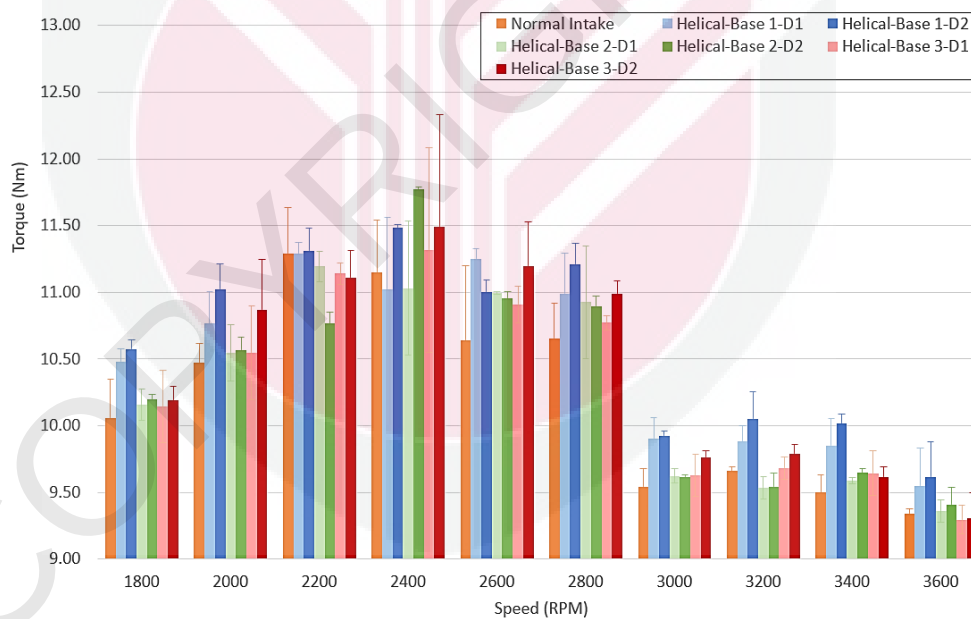


Figure 5.4: Comparison of engine performance - Brake Torque for Base-Swirl Air Intake at Different Distance Against Speed

From Figure 5.3 and 5.4, the result shows that many of the swirl devices at a 0.7 m distance (D2) could generally have better performance compared to a 0 m distance (D1). Still, throughout the overall comparison, the trend and

relationship of the distance towards the engine performance are not really clear and consistent. This indicates that the effect of the swirl device position in terms of the distance from the inlet port may not be significant. The significance level of this factor will be further explained and concluded in the statistical analysis section.

In terms of brake specific fuel consumption (BSFC) and engine efficiency (BTE), Figure 5.5 and 5.6 show both BSFC and BTE of the engine operating with the base-swirl devices are comparable against normal intake, where 0.5 revolution (Helical-Base 2) has better BSFC and BTE compared to normal intake with 229.22 g/kWh (2.4% BSFC reduction) and 37% (2.6% BTE improvement) respectively for speed 3600 rpm. In addition to the BSFC and BTE improvement, 0.5 revolution also performs better in the brake power compared to normal intake with approximately a 1% increase.

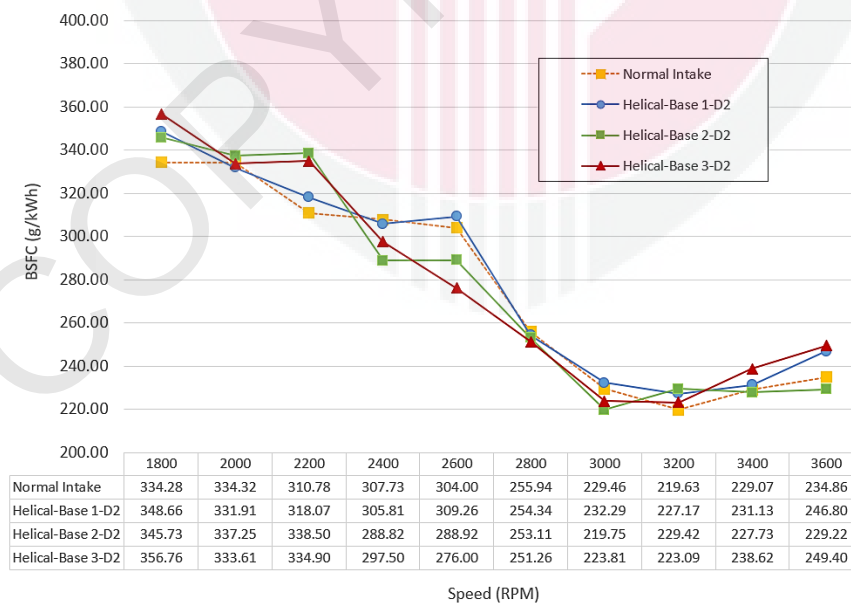


Figure 5.5: Comparison of engine performance - BSFC for Base-Swirl Air Intake Against Speed

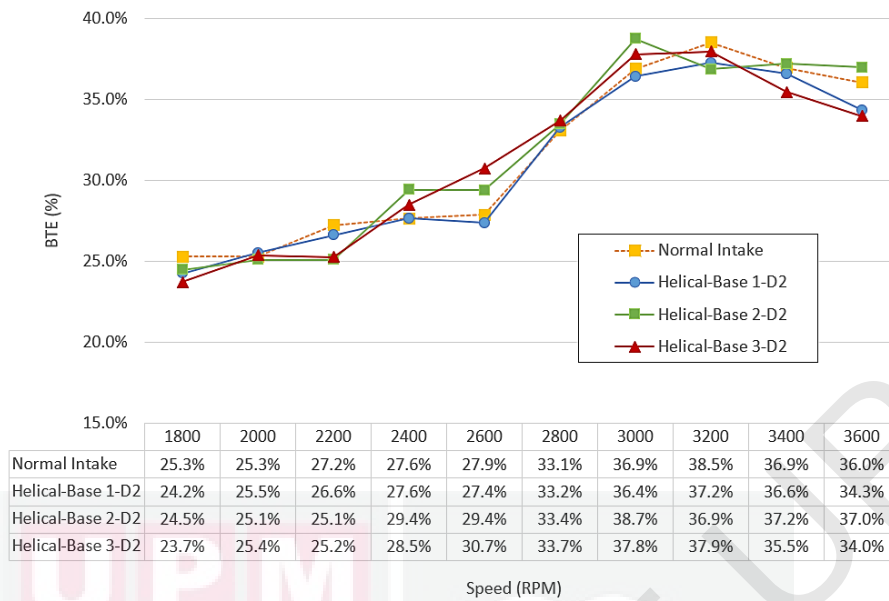


Figure 5.6: Comparison of engine performance - BTE for Base-Swirl Air Intake Against Speed

For 0.25 and 0.75 revolutions, although both of them perform differently in the maximum brake power, they result in lower BTE compared to normal intake following the higher BSFC recorded at 3600 rpm. This result is expected since BTE value can be directly attributed to higher BSFC, where BTE purely depends upon the fuel usage besides the engine design (S. Ramalingam & Rajendran, 2019). BSFC is inversely proportional to thermal efficiency which defines how effective the engine is in utilising the energy input (Yamin & Abu Mushref, 2019).

BTE indicates how efficiently the heat from the supplied fuel is converted into useful work. Results of BSFC and BTE at lower engine speeds are inconsistent, thus no conclusion could be drawn in relation to blade revolutions as the BSFC results could also be affected by various parameters such as the operating temperatures (e.g. air intake temperatures) (Baneasa et al., 2017).

Figure 5.7 and 5.8 present a different comparison which includes the swirl device location factor at different distances. Similar to the brake power analysis, the trend and relationship of the distance towards the engine performance are not clear and consistent. This indicates that the effect of the swirl device location in terms of the distance from the inlet port may also not be significant to the BSFC and BTE.

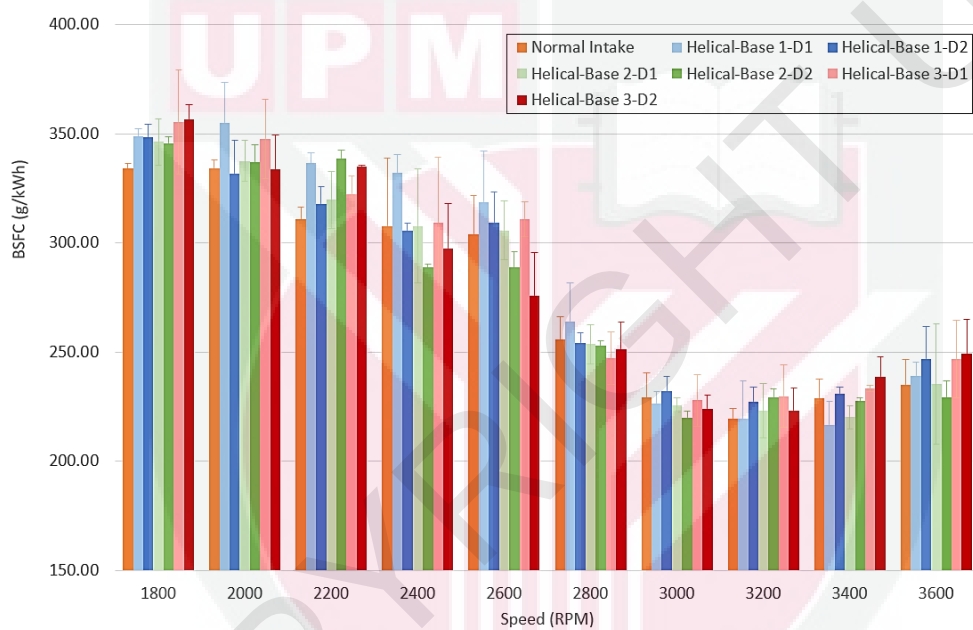


Figure 5.7: Comparison of engine performance - BSFC for Base-Swirl Air Intake at Different Distance Against Speed

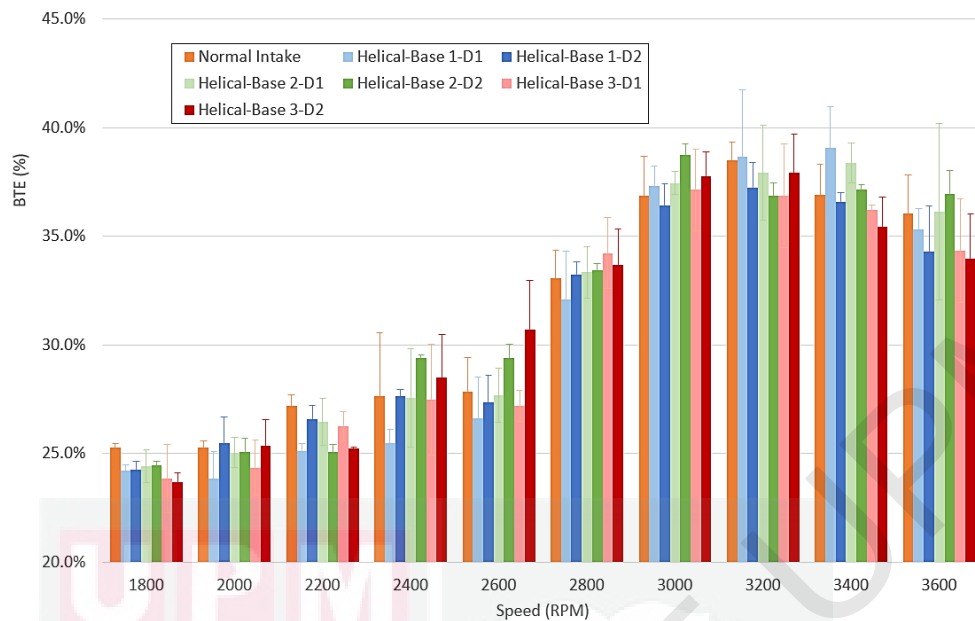


Figure 5.8: Comparison of engine performance - BTE for Base-Swirl Air Intake at Different Distance Against Speed

Break Mean Effective Pressure (BMEP) is calculated based on the engine brake power or torque. It is the average pressure that is acting on the piston throughout the cycle to produce the output power or work. BMEP is mainly suitable for comparative analysis. It is a normalized performance parameters measure that is useful to compare different displacements of the engine (Heywood, 2018). The purpose of this section is to generally compare the current performance level of the tested engine with the typical range to provide an insight into the current engine position compared to the other typical engines.

The typical range of BMEP for a naturally aspirated four-stroke diesel engine (passenger cars) was reported within 5 to 7.5 bar (Heywood, 1988). In more recent design and operating data, the typical BMEP has slightly increased within 7 to 9 bar as reported by Heywood (2018) due to the steady improvement in diesel engine performance and efficiency. As observed from

Figure 5.9 and 5.10, the BMEP of the tested engine is between 5.34 to 6.75 bar which is within the previous range and close to the recent typical range for passenger cars. The BMEP produced by the tested engine is expected to be slightly lower than the typical range due to some limitations. This engine uses mechanical fuel injection without any electronic control and feedback as commonly available in any passenger cars. Not to mention, the engine is an air-cooled type which is more applicable for a lighter engine application such as scooters, bikes and electric generators.

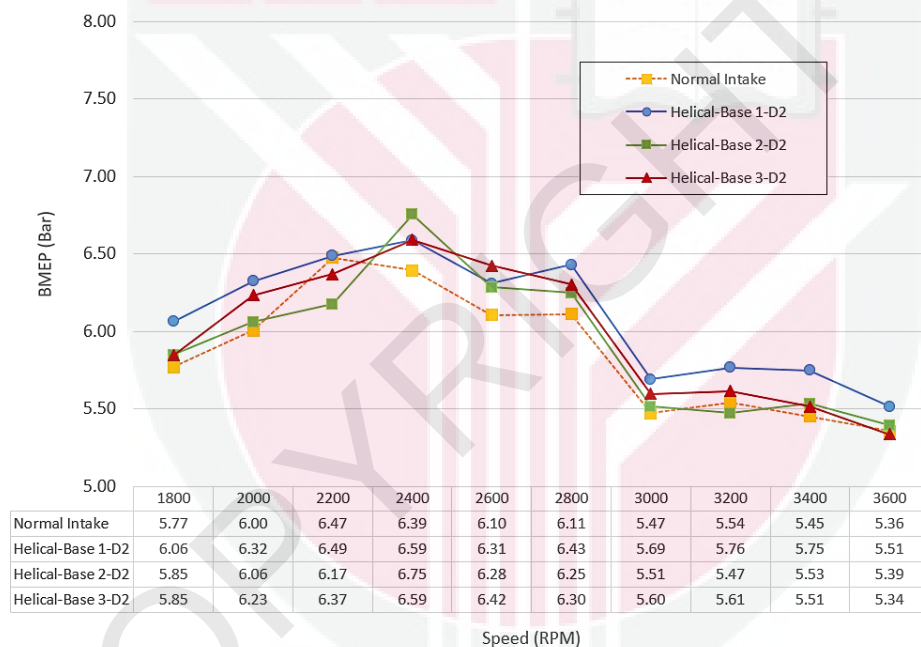


Figure 5.9: Comparison of engine performance - BMEP for Base-Swirl Air Intake Against Speed

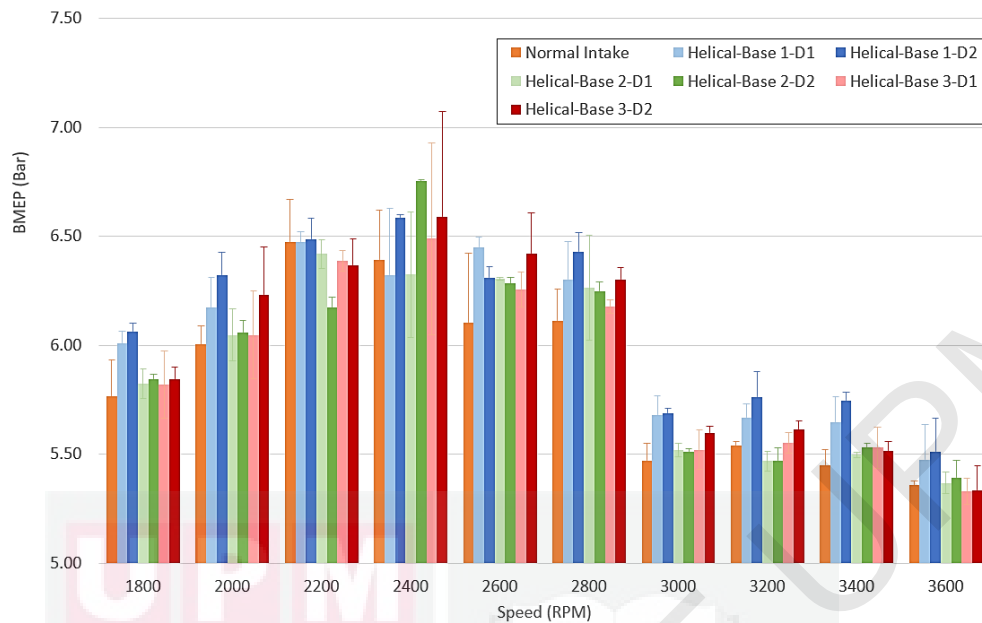


Figure 5.10: Comparison of engine performance – BMEP for Base-Swirl Air Intake at Different Distance Against Speed

Heywood (2018) also mentioned that the BMEP measures the engine's ability to obtain or provide high airflow and use it effectively over the full operating range. This is meant for an engine that is operated at all speeds at which the engine will be used with full throttle or with maximum fuel pump setting. As the current engine is tested according to this operating condition, the BMEP can further provide information on the engine air intake system's ability to provide high airflow and its effective use in the engine. Overall, the engine with swirl devices shows higher BMEP values mostly at every engine speed when compared to the normal intake.

At high load conditions, the tested engine operated with base-swirl devices shows an increase in the maximum BMEP at approximately 5.3%, while at lower load conditions the maximum BMEP has increased by 2.9%. For 0.25 revolution, BMEP is totally above the normal intake and generally performs better compared to 0.5 and 0.75 revolutions.

5.3.2 In-Cylinder Pressure, Pressure Rise Rate and Heat Release Rate Profiles

Diesel engines typically have two major stages of combustion known as rapid or premixed combustion and mixing-controlled combustion (Alrazen et al., 2016). The rapid combustion is counted from the end of the delay period or the beginning of the combustion until the premixed fuel and air during the ignition delay have been consumed. This can be best referred to the In-Cylinder Pressure - HRR comparison chart in Figure 5.11 (a) and 5.11 (b) for 1800 and 3600 rpm respectively. These figures illustrate the phase of rapid combustion between the initial sharp rise of the HRR profile until the lowest point after the highest peak. Once the fuel-air premix is consumed in this phase, the burning continues for the available fuel-air mixture in the next controlled combustion phase. This phase can be referred to Figure 5.11 (a) for high load conditions which shows the rise of the second HRR peak after the end of the premixed phase. The HRR may also not reach a second peak as mentioned by (Heywood, 1988) and this is shown in Figure 5.11 (b) for low-load conditions. The HRR is calculated based on PRR values, therefore, the PRR profiles will be identical to HRR.

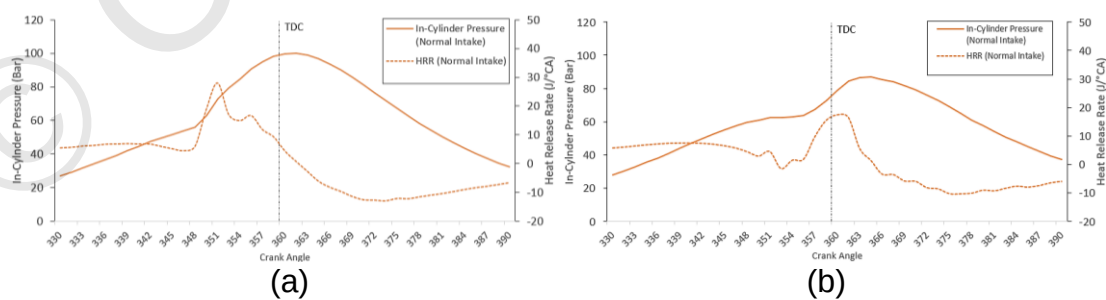


Figure 5.11: In-Cylinder Pressure and HRR Profiles for Normal Air Intake Against Crank Angle. (a) 1800 rpm – High load, (b) 3600 rpm – Low load

At high and low load conditions, the difference in the HRR peak is mainly due to the fuel-air homogeneity level within the combustion chamber. At high load conditions, the fuel spray is expected to be more atomised and therefore have better mixing with air, resulting in a more homogeneous fuel-air mixture, thus resulting in a higher HRR peak (Liang et al., 2019). Different conditions of the fuel-air mixture in the premixed combustion and controlled combustion phase also result in the first peak (rapid burning) and the second peak (remaining fuel burning) respectively following the time required to burn the fuel-air mixture completely.

At low load conditions, the fuel-air mixture is expected to be less homogeneous within the combustion chamber due to variations in combustion duration and fuel concentration. The only peak observed during low load conditions corresponds to the premixed or rapid combustion phase due to the rapid burning of the well-mixed fuel-air mixture right after the fuel injection. The subsequent peak of heat release rate during the controlled combustion is absent due to slow burning following the remaining fuel not mixed with air initially. More time is required for the fuel-air mixture to be burnt completely at this phase.

In comparing the normal intake with the swirl air intake, the trend is obvious at low load (3600 rpm) where all the swirl devices exhibit shorter combustion duration than that of the normal intake as shown by the In-cylinder pressure profiles in Figure 5.12 (b). Figure 5.13 (b) further shows the difference between them, where the early start of combustion and shorter ignition delay are observed for the swirl devices. These results are consistent with the earlier

simulation findings of the swirl air intake model capability to sufficiently produce swirl intensity, maximum TKE and lower average TED which promotes better homogeneity of the fuel-air mixture.

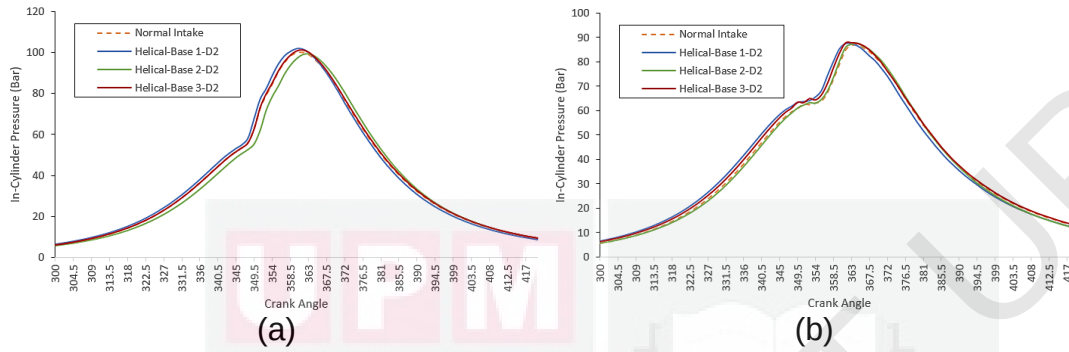


Figure 5.12: In-Cylinder Pressure Profile for Base-Swirl Air Intake Against Crank Angle. (a) 1800 rpm – High load, (b) 3600 rpm – Low load

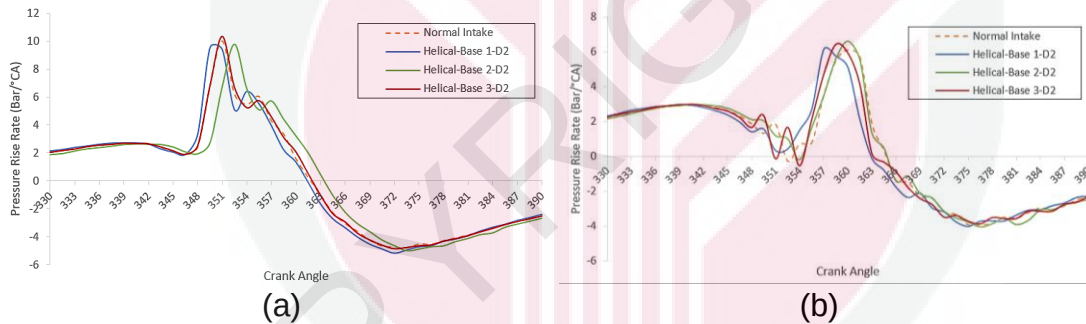


Figure 5.13: Pressure Rise Rate Profile for Base-Swirl Air Intake Against Crank Angle. (a) 1800 rpm – High load, (b) 3600 rpm – Low load

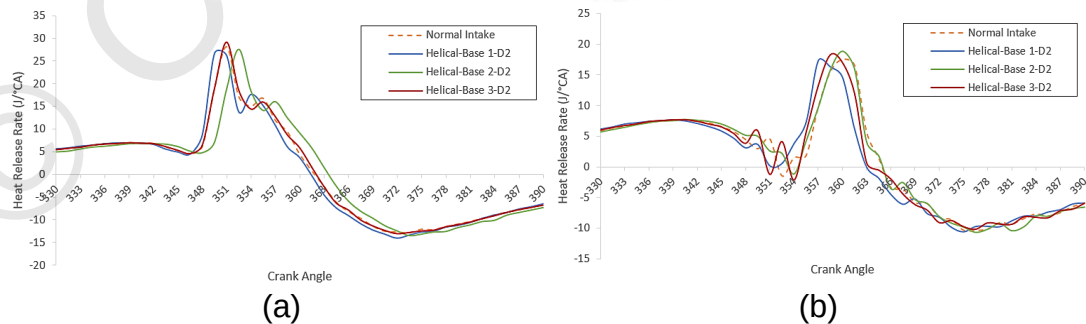


Figure 5.14: Heat Release Rate Profile for Base-Swirl Air Intake Against Crank Angle. (a) 1800 rpm – High load, (b) 3600 rpm – Low load

The shorter ignition delay can potentially improve combustion timing and more rapid energy release. Consequently, the impact of a shorter ignition delay may result in higher BMEP, allow for an early start of combustion, better utilization of the available fuel-air mixture and finally generate higher pressure on the power stroke. Early discussion on the brake power and BMEP results are consistent with this finding where the engine with swirl device can deliver higher brake power and BMEP at low load conditions, but subjected to the limitation factor of air resistance.

At high load conditions (1800 rpm), the in-cylinder pressure and HRR profile in Figure 5.12 (a) and 5.14(a) show almost similar patterns of early combustion. However, in this condition, the effect on the combustion behaviour may also be dominated by various factors such as engine and air intake temperature. It is observed during the test that the engine temperature at high load conditions increases rapidly, where controlling the temperature within the operating range is always needed. In fact, the tested engine depends only on the medium of air for cooling, so the variation of engine and air intake temperature could have affected the result in any way.

The full range of in-cylinder pressure results is also presented in Figure 5.15 and 5.16, where the maximum in-cylinder pressure is used to compare the peak pressure from each test condition. The maximum peak pressure generally corresponds to the power output level. Higher in-cylinder pressure indicates a greater energy release during combustion, which translates to increased brake power and torque. In terms of the engine's overall health, maximum in-cylinder pressure can provide an indication of knocking potentials

and other combustion problems such as compression loss due to internal leakage (valve or piston problems). This can be compared with normal engine operations to detect related engine problems.

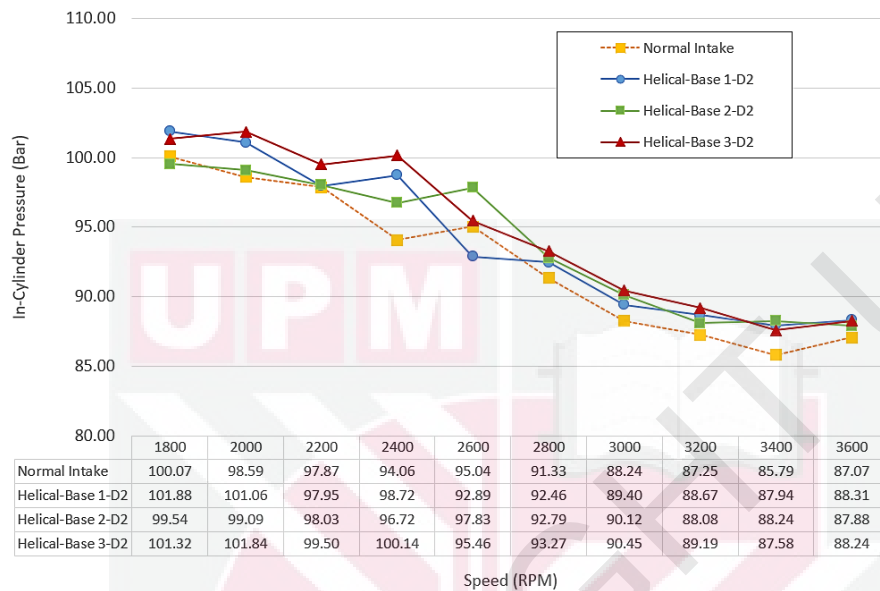


Figure 5.15: Comparison of Engine Performance - Maximum In-Cylinder Pressure for Base-Swirl Air Intake Against Speed

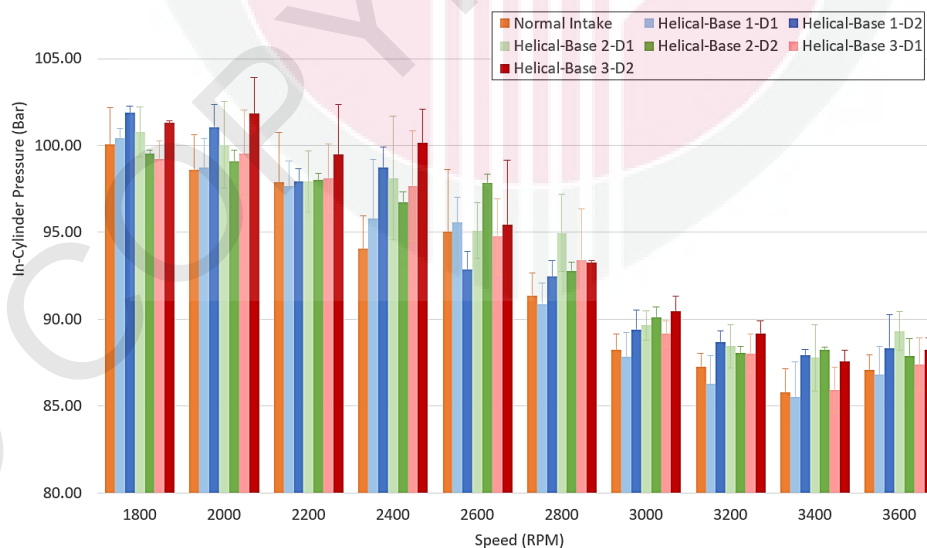


Figure 5.16: Comparison of Engine Performance - Maximum In-Cylinder Pressure for Base-Swirl Air Intake at Different Distances Against Speed

It can be observed from the comparison chart in Figure 5.15 and 5.16 that an engine with swirl devices generally exhibits higher maximum in-cylinder pressure mostly at every engine speed when compared to the normal intake without any abnormal trend. Generally, the higher maximum in-cylinder pressure in the swirl air intake device may suggest that this air intake system has better overall engine performance in terms of power, efficiency and torque compared to normal intake.

The maximum value of peak PRR and HRR can provide further information on the combustion characteristics and engine performance. The maximum PRR represents the peak rate of pressure rise inside the cylinder, whereas a higher maximum PRR indicates a more rapid pressure rise which may increase the brake power and torque. Meanwhile, the maximum HRR represents the peak rate at which energy is released during the combustion process, where a higher maximum HRR results in a greater release of energy due to more intense and rapid combustion. From Figure 5.17 to 5.20, the comparison of maximum PRR and HRR shows the engine with swirl devices generally has better combustion mostly at every engine speed compared to normal intake. This is consistent with the maximum in-cylinder pressure result discussed earlier.



Figure 5.17: Comparison of Engine Performance - Maximum Pressure Rise Rate for Base-Swirl Air Intake Against Speed



Figure 5.18: Comparison of Engine Performance - Maximum Pressure Rise Rate for Base-Swirl Air Intake at Different Distances Against Speed



Figure 5.19: Comparison of Engine Performance - Maximum Heat Release Rate for Base-Swirl Air Intake Against Speed

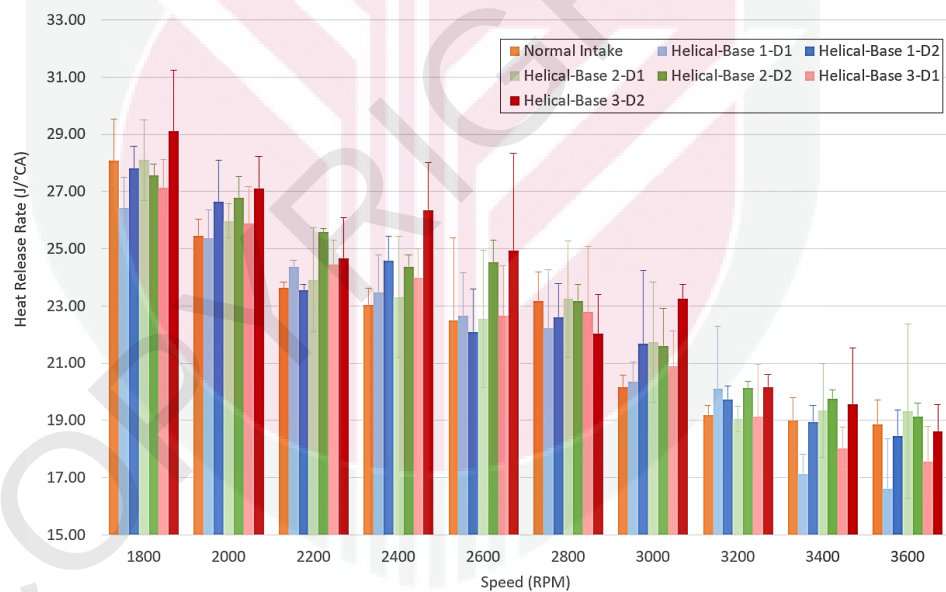


Figure 5.20: Comparison of Engine Performance - Maximum Heat Release Rate for Base-Swirl Air Intake at Different Distances Against Speed

5.3.3 Emission results

The fuel-air homogeneity in a diesel engine can have significant effects on exhaust emissions. As the swirl air intake charge tends to promote better fuel-air mixing in the cylinder, exhaust emissions are analysed to observed how the inducted air charge may affect the production of the exhaust emissions such as carbon monoxide (CO), carbon dioxide (CO₂), oxygen (O₂), nitrogen oxides (NO_x) and unburned hydrocarbon (UHC).

As mentioned by many researchers (Arjunraj et al., 2021; Bhat et al., 2018; Sadeq et al., 2019), carbon monoxide (CO) is a direct product of incomplete combustion, where the lack of oxygen content, short combustion duration and too rich fuel-air ratio are causing the incomplete combustion to happen. At different engine loads, carbon monoxide levels can significantly change due to the variation of the fuel-air ratio at either a lean or rich mixture. More fuel is required to respond to the greater power required by the high-load operation, causing a richer air-fuel mixture in the cylinder. In a rich fuel-air mixture, CO emissions tend to increase due to the lower supply of air limiting the availability of oxygen necessary for complete combustion. The CO is formed when carbon atoms from the fuel partially oxidise but do not combine with enough oxygen to form CO₂, instead the excess carbon atoms form the CO. On a contrary, a leaner mixture will bring about lower carbon monoxide in the exhaust emissions (Kesharvani et al., 2023). This is practically accurate with the experiment results where the CO level is greater at high load (1800 rpm) compared to low load (3600 rpm) conditions.

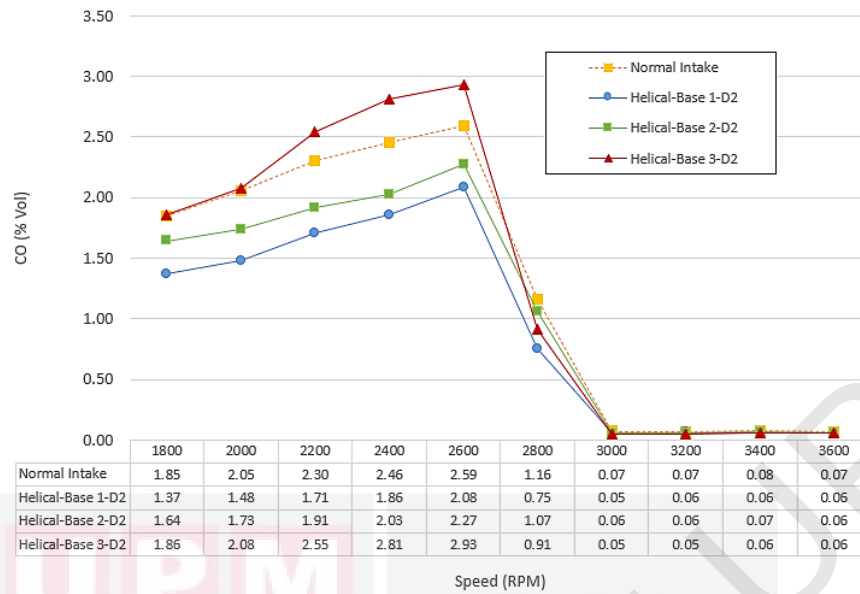


Figure 5.21: Comparison of Engine Emissions – CO for Base-Swirl Air Intake Against Speed

In a similar load condition, the effect on CO is somehow different when involved with a different inducted air charge characteristic. Based on the previous air-fuel ratio observation on the swirl devices, although the device parameters are affecting the level of air-fuel ratio, they do not totally change the air-fuel ratio profile since all of them are still at a lean mixture with excess air relative to the fuel. Specifically, a slight change to the air-fuel ratio has no significant effect on the CO emissions, instead, the homogeneity of the fuel-air mixture is now dominating the CO formation.

Figure 5.21 shows the comparison of the CO emissions for the selected base-swirl air intake. Against the normal intake, 0.25 and 0.5 revolutions with a sufficient swirl airflow result in about 34.8% and 18.18% lower CO emissions mainly due to the well-mixed fuel and air that ensures better combustion efficiency, reducing the formation of CO. For 0.75 revolution, the greater air resistance in the device restricts a more homogeneous fuel-air mixture from

being formed causing its CO level to be the highest among the others with 14.52% higher compared to normal intake.

Carbon dioxides (CO_2), on the other hand, are the result of the complete combustion of carbon in the fuel. As the primary combustion product, CO_2 is formed when the carbon in the fuel combines with oxygen given that sufficient oxygen is available. Theoretically, in case of more complete combustion occurring at one standard cycle, typically a higher CO_2 emission should be produced. Other than being affected by the homogeneity of the fuel-air mixture, CO_2 is primarily dependent on the fuel-to-air ratio and combustion efficiency.

The trend of CO_2 emissions for the swirl and normal intake is in agreement with the air-fuel ratio and the trend of CO emissions in most cases as shown in Figure 5.22. A lower air-fuel ratio than the normal intake tends to produce higher CO_2 emissions levels due to more complete combustion. In the case of the normal air intake combustion which runs on a lean mixture higher than the stoichiometric ratio, the incomplete combustion will be part of the process in the excess of air relative to the fuel. Since the base-swirl air intake has a slightly lower air-fuel ratio, the reduction of the excess air leads to more complete combustion, thus causing higher CO_2 emissions as can be seen in Figure 5.22. The highest CO_2 emissions level is represented by 0.5 revolution with 43% higher compared to normal intake.

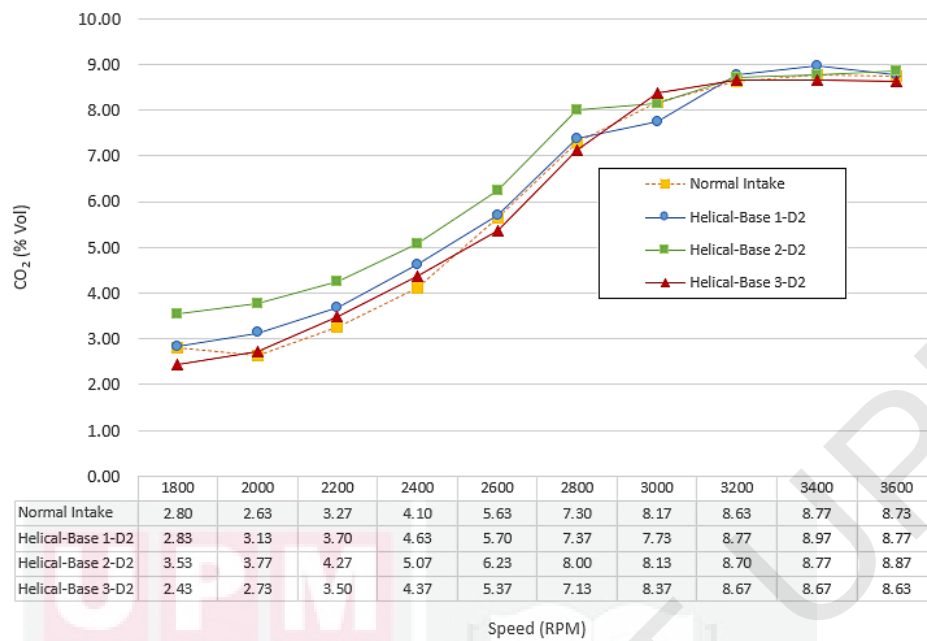


Figure 5.22: Comparison of Engine Emissions – CO₂ for Base-Swirl Air Intake Against Speed

The other revolutions provide almost similar or slightly above the normal intake. The CO₂ differs at high and low load, where higher CO₂ level is observed at low load conditions. This indicates that this engine is having more complete combustion at a low engine load. However, no clear trend in the CO₂ emissions level at similar load conditions that can be seen in relation to the fuel-air mixture homogeneity due to the base-swirl air intake application.

The oxygen (O₂) level is also presented in Figure 5.23, where at low load conditions, the engine produced a lower amount of O₂ in the exhaust gases.

This is in agreement with the higher CO₂ level at low load as the product of more complete combustion. More complete combustion means more effective use of air supply, leading to lower air residual in the exhaust gases. In a similar load condition, the O₂ level has a different effect when involved with a different inducted air charge characteristic where a homogeneous fuel-air mixture now

is dominating the O_2 formation. A higher fuel-air homogeneity may lead to a higher oxygen consumption during combustion, resulting in less air residual in the exhaust gases.

0.5 revolution is the most balanced design with the high swirl intensity as well as moderate air resistance compared to 0.25 revolution with the lower swirl and air resistance. These two revolutions generally have better results compared to 0.75 revolution with the highest swirl but suffered from the higher resistance. The effect of the design can be clearly seen in Figure 5.23 due to homogeneity where 0.5 revolution has the lowest O_2 level about 14.4% lower compared to normal intake, followed by 0.25 about 7.2%, and lastly 0.75 revolutions about 2.8%. 0.25 and 0.5 revolutions shall be selected for future consideration since they are capable of supplying highly effective swirl airflow which promotes a more homogeneous fuel-air mixture and higher mixing rate compared to normal intake.

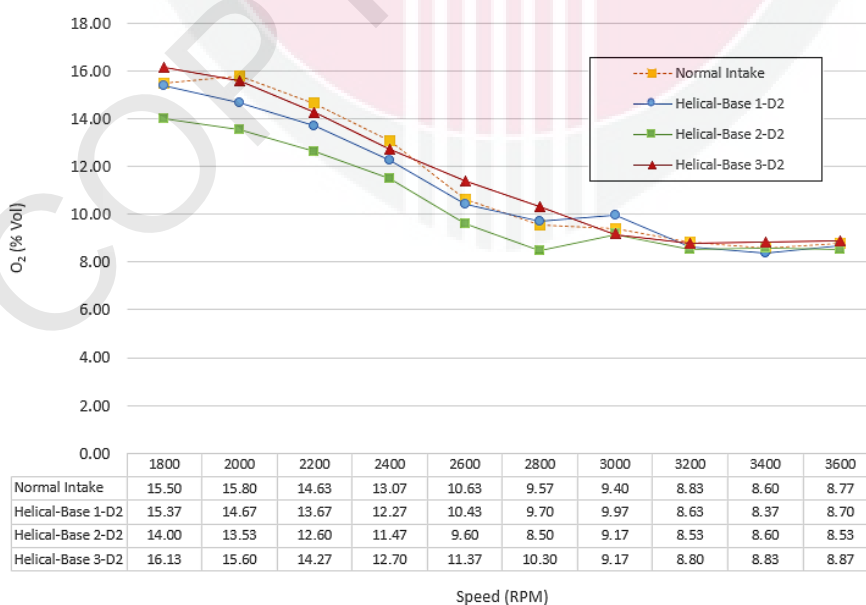


Figure 5.23: Comparison of Engine Emissions – O_2 for Base-Swirl Air Intake Against Speed

NO_x formation is generally associated with in-cylinder temperature, where it occurs at high temperatures by the nitrogen and oxygen in the air reacting to form nitrogen oxides at elevated combustion temperature. However, the NO_x produced in this experiment is not totally dependent on the combustion temperature as it is more likely to be dominated by the fuel-air mixture homogeneity and combustion efficiency through the application of swirl devices. In Figure 5.24, the 0.25 revolution recorded the lowest NO_x level among the others with 55.3% lower compared to normal intake, followed by 0.5 revolution with 30.6% lower. For 0.75 revolution, the NO_x level is somehow identical to the normal intake, where the higher and lower levels of NO_x are observed at certain engine speed.

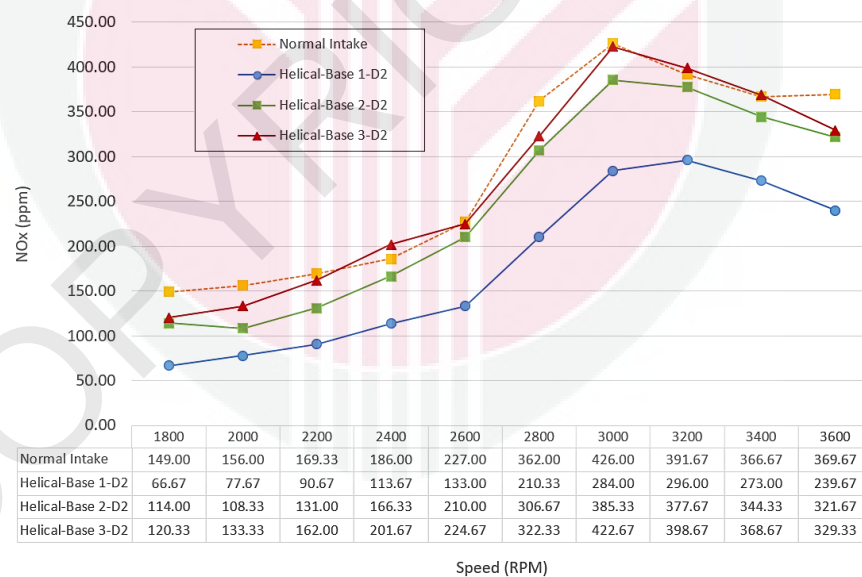


Figure 5.24: Comparison of Engine Emissions – NO_x for Base-Swirl Air Intake Against Speed

Besides providing sufficient swirl intensity, 0.25 and 0.5 revolutions produce minimal air resistance compared to 0.75 revolution. As discussed previously, all these factors contribute to a better homogeneity of the fuel-air mixture as well as increasing the fuel-air mixing rate. Better homogeneity increases combustion efficiency and minimizes the localized rich or lean regions. This led to more complete fuel oxidation and reduction in the partially burned fuel molecules that can contribute to NO_x emissions.

Meanwhile, a higher mixing rate facilitates a more rapid and thorough mixing of fuel and air. This significantly reduces the residence time or time limit for the nitrogen and oxygen to be exposed to the high combustion temperature. Both ways reduce the formation of NO_x in the exhaust gases. The effect of longer residence time that causes the domination of NO formation has been described by Heywood (2018) in explaining the behaviour of the flame front and post-flame combustion gases.

Unburned Hydrocarbons (UHC) refer to the hydrocarbon compound that has not undergone complete combustion. It is a typical emission element that is present in the exhaust gases based on the fact that the diesel engine will always be running in a lean condition above the stoichiometric ratio. Figure 5.25 shows a higher UHC emission at high load compared to low load conditions. This is due to the factor of fuel injection and reduction of quenching as mentioned by Sadeq et al. (2019) and Heywood (2018) that causes the UHC to be higher at high load conditions.

Following the previous discussion on the fuel-air mixing homogeneity resulting from the usage of the swirl devices, generally, all the swirl devices produce lower UHC emissions by 34.6% compared to normal intake. This is expected since the swirl air intake tends to promote better fuel-air mixing in the cylinder which causes more complete combustion that consequently reduces unconsumed UHC, thus leaving fewer unburned UHC in the exhaust gases (Sadeq et al., 2019). Figure 5.26 shows the summary of the exhaust emissions level at high and low load conditions.



Figure 5.25: Comparison of Engine Emissions – UHC for Base-Swirl Air Intake Against Speed

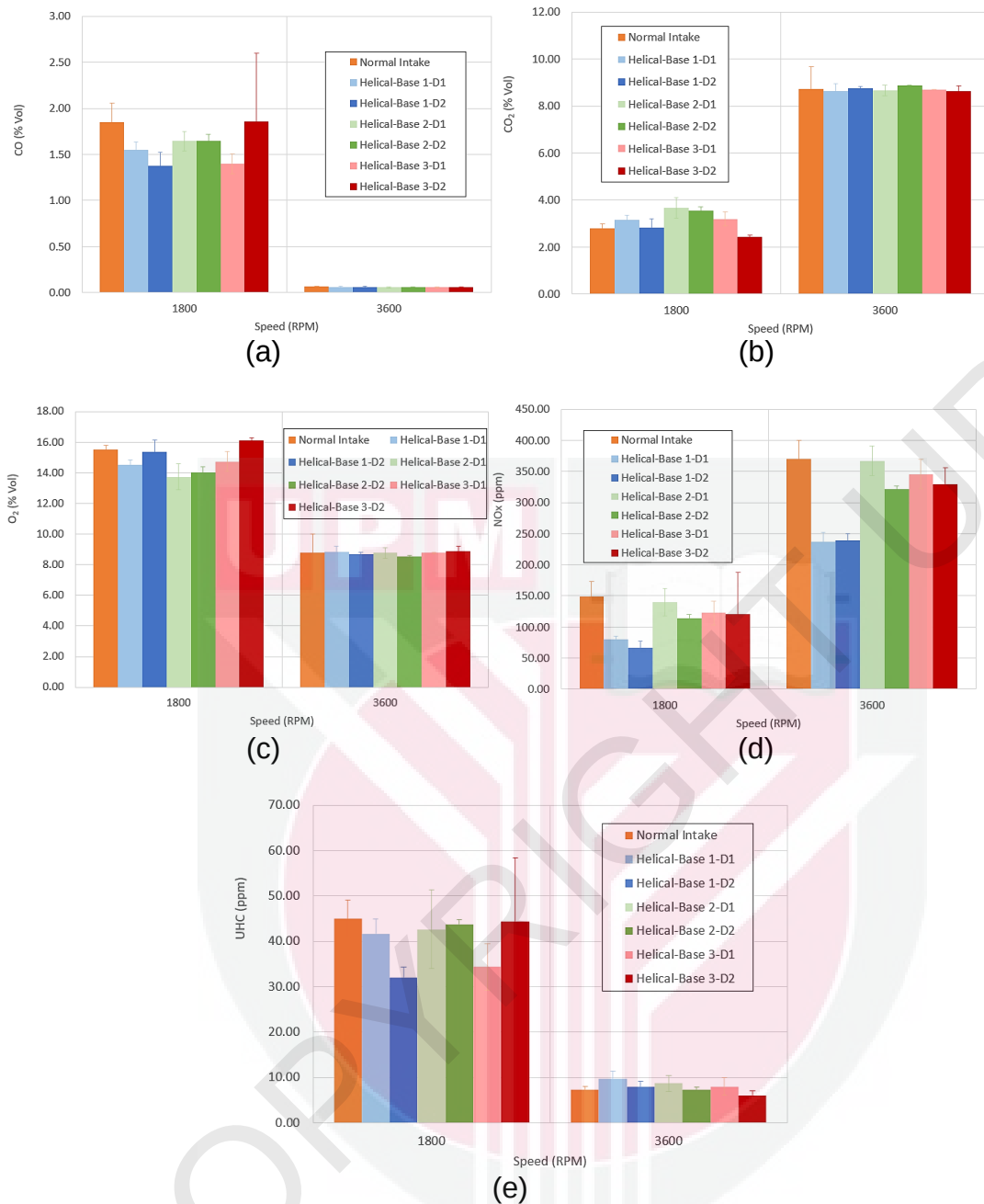


Figure 5.26: Comparison of Engine Emissions for Base-Swirl Air Intake at Different Distances Against 1800 and 3600 rpm, (a) CO, (b) CO₂, (c) O₂, (d) NO_x and (e) UHC

5.4 Taguchi Statistical Analysis

The experiment for the swirl base air intake device is conducted by employing the L_93^2 Orthogonal Array Design according to the Taguchi method (Section 3.4.11 Swirl Air Intake Device Experimental). Along with the helical blade revolution that changes the airflow direction, the device location has been selected as one of the factors to have experimented with following the possibility of the swirl decay within the air intake system. The significance level of these effects shall be concluded from the statistical results of the main and interaction effects between each factor. For this purpose, similar to the parametric study statistical analysis, the Analysis of Variance (ANOVA) is also used to statistically determine the significance of factors and quantification of the effects based on the established experimental plan. For this section, the ANOVA analysis only focuses on the brake power performance since the prime objective of using the swirl air intake device in this study is to improve the engine output performance.

In ANOVA analysis, the way how the P-value is calculated and compared against the significant value (α -level ≤ 0.05) is similar to the previous analysis in the parametric simulation study, where the P-value less than the significant value (α -level ≤ 0.05) indicates that the factor studied has a statistically significant effect on the response observed. The analysis is also done according to the following hypothesis, where the factor can be concluded as significant depending on the calculated P-value either in the acceptance region of the null hypothesis (H_0) or the other (H_1).

H_0 : There is no mean difference in the studied response for the selected factor

H_1 : There is a mean difference in the studied response for the selected factor

5.4.1 Engine Performance-Brake Power Response Analysis

The results of ANOVA analysis for the experimental result on the brake power against the controlled factors are summarised in Table 5.3 (Low load) and 5.6 (High load). The ANOVA table specifies the P-value for all the studied factors. Overall, the 'IM Helical Revolution' factor has the largest change and greatest effect on the means response at low and high load conditions as represented by the speed of the engine of 3600 and 1800 rpm respectively. At low load conditions, the effect from the 'IM Helical Revolution' is concluded as significant towards the brake power with a P-value of 0.010 based on the ANOVA test. Meanwhile, the 'IM Position' factor is concluded as not significant with a P-value of 0.060, where the value is higher than the significance P-value of 0.050.

Table 5.3: ANOVA analysis results for the Brake Power (at Low Load - 3600 rpm) response

Source	Degrees of Freedom	Sum of Squares	Mean Square	P-value
IM Helical Revolution	2	0.011877	0.005938	0.010
IM Position	2	0.004092	0.002046	0.060
Error	4	0.001328	0.000332	
Total	8	0.017297		

For the interaction effect between these two factors, the P-value could not be calculated due to the saturated model. No error term could be estimated from a saturated model, while no model reduction can be performed with the existing factors. In interpreting the behaviour of the studied factors to the response, the Main Effects Plot and Interaction Plot are analysed. Figure 5.27 shows the plot having two lines across the parameter level for two factors specified at 3600 rpm. It is observed that both response means are not similar across all factor levels, where the slope determines the magnitude of the main effect. Although the plot shows there is a main effect for both factors across the level, the ANOVA test estimates only 'IM Helical Revolution' is statistically significant since the P-value is less than the Significance Level ($P\text{-value} < 0.05$) as summarised by Table 5.3. In terms of the factor's interaction, Figure 5.28 suggests that there is an interaction effect between these two factors as the line is non-parallel to each other. This can be shown by the different slopes of the two lines.

Nevertheless, since the ANOVA test cannot estimate the significance value of the interaction factors due to no error term, there is no conclusion that can be drawn as to whether the interaction effects between 'IM Helical Revolution' and 'IM Location' are significant to the brake power or not. ANOVA is done to assess the significance of the differences between factors through the P-value obtained from the hypothesis test. The error term represents the random variability within each group. It is needed to estimate the P-value since the value is calculated based on the ratio of variability between the factor group and variability within the factor group.

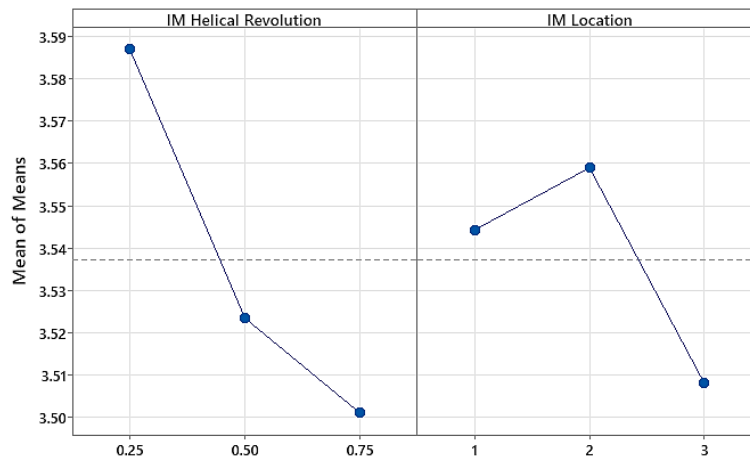


Figure 5.27: Main Effects Plot for Means (Brake Power at 3600 rpm)

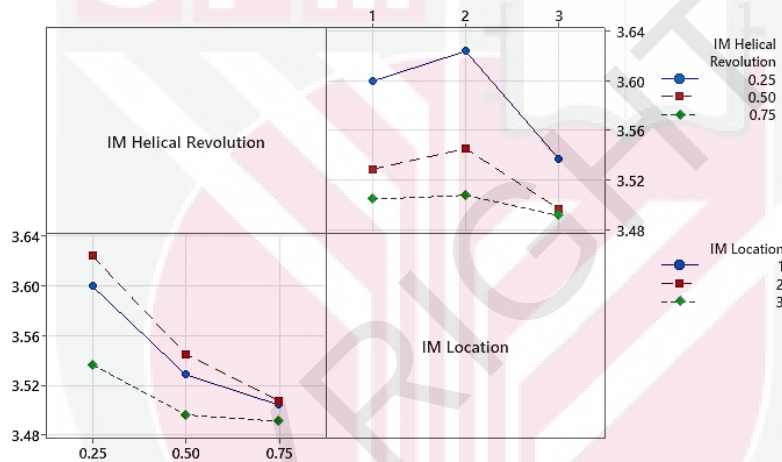


Figure 5.28: Interaction Plot for Means (Brake Power at 3600 rpm)

Further analysis of the factor's ranking, Table 5.4 and 5.5 shows the 'IM Helical Revolution' having the 1st ranking followed by 'IM Location' based on the largest difference in the response level and Signal-to-Noise (SN) ratio. In a Taguchi design, the SN ratio that is based on the Larger-the-Better will maximise the response of the study. Table 5.5 highlights the possible combination of the factor parameters that can produce the optimum brake power response, where the optimum parameters for the 'IM Helical Revolution' are at Level 1 and 'IM Location' at Level 2. In terms of the parameter's value, Level 1 'IM Helical Revolution' is 0.25 revolution, while Level 2 'IM Location' is

the Distance 2 from the inlet port. With the combination of the selected optimum parameters, the device is expected to produce greater brake power.

Table 5.4: Ranking for the response table of means (Brake Power at 3600 rpm)

Level	IM Helical Revolution	IM Location
1	3.587	3.544
2	3.524	3.559
3	3.501	3.508
Delta	0.086	0.051
Rank	1	2

Table 5.5: Ranking for the response table of signal-to-noise (SN) ratio at 3600 rpm

Level	IM Helical Revolution	IM Location
1	<u>11.09</u>	10.99
2	10.94	<u>11.03</u>
3	10.88	10.90
Delta	0.21	0.12
Rank	1	2
Note: IM Helical Revolution: - Level 1 (0.25), Level 2 (0.5), Level 3 (0.75) IM Location: - Level 1 (Distance 1), Level 2 (Distance 2), Level 3 (Distance 3)		

At high load conditions (1800 rpm), the effect from the 'IM Helical Revolution' is also concluded as significant towards the brake power with a significant P-value of 0.042 based on the ANOVA test. Meanwhile, the 'IM Position' factor is concluded as not significant with a P-value of 0.749, where the value is higher than the significance P-value of 0.050. For the interaction effect between these two factors, the P-value also could not be calculated due to no error term could be estimated.

Table 5.6: ANOVA analysis results for the Brake Power (at High Load - 1800 rpm) response

Source	Degrees of Freedom	Sum of Squares	Mean Square	P-value
IM Helical Revolution	2	0.004800	0.002400	0.042
IM Position	2	0.000192	0.000096	0.749
Error	4	0.001240	0.000310	
Total	8	0.006232		

For the Main Effects Plot, Figure 5.29 shows that both response means are not similar across all factor levels, where the slope determines the magnitude of the main effect. Although the plot shows there is a main effect for both factors, however, the slope change for 'IM Location' can be seen very much less than 'IM Helical Revolution'. Further ANOVA test concludes only 'IM Helical Revolution' is statistically significant since the P-value is less than the Significance Level ($P\text{-value} < 0.05$) as summarised by Table 5.3.

In terms of the factor's interaction, Figure 5.30 suggests that there is an interaction effect between these two factors as the lines are non-parallel to each other as well as cross. This can be shown by the different slopes of the two lines. Nevertheless, since the ANOVA test cannot estimate the significance value of the interaction factors due to no error term, again there is no conclusion that can be drawn whether the interaction effects between 'IM Helical Revolution' and 'IM Location' are significant to the brake power or not. Similar to low-load conditions, more studied factors could be added to estimate the interaction effect for future tests.

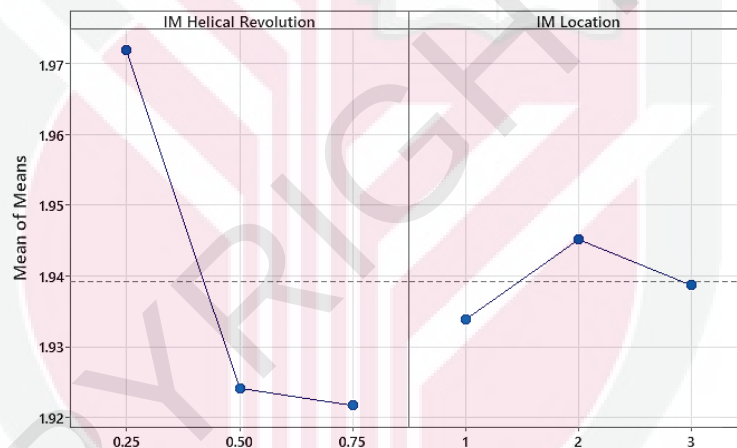


Figure 5.29: Main Effects Plot for Means (Brake Power at 1800 rpm)

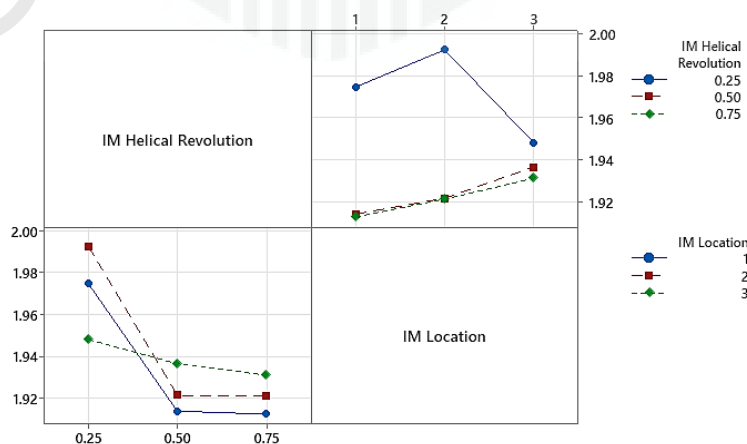


Figure 5.30: Interaction Plot for Means (Brake Power at 1800 rpm)

Further analysis of the ranking of the factor, Tables 5.7 and 5.8 shows the 'IM Helical Revolution' having the 1st ranking followed by 'IM Location' based on the largest difference in the response level and Signal-to-Noise (SN) ratio based on the Larger-the-Better. Table 5.7 highlights the possible combination of the factor parameters that can produce the optimum brake power response at high load conditions, where the optimum parameters for the 'IM Helical Revolution' is at Level 1 and 'IM Location' at Level 2, which is similar to the optimum parameters for low load condition. With the combination of the selected optimum parameters, the device is expected to produce greater brake power.

Table 5.7: Ranking for the Response Table of Means (Brake Power at 1800 rpm)

Level	IM Helical Revolution	IM Location
1	1.972	1.934
2	1.924	1.945
3	1.922	1.939
Delta	0.050	0.011
Rank	1	2

Table 5.8: Ranking for the Response Table of Signal-to-Noise ratio at 1800 rpm

Level	IM Helical Revolution	IM Location
1	<u>5.897</u>	5.728
2	5.685	<u>5.778</u>
3	5.674	5.750
Delta	0.223	0.050
Rank	1	2

5.5 Performance confirmation of diesel engine with the swirl parametric device (0.25 and 0.5 revolutions)

In this section, the diesel engine performance with the swirl parametric device is analysed with the comparison of the swirl-parametric performance against the normal intake. The tested parametric devices at the performance verification stage are according to the major parametric models in the simulation stage, where it is based on the four chosen parametric models of simulation run 2, 7, 44 and 49. The chosen models are taken considering the highest to smallest possible geometries suitable for the experimental engine size. The parametric simulation run 2, however, will not be part of the discussion since this device geometry is actually similar to the base-swirl device (Helical-Base 2) geometry and the effect of this parameter has been thoroughly discussed in the previous section.

In addition to the parametric geometry selection, the swirl-parametric device is also tested using a different blade revolution. Referring to the previous analysis, only 0.25 and 0.5 are tested in the second stage testing, while 0.75 revolution is omitted due to lower performance - brake power at low load (3600 rpm) with 0.4% less than that of the normal intake. Although it also results in slightly higher performance at higher loads, however, there is no clear improvement to the overall performance and emissions. Table 5.9 shows the summary of the parametric combinations that are tested in the parametric verification stage.

Table 5.9: Summary of the tested swirl-parametric combinations

Identification	Parametric Simulation run	Draft Angle (°)	Length (mm)	Helical Revolution
Helical-P 1	7	1.26	120	0.25
Helical-P 2	44	18	70	
Helical-P 3	49	18	120	
Helical-P 4	7	1.26	120	0.50
Helical-P 5	44	18	70	
Helical-P 6	49	18	120	

5.5.1 Engine Performance of the baseline versus swirl parametric

The swirl airflow generated by the parametric swirl device is expected to further increase the rate of fuel-air mixing in the cylinder compared to normal airflow. Due to the potential effect of the ram air intake by the significant change of the draft angle and length, a greater amount of oxygen-rich air is provided during each intake stroke. The ram air intake increases the volume and pressure of the inducted air charge, leading to a higher air density. The increased air density means there is a greater mass of oxygen per unit volume of air which provides the engine with a larger quantity of air during intake stroke. By the increase of the oxygen content in the cylinder, more complete combustion can be produced. This results in better combustion efficiency and power output as well as the improvement of the overall engine performance.

This effect can be observed from the comparison graph for the brake power and torque as shown in Figures 5.31 and 5.32, where all the swirl-parametric air intake devices perform noticeably better than the normal intake. At 3600 rpm, the highest maximum brake power is recorded by the Helical-P 2 at 3.73 kW with a 5.8% increase compared to the normal intake. The largest increase in brake power provided by this device is 17.1% at 2000 rpm. It is worth highlighting that, all six of the swirl-parametric devices tested experimentally showed a greater performance compared to normal intake of at least 1.5% to 18.4% at various engine speeds.



Figure 5.31: Comparison of engine performance - Brake Power for Parametric-Swirl Air Intake Against Speed

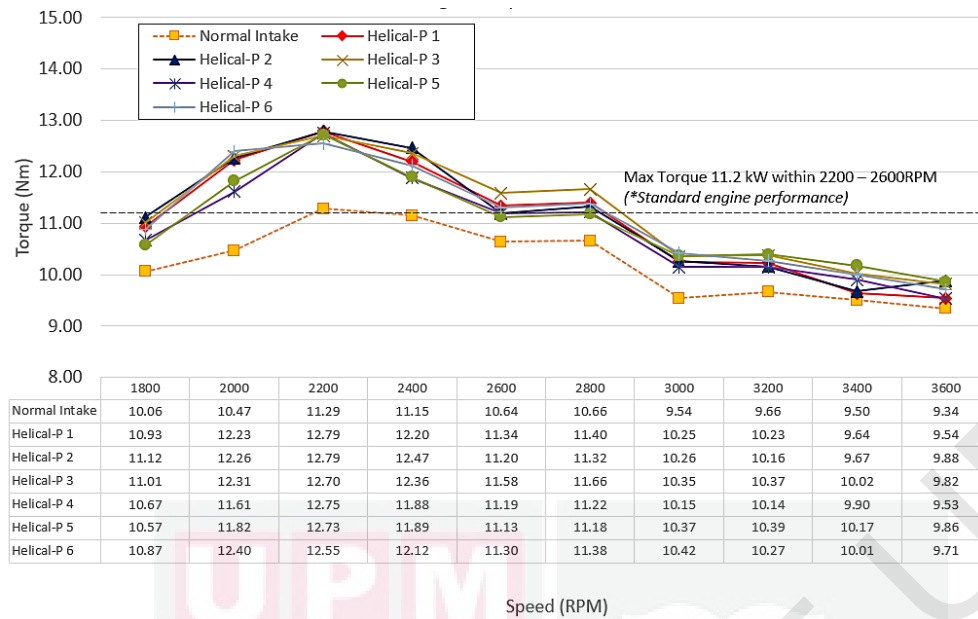


Figure 5.32: Comparison of engine performance - Brake Torque for Parametric-Swirl Air Intake Against Speed

In terms of the performance among the parametric combinations, Helical-P 2 and Helical-P 5 result in the highest brake power and torque compared to their respective helical revolutions. The parametric combination for these two devices is 70mm length and 18° draft angle. Helical-P 3 and Helical-P 6 with the combination of 120 mm length and 18° draft angle perform the second highest performance, then the least performing devices are the Helical-P 1 and Helical-P 4 with the combination of 70 mm length and 1.26° draft angle. From this comparison, it can be concluded that the ram swirled provided by the combination of the larger draft angle along with the swirling airflow has significantly contributed to the increase of brake power and torque. Since the performance of the least performing devices (Helical-P 1 and Helical-P 4) are better compared to normal intake, this further indicates the effectiveness of the ram-swirled devices in increasing the performance of a CI engine.

Overall, the parametric-swirl air intake results are in agreement with the CFD findings that have been thoroughly discussed in Chapter 4, where the increase of the outlet velocity due to the increasing draft angle causes a relative bulk motion of air intake charge, causing the 'ram' effect to dominate the flow field. The actual result provides evidence of the engine benefiting from the increase of both fuel-air mixing rate (due to air swirl) and mass flow rate (due to ram pressure).

In the previous base-swirl air intake analysis, the effective airflow in the intake manifold is very much affected by the increasing air resistance caused by the inner blades. The more the blade revolves, the more air flows to be blocked by the blade direction which is almost normal for the flow direction. More blade revolution causes a higher air resistance which reduces the airflow efficiency. This effect becomes more significant at higher engine speeds. A similar trend is observed in the parametric-swirl air intake regardless of the length and draft angle, where the device with 0.25 revolution performs slightly better than 0.5 revolution in terms of brake power and torque at higher engine speed.

This can be seen in the comparison graph shown in Figures 5.33 and 5.34, where there is a slight decreasing trend in the brake power and torque at 3600 rpm with the increase of the blade revolutions. For example, the 0.5 revolution which consists of Helical-P 4, P 5 and P 6 results in approximately 0.1%, 0.2% and 1.1% slightly lower performance than that of the 0.25 revolution with their respective combination of Helical-P 1, P 2 and P 3. No clear trend can be concluded at lower speeds which is similar to the condition in the base-swirl intake analysis.

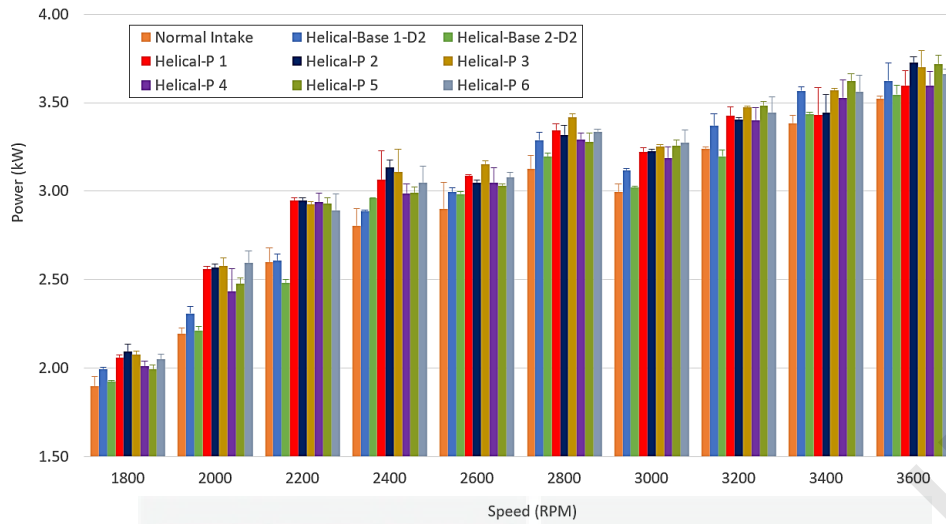


Figure 5.33: Comparison of engine performance - Brake Power for Base and Parametric-Swirl Air Intake Against Speed

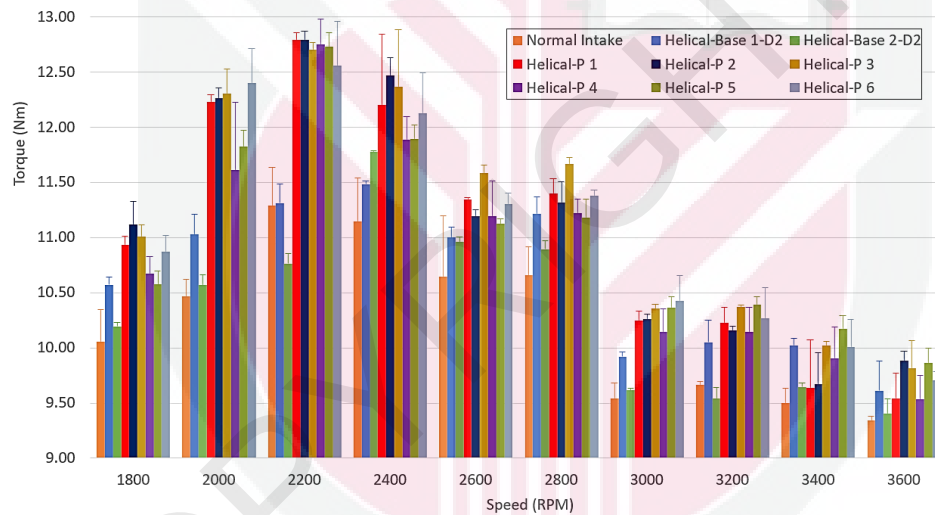


Figure 5.34: Comparison of engine performance - Brake Torque for Base and Parametric-Swirl Air Intake Against Speed

In terms of brake specific fuel consumption (BSFC) and engine efficiency (BTE), Figures 5.35 and 5.36 show the BSFC and BTE trendlines for the engine operating with the parametric-swirl devices which are generally better than that of the normal intake. By category, a device with 0.25 revolution (Helical-P 1, Helical-P 2 and Helical-P 3) has the lowest BSFC and highest

BTE, followed by the 0.5 revolution (Helical-P 4, Helical-P 5, and Helical-P 6) and the normal intake as shown in Figure 5.37 and 5.38.

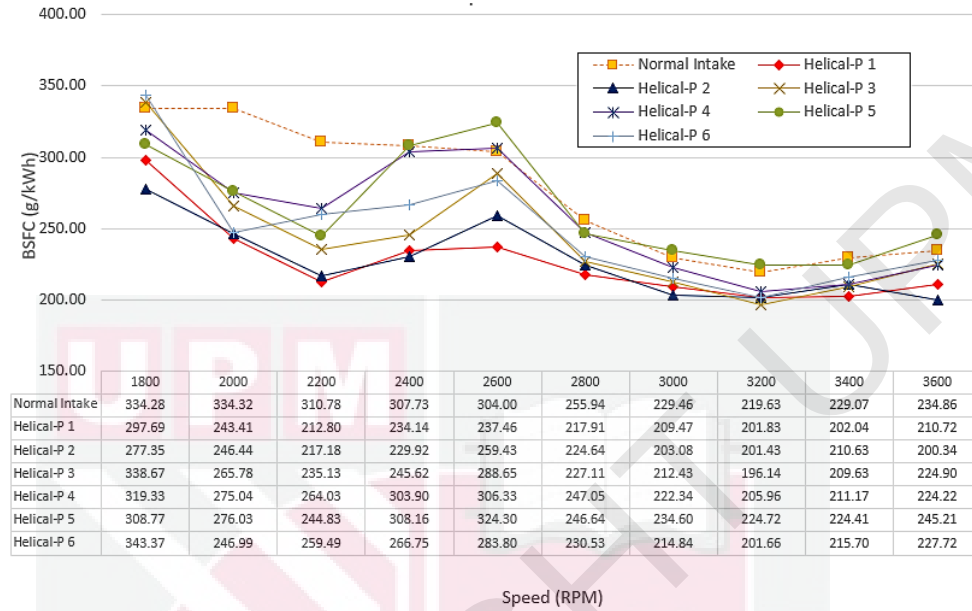


Figure 5.35: Comparison of engine performance - BSFC for Parametric-Swirl Air Intake Against Speed

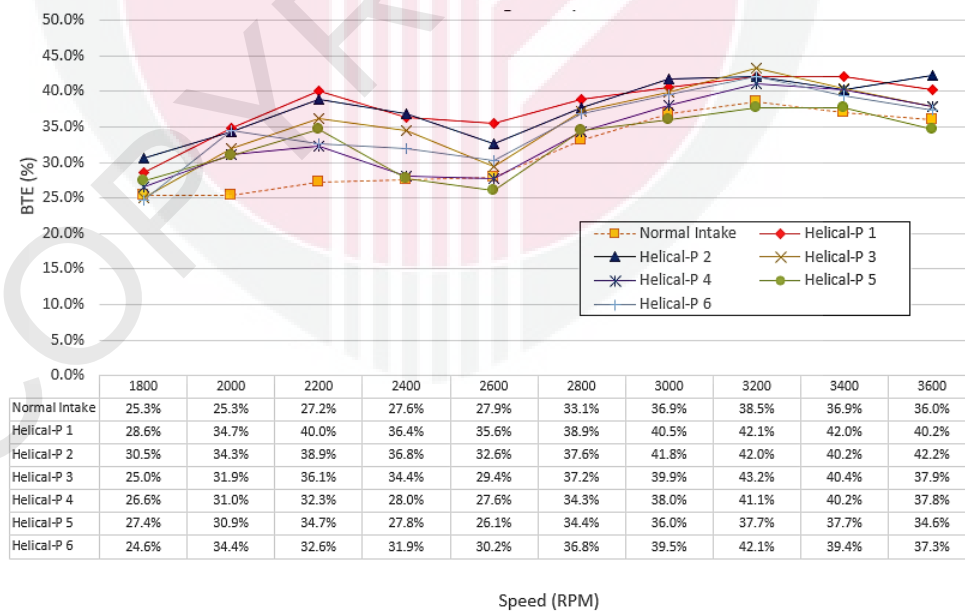


Figure 5.36: Comparison of engine performance - BTE for Parametric-Swirl Air Intake Against Speed

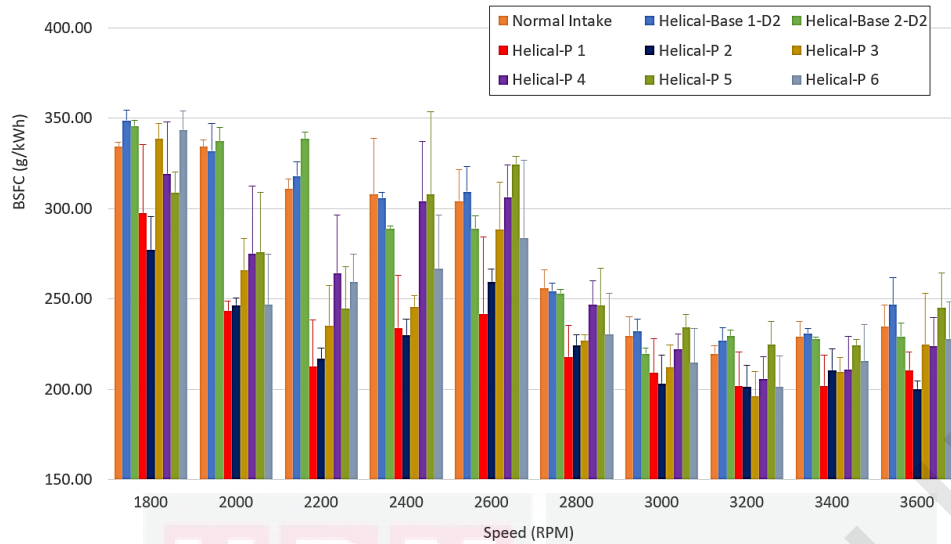


Figure 5.37: Comparison of engine performance - BSFC for Base and Parametric-Swirl Air Intake Against Speed

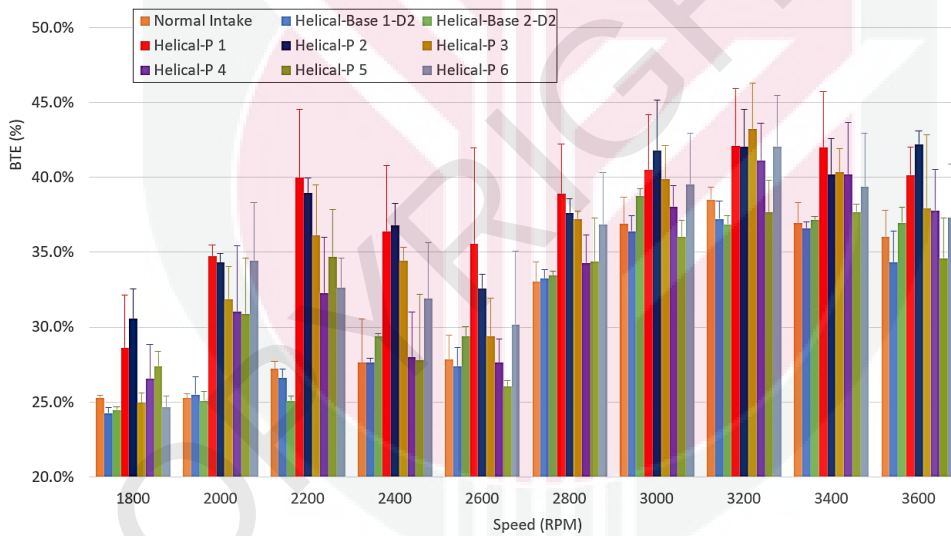


Figure 5.38: Comparison of engine performance - BTE for Base and Parametric-Swirl Air Intake Against Speed

At higher load conditions, it is observed that the highest improvement in BSFC and BTE compared to normal intake is given by Helical-P 1 and Helical-P 2 with approximately 30 ~ 32% BSFC reduction and 43 ~ 47% BTE improvement. In addition to the BSFC and BTE improvement, Helical-P 1 and Helical-P 2 also perform better in the brake power with the highest

improvement by 16.8 ~ 17.2% increase. At low load conditions, these two devices are still able to improve the BSFC and BTE by approximately 10 ~ 15% BSFC reduction and 11 ~ 17% BTE improvement, while the brake power has also improved by 2.1 ~ 5.8%.

Overall, the effect from the ram-swirled air intake device provides a significant improvement on both BSFC and BTE regardless of their parametric combination. The results provide evidence that the ram-swirled air intake is mostly able to improve the BSFC and BTE by the increased mass flow of air, higher air density and better fuel-air mixing rate. A greater air mass flow causes more efficient combustion, while the increase of air density due to the ram pressure promotes more complete combustion of the injected fuel. More complete combustion led to a higher brake power and torque which further enhanced the BTE of the engine. The use of the ram-swirled air intake may improve the fuel-air mixing with the fuel evenly distributed where the proper fuel-air mixing may facilitate better combustion and optimum use of injected fuel. This may result in reduced BSFC and improved BTE.

BMEP is also calculated to compare the performance of the parametric-swirl air intake with the typical range of diesel engines. As observed from Figures 5.39 and 5.40, the BMEP of the tested engine is between 5.47 to 7.34 bar which is still within the previous range and comparable to the recent typical range for passenger cars. The BMEP produced by the tested engine using parametric-swirl air intake is expected to be slightly higher than the base-swirl air intake since the engine receives more inducted air charged by the ram effect. At high load conditions, the tested engine operated with parametric-swirl

devices shows an increase in the maximum BMEP at approximately 13.3% compared to normal intake, while at low load conditions, the maximum BMEP has increased by 5.8%. Helical-P 2 shows the most consistent parametric combination that can provide among the highest BMEP at both high and low load conditions with approximately 7.33 and 5.67 bar respectively.

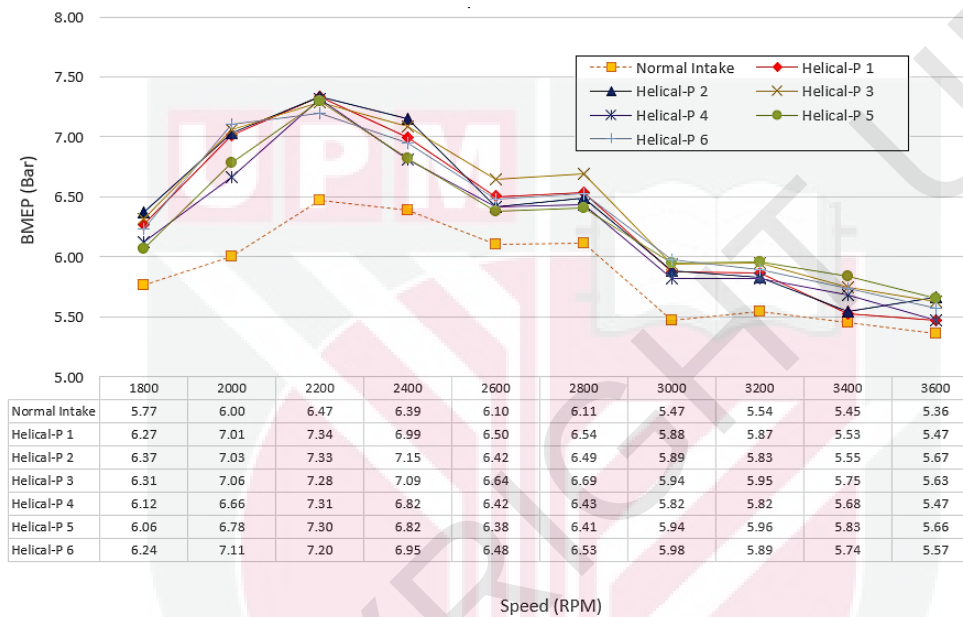


Figure 5.39: Comparison of engine performance - BMEP for Parametric-Swirl Air Intake Against Speed

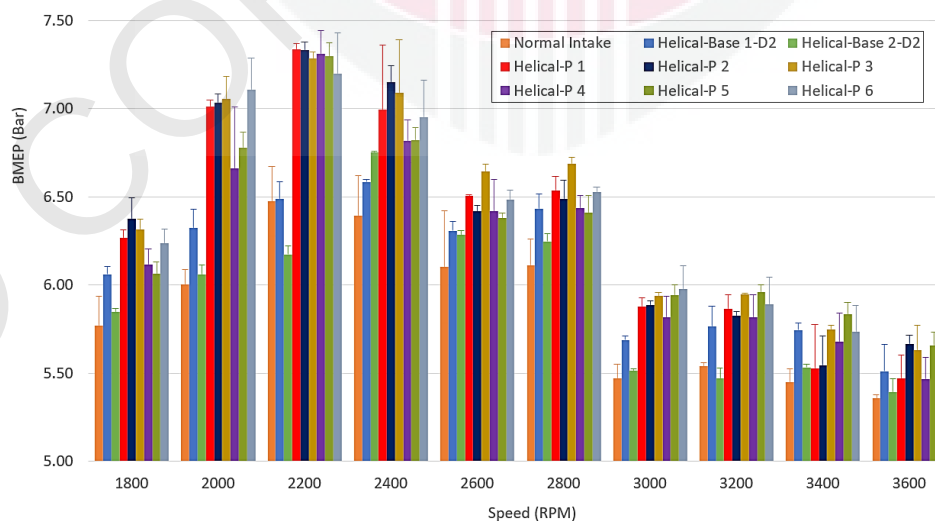


Figure 5.40: Comparison of engine performance - BMEP for Base and Parametric-Swirl Air Intake Against Speed

In most cases, all the results for parametric-swirl air intake are still greater than that of the highest BMEP recorded by the previous base-swirl air intake device, indicating the effectiveness of the ram-swirled air intake in the CI engine operation. As the BMEP measures the engine's ability to obtain or provide high airflow and use it effectively over the full operating range, the BMEP can provide further information on the engine air intake system's ability to provide high airflow and its effective use in the engine.

5.5.2 In-Cylinder Pressure. Pressure Rise Rate and Heat Release Rate Profiles

In general, the in-cylinder pressure, pressure rise rate and heat release rate of the parametric-swirl air behave similarly to the previous base-swirl air intake with two major stages of combustion observed from the profiles as presented in Figure 5.41, 5.42 and 5.43. These two stages are known as rapid or premixed combustion and mixing-controlled combustion. The only difference between these two intake operations is the maximum pressure which causes a slight variation in the ignition delay and start of combustion.

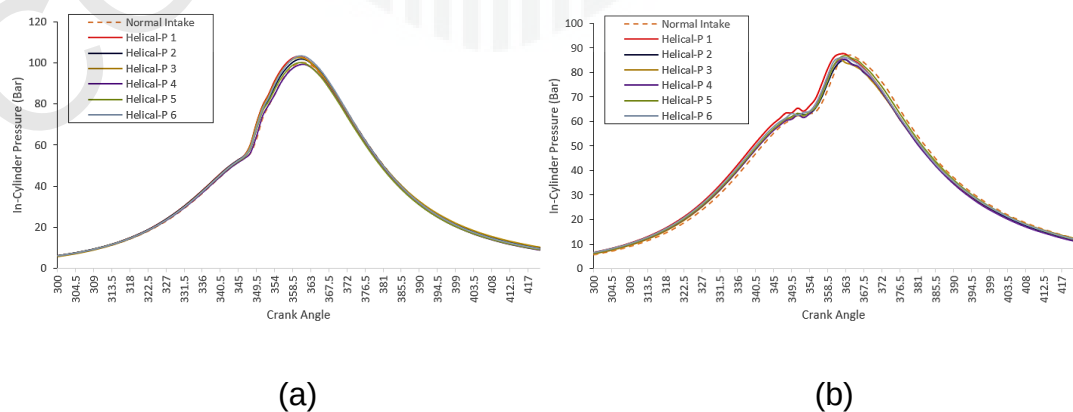


Figure 5.41: In-Cylinder Pressure Profile for Parametric-Swirl Air Intake Against Crank Angle. (a) 1800 rpm – High load, (b) 3600 rpm – Low load

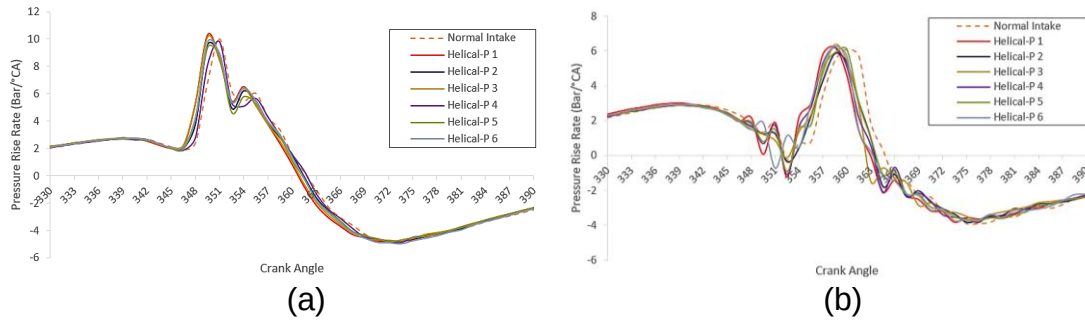


Figure 5.42: Pressure Rise Rate Profile for Parametric-Swirl Air Intake Against Crank Angle. a) 1800 rpm – High load, (b) 3600 rpm – Low load

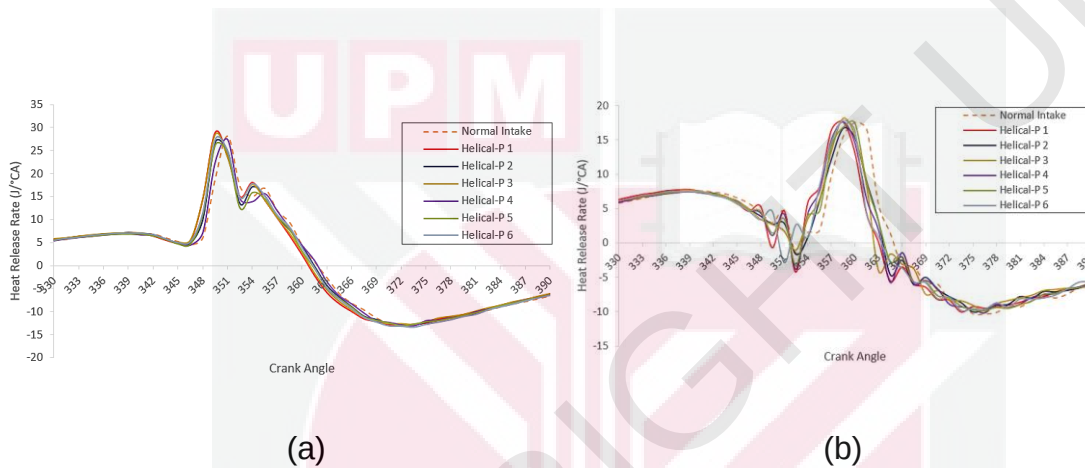


Figure 5.43: Heat Release Rate Profile for Parametric-Swirl Air Intake Against Crank Angle. (a) 1800 rpm – High load, (b) 3600 rpm – Low load

In Figure 5.41 (b) and 5.43 (b), it is observed that there is a pronounced pressure oscillation before and after the peak in-cylinder pressure and HRR at low load conditions (3600 rpm). This trend is similar to the base-swirl air intake and also normal air intake in the previous analysis. The obvious pressure oscillations at low and high load conditions are mainly due to the differences in the ignition delay and level of energy released.

At low load conditions, a longer ignition delay causes a delay in the start of combustion, resulting in a less controlled and more erratic combustion process. The variations in the combustion timing can contribute to pressure oscillations. On the other hand, there is a lower fuel demand at low load conditions which causes lower energy release. This results in a less stable flame front, making the combustion process more susceptible to pressure variations.

The full range of in-cylinder pressure results are presented in Figure 5.44 and 5.45, where the tested engine operated with the parametric-swirl devices exhibits slightly higher maximum in-cylinder pressure or at least on par with the normal intake in most cases. Generally, a higher maximum in-cylinder pressure in the swirl air intake device may suggest that this air intake system has better overall engine performance in terms of power, efficiency and torque compared to normal intake.

The in-cylinder pressure trendlines are also close to each other and no abnormality was found without any extremely lower or higher in-cylinder pressure, showing the normal operation of the engine. The trendline shown in Figure 5.46 is also in agreement with the typical in-cylinder pressure level at high and low load conditions, where the maximum in-cylinder pressure is normally greater when the engine load is higher.

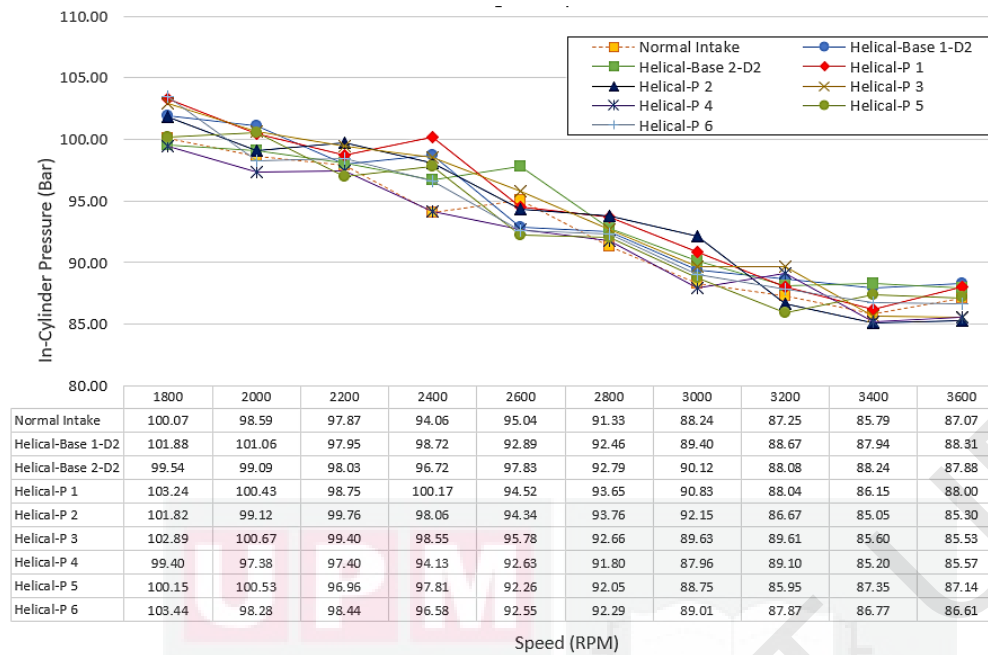


Figure 5.44: Comparison of engine performance – Maximum In-Cylinder Pressure for Parametric-Swirl Air Intake Against Speed

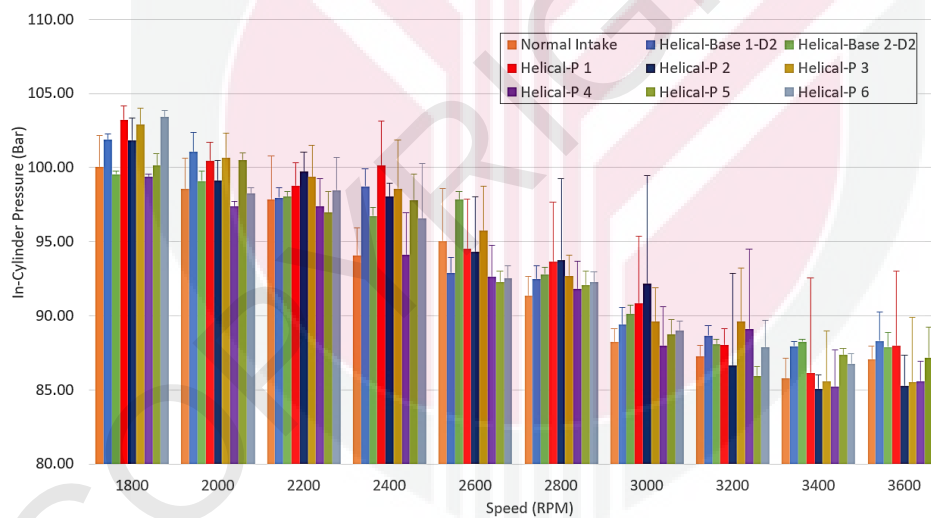


Figure 5.45: Comparison of engine performance – Maximum In-Cylinder Pressure for Base and Parametric-Swirl Air Intake Against Speed

From Figure 5.46 to 5.49, the comparison of maximum PRR and HRR shows the engine with parametric-swirl devices generally has slightly better combustion or at least on par with the normal intake in most cases. This is consistent with the maximum in-cylinder pressure result discussed earlier. The

trendlines for both HRR and PRR behave similarly to the base-swirl air intake, where the higher maximum PRR and HRR may indicate more rapid pressure rise and energy released during the combustion process, this can be due to more intense and rapid combustion.



Figure 5.46: Comparison of engine performance – Pressure Rise Rate for Parametric-Swirl Air Intake Against Speed

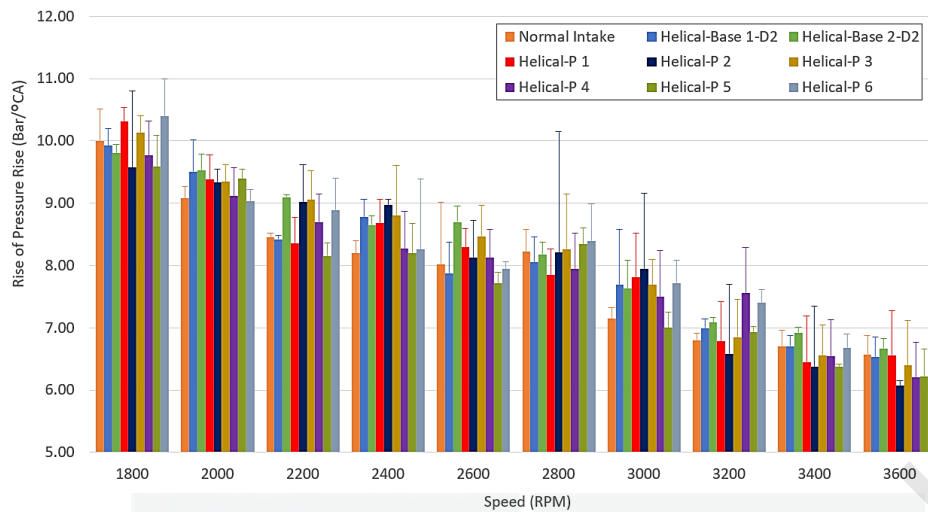


Figure 5.47: Comparison of engine performance – Pressure Rise Rate for Base and Parametric-Swirl Air Intake Against Speed

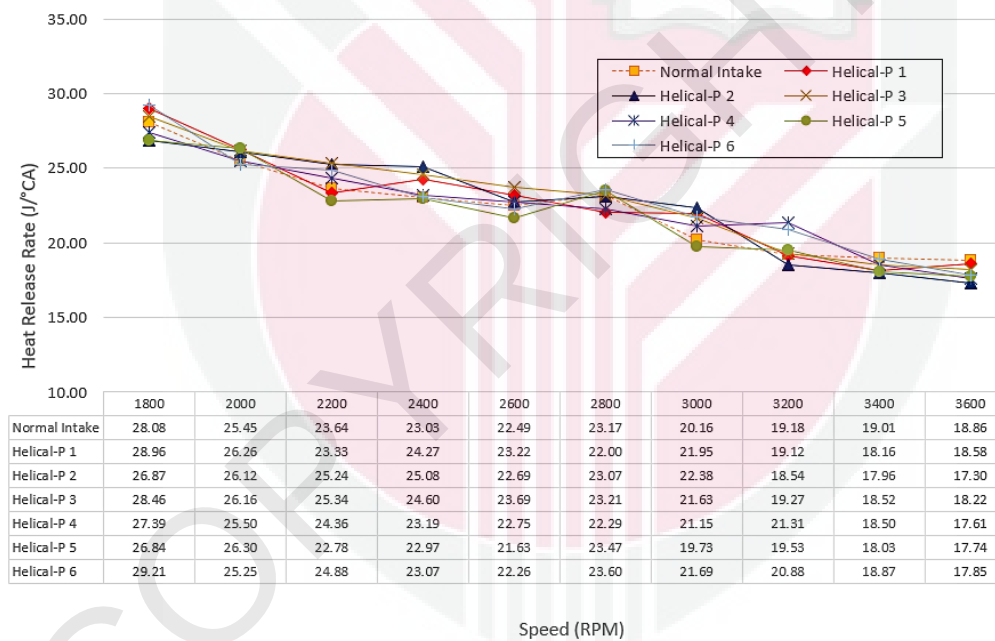


Figure 5.48: Comparison of engine performance – Heat Release Rate for Parametric-Swirl Air Intake Against Speed

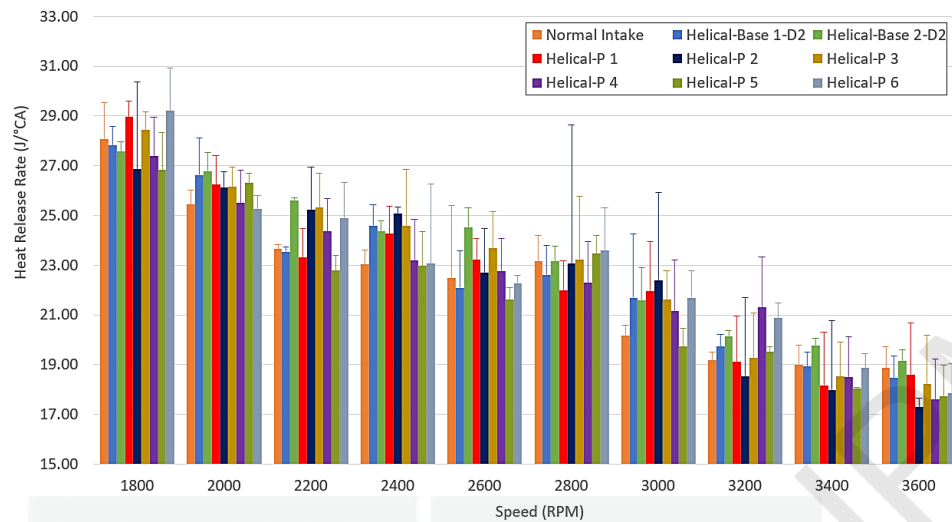


Figure 5.49: Comparison of engine performance – Heat Release Rate for Base and Parametric-Swirl Air Intake Against Speed

5.5.3 Emissions Results

In this section, five common exhaust emissions (CO, CO₂, O₂, NO_x and UHC) collected from the verification of the parametric-swirl engine performance will be further discussed. All these exhaust emissions have identical trends with the normal air intake at high and low load conditions. This indicates the likelihood of the parametric-swirl device to perform at the typical range of the CI engine combustion. Nevertheless, the ram-swirled air intake has shown a substantial impact on the engine performance and several combustion behaviours as discussed in the previous chapter. The significant change in the intake air-charged characteristics may have an additional effect to the production of the exhaust emissions.

In the previous analysis of the CO emissions from the impact of the air-fuel ratio, a slight change had no significant effect on the CO emissions level, instead, the homogeneity of the fuel-air mixture dominated the CO formation.

The parametric combination causes the homogeneity of the fuel-air mixture to be more dominant due to the effect of the inducted ram-swirled air. This effect can be seen in Figure 5.50 which shows a further reduction in the CO emissions by up to 71.2%, where the highest reduction is mainly at the part load conditions. This is even lower than the base-swirl devices at approximately 36.4% difference. The significant reduction is mainly due to a more homogeneous fuel-air mixture that ensures better combustion efficiency, reducing the formation of CO.

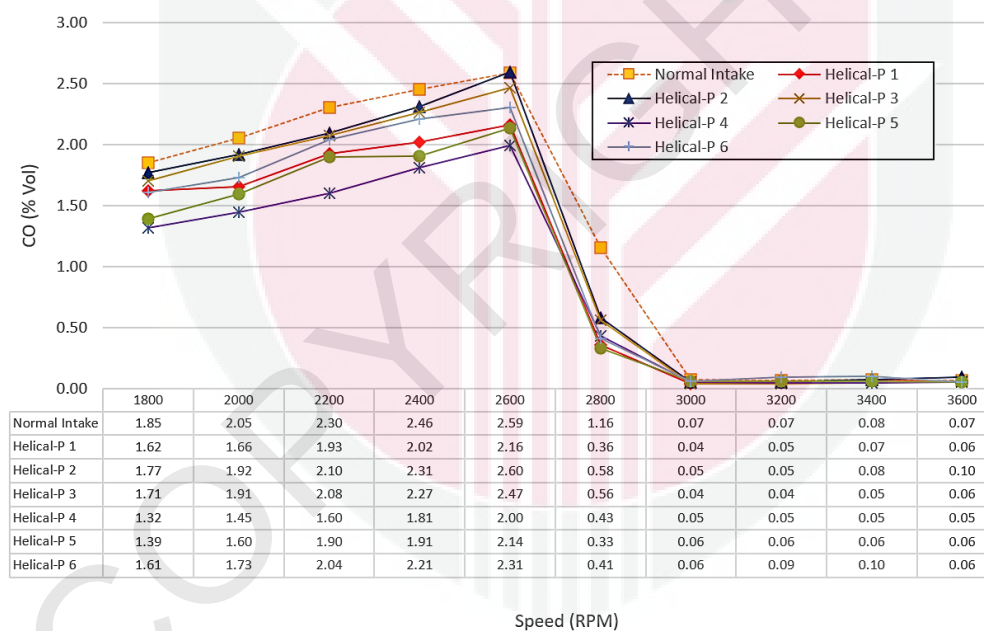


Figure 5.50: Comparison of Engine Emissions – CO for Parametric-Swirl Air Intake Against Speed

For carbon dioxides (CO₂) emissions, there is a trend of increasing CO₂ levels in relation to the homogeneity of the fuel-air mixture due to the parametric-swirl device at similar load conditions. In this case, higher CO₂ emissions are achieved due to more complete combustion resulting from the homogeneous

fuel-air mixture. As a primary combustion product, higher CO₂ should be typically produced in case of more complete combustion.

In Figure 5.51, the trend of CO₂ emissions for the swirl and normal intake is in agreement with the air-fuel ratio and the trend of CO in most cases. A lower air-fuel ratio than the normal intake tends to produce higher CO₂ emissions levels due to more complete combustion. The parametric-swirl air intake has a slightly lower air-fuel ratio compared to normal intake, where the reduction of the excess air leads to more complete combustion, thus causing higher CO₂ emissions levels.

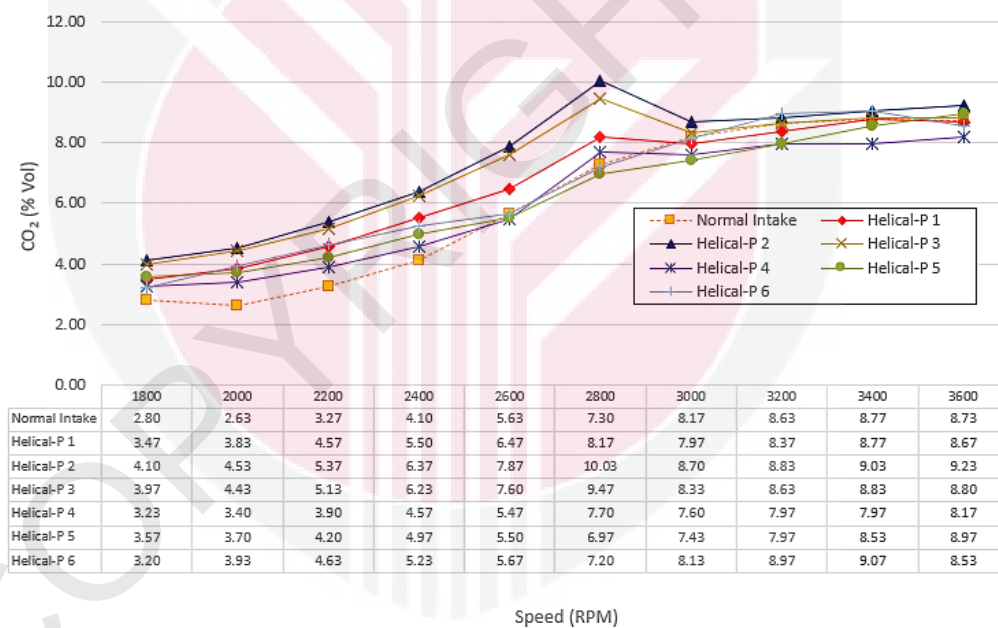


Figure 5.51: Comparison of Engine Emissions – CO₂ for Parametric-Swirl Air Intake Against Speed

The highest CO₂ emissions level is represented by Helical-P 2 which is generally 72.2% higher compared to normal intake. The effect of the air-fuel ratio can be seen at high and low load conditions. A higher CO₂ level is observed at low load conditions compared to high load conditions. This

indicates that this engine is having more complete combustion at low load conditions.

Oxygen (O_2) level is also observed where the O_2 is largely affected by the ram-swirled air intake as can be seen in Figure 5.52. In general, all the parametric combinations result in lower O_2 levels compared to normal intake mostly at the part load to a higher load condition, indicating the effectiveness of the parametric-swirl device to promote the homogeneous mixture for better combustion.

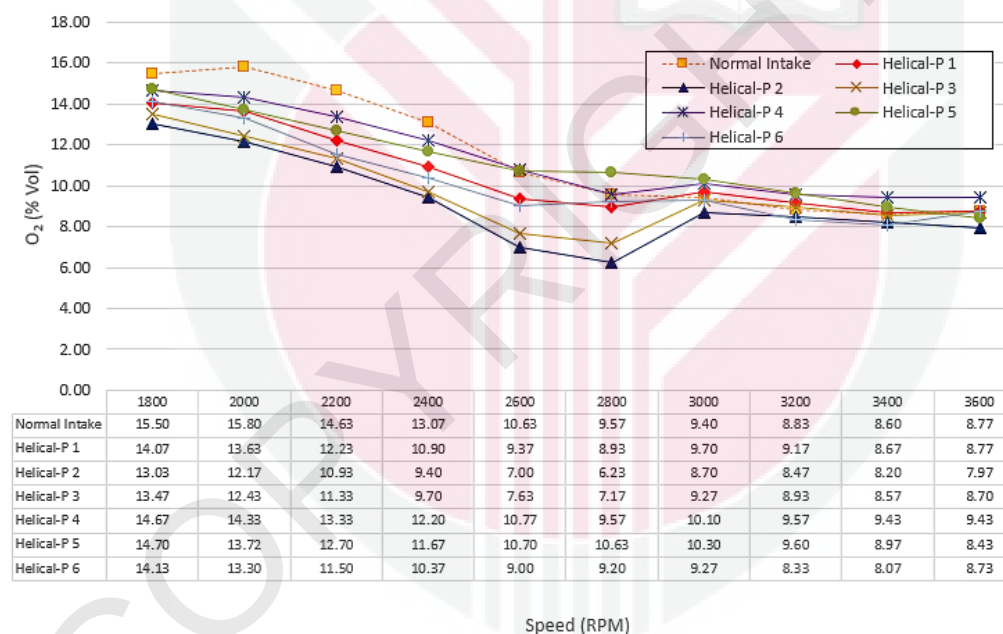


Figure 5.52: Comparison of Engine Emissions – O_2 for Parametric-Swirl Air Intake Against Speed

The highest reduction is given by the Helical-P 2 with approximately 35% lower compared to normal intake and this is even lower than that of the base-swirl air intake at a 20% difference. As a higher fuel-air homogeneity leads to a

higher oxygen consumption during combustion, there is less air residual in the exhaust gases, resulting in a lower O₂ level.

As previously discussed for the base-swirl air intake analysis, the NO_x produced is not totally dependent on the combustion temperature as it is more likely to be dominated by the fuel-air mixture homogeneity in the engine where the inducted air intake characteristics are significantly improved. In Figure 5.53, all the parametric combinations recorded steadily low NO_x levels with the lowest being 47.7% lower compared to normal intake. The results are comparable with the trend of the NO_x emission produced by the base-swirl devices.

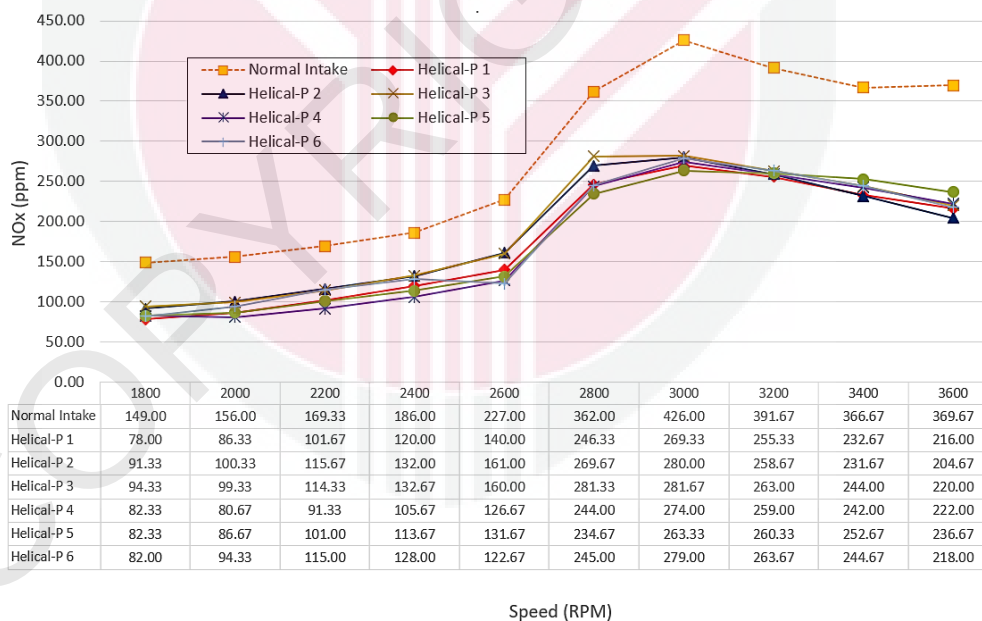


Figure 5.53: Comparison of Engine Emissions – NO_x for Parametric-Swirl Air Intake Against Speed

This is consistent with the design capability to provide sufficient swirl airflow with minimum air resistance. The effect of ram in the parametric combinations

further contributes to the overall lower NO_x level for both 0.25 and 0.5 revolutions. As previously discussed, the ram-swirled air intake may contribute to a better homogeneity of the fuel-air mixture as well as increase the fuel-air mixing rate, thus reducing the formation of NO_x.

As shown in Figures 5.54, Unburned Hydrocarbons (UHC) in the parametric-swirl air intake show a significant reduction at part load to higher load conditions, where the lowest UHC emissions are recorded at 40.7% lower compared to normal intake. This is even lower than that of the base-swirl air intake at a 6.1% difference. Following the previous findings on the effect of the ram-swirled air intake, the parametric combinations are expected to promote better fuel-air mixing which causes better combustion, thus leaving fewer unburned fuels in the exhaust gases. Figure 5.55 shows the summary of the exhaust emissions level at high and low load conditions.



Figure 5.54: Comparison of Engine Emissions – UHC for Parametric-Swirl Air Intake Against Speed

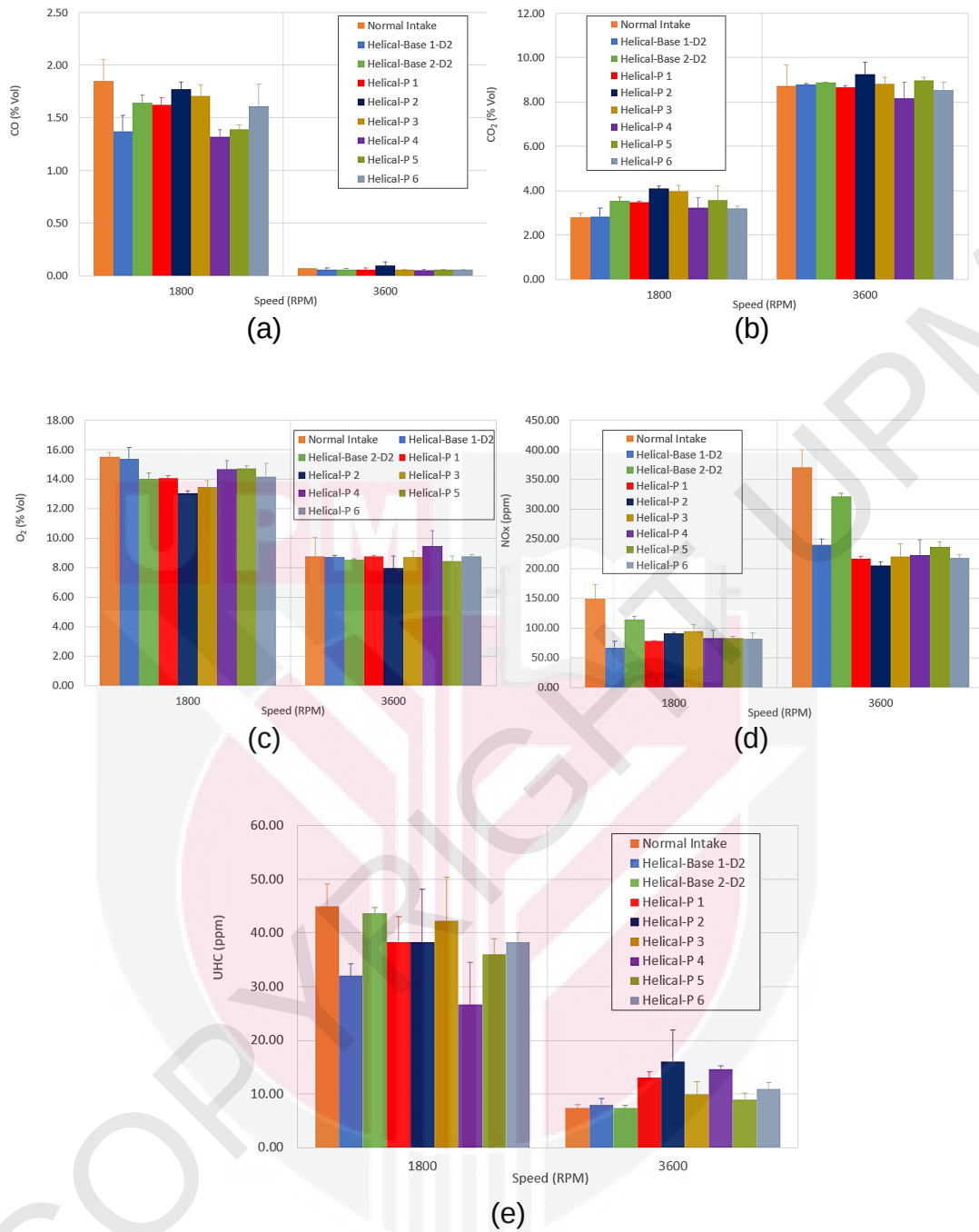


Figure 5.55: Comparison of Engine Emissions for Base and Parametric-Swirl Air Intake Against 1800 and 3600 rpm, (a) CO, (b) CO₂, (c) O₂, (d) NO_x and (e) UHC

5.6 Engine Vibration Characteristics

Due to the nature of the combustion process, an engine is well-known for its vibration characteristics, where the mounting of the engine on a vehicle or specific equipment plays a crucial role in isolating the vibrations. The vibration of a diesel engine comes from various sources. It includes but is not limited to the combustion-related vibrations (e.g. rapid combustion process), the reciprocating motion of the reciprocating components (e.g. crankshaft, pistons, connecting rod), speed of the engine, harmonics and resonances due to the interaction with various engine components, components wear, misalignment and unbalanced rotating parts. Moreover, the modified engine vibration can have a different characteristic compared to the original structure of engine components.

For this engine, there are several modifications made to suit the instruments for data collection such as a modification on the exhaust pipe to be equipped with the oxygen sensor. It is important to highlight that the vibration data of the normal intake condition for this engine includes the additional instruments and modified components. It will not represent the vibration of the original version of the engine. Thus, the baseline vibration with all the existing setups is initially measured for a more accurate comparison with the swirl devices. This section intends to show and quantify the overall vibration level between the tested engine while the engine is mounted on the dynamometer bench. To fully understand the vibration characteristics of an engine, detailed testing with several techniques is required. In view of the modified engine conditions and

limitation to the data of each vibrating source, the Root Means Square (RMS) value is used as an indicator to quantify and analyse the produced vibration.

5.6.1 Root Mean Square (Average RMS) Vibration Level

The RMS represents the statistical measurement of the vibration's magnitude over a period of time, where it is used to measure the amplitude of the vibrations produced by the reciprocating motion of engine components caused by the rapid combustion of fuel with the available air in the combustion chamber. The RMS value is calculated by the square root of the average of the squared instantaneous vibration values over a specific period. RMS provides a measure of the overall vibration amplitude, indicating the severity and intensity of the vibrations. This indicator is also useful to assess the engine's performance, health and any other potential issues provided that the engine vibration under normal operating conditions has been properly measured.

Figure 5.56 shows a comparison of engine vibration for base-swirl devices against speed. The vibration levels for all the swirl devices are noticeably lower than that of the normal intake with the lowest RMS level achieved by the 0.25 revolution at most of the engine speeds. At 0.25 revolution produces up to 41.1% lower RMS level compared to normal intake, while 0.5 and 0.75 produce up to 32.7% and 28.7% lower RMS levels respectively. In normal operating conditions, higher engine speeds generally result in higher-frequency vibrations. It is observed that the intensity of the vibration slightly increases at higher engine speeds.

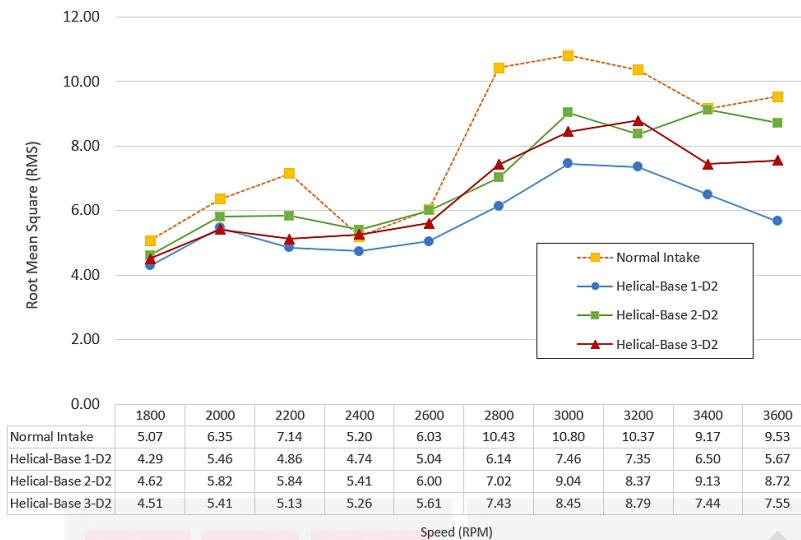


Figure 5.56: Comparison of engine vibration – Root Mean Square (RMS) for Base-Swirl Air Intake Against Speed

Figure 5.57 shows a comparison of engine vibration for parametric-swirl devices against speed. Except for 3000 rpm, the vibration levels for all the swirl devices are apparently higher than that of the normal intake with the lowest RMS level among the other parametric devices achieved by the Helical-P 2. Helical-P 2 produces up to 70% higher RMS level, while the highest RMS level with over 100% increase compared to normal intake are Helical-P 4, Helical-P 5 and Helical-P 6.

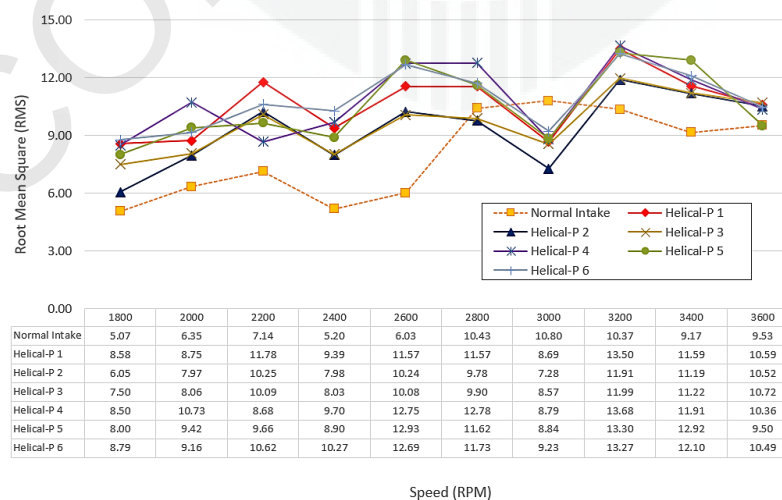


Figure 5.57: Comparison of engine vibration – Root Mean Square (RMS) for Parametric-Swirl Air Intake Against Speed

This substantial increase in the vibration level for all the parametric-swirl devices is more likely due to the significant increase in the engine performance by the effect of ram swirled air intake device. As noticed by the increase of the maximum in-cylinder pressure for most cases in the previous section (5.5.2 In-Cylinder Pressure, PRR and HRR profiles), the higher pressure and forces within the engine cylinders may lead to a higher vibration due to the combustion process compared to normal intake.

Other than the combustion-related vibrations, the parametric-swirl device can also contribute to the additional structural vibrations of the engine. The device with an irregular shape, thin walls and relatively larger size of either 120 mm length and 18° draft angle may lead to the harmonics patterns and resonance, resulting in amplified vibrations. This is different from the base-swirl air intake device which is constructed with a more rigid body and thick walls.

5.6.2 Time and Frequency Domain

Table 5.10 and 5.11 summarize the vibration mid-level sample of time and frequency domains captured by the tri-axial accelerometer sensor at 3600 and 1800 rpm respectively. At 3600 rpm, the vibration is measured at approximately 34% to 38% engine load, while for 1800 rpm at approximately 90 to 100% engine load. Since the base and parametric-swirl device of 0.25 blade revolution designs perform better in the experiment stage in terms of brake power and torque compared to 0.5 and 0.75 revolutions, thus only these designs are taken as a representative sample for time and frequency domains.

Table 5.10: Mid-Level Sample of Time and Frequency Domain at 3600 rpm

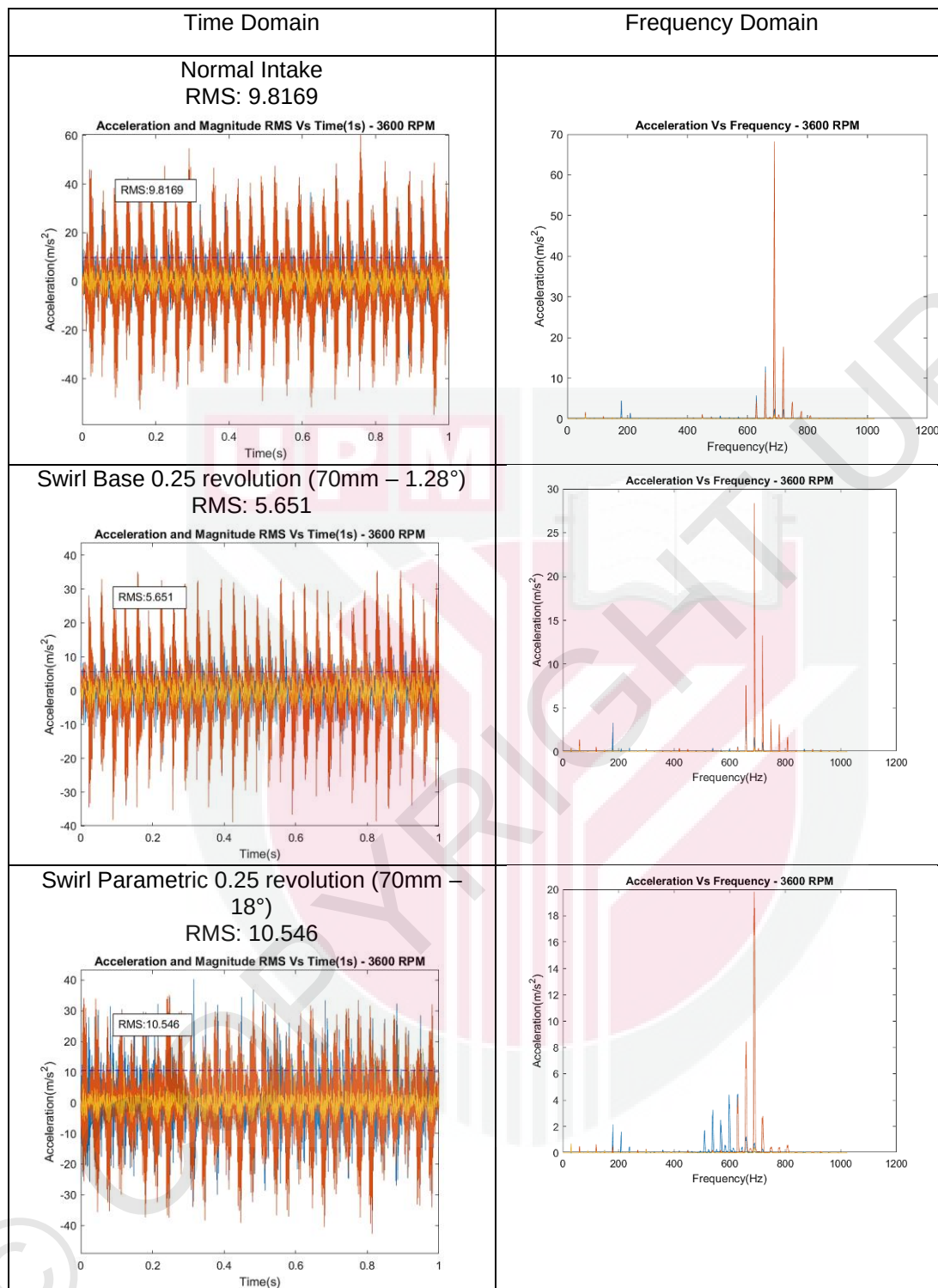
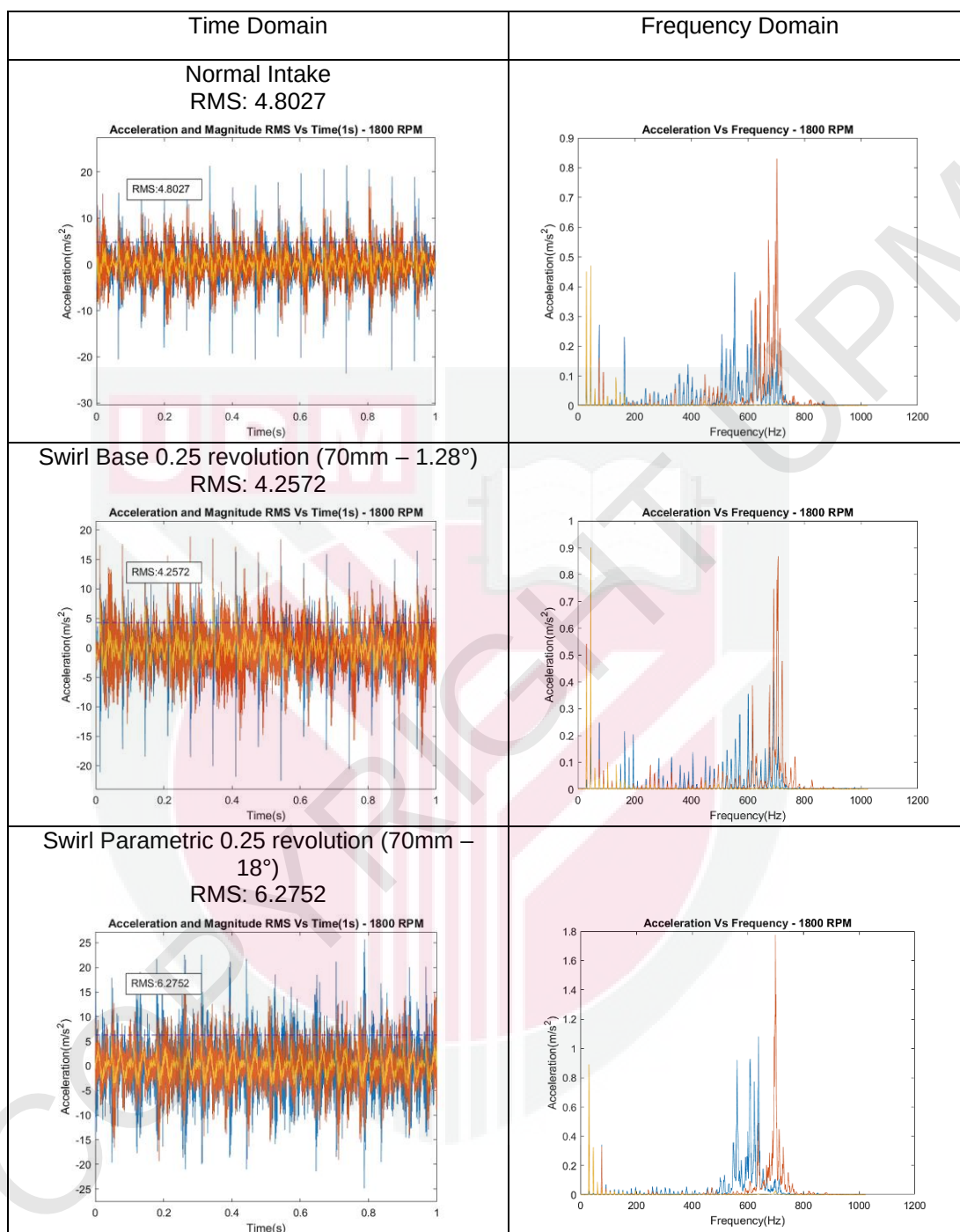


Table 5.11: Sample of Time and Frequency Domain at 1800 rpm (Mid-Level)



Based on the presented RMS magnitude and acceleration data results, it can be concluded that the engine vibration level is fairly normal for all tested swirl air intake devices compared to the baseline engine of normal intake. No obvious trend that could be associated with engine knocking at full load

conditions. This can be further viewed from the behaviour of the in-cylinder pressure profiles presented in sections 5.3.2 and 5.5.2, where there are no erratic and excessive oscillations comparable to the knocking behaviour presented by Moshrefi & Shoaee (2017) in Figure 2.15.

In view of the trend in the base-swirl air intake device, there is a clear possibility of engine vibration level reduction that can be associated with the usage of this device. Even with an additional vibration source that can be possibly generated from the assembly of this device, the reduction is quite significant within 28.7% to 41.1% compared to normal intake. Base-swirl air intake design focuses on effective swirl formation and minimum flow restriction without a significant ram pressure. The improvement to the combustion process by the benefit of these characteristics may further reduce the engine vibration due to the refinement of the diesel detonation in the combustion chamber.

5.7 Estimated Measurement Uncertainty

By employing the technique and formula for measurement uncertainty in the experimentation as discussed in Chapter 3 (Section 3.4.7, Estimation of Measurement Uncertainty), Table 5.12 summarizes the estimated measurement uncertainty based on a dataset of experimental variables involved in this study. Since the physical quantities in every measurement are subject to uncertainty, the actual measurements will never be made under perfect conditions. Therefore, all the measurement results presented in this thesis shall be accompanied by the listed uncertainty to make a more reliable

decision. The related expressions used to estimate the selected uncertainties are discussed next.

Table 5.12: Estimated Measurement Uncertainty for Experimental Variables

Variable	Uncertainty
Brake Torque (Nm)	± 0.215253312
Engine Speed (rpm)	± 24.05828052
Brake Power (kW)	± 0.03422926
Time taken for SFC (s)	± 0.115470054
Air Intake Temperature ($^{\circ}\text{C}$)	± 4.73618633
Exhaust Gas Temperature ($^{\circ}\text{C}$)	± 18.67482483
Air Velocity (m/s)	± 0.150701126
Fuel Mass Flow Rate (kg/h)	± 0.072424257
Engine Vibration (RMS)	± 3.348228885
Brake Specific Fuel Consumption (g/kWh)	± 36.06436292
Brake Thermal Efficiency (%)	± 5.200070643
BMEP (Bar)	± 0.138844633
Cylinder Pressure (Bar)	± 1.787137774
HRR ($\text{J}/^{\circ}\text{CA}$)	± 1.742514154
PRR ($\text{Bar}/^{\circ}\text{CA}$)	± 0.609998954
Air/Fuel Ratio	± 0.091791212
O_2	± 1.434659878
CO (%)	± 0.072724019
CO_2 (%)	± 1.101469499
HC (PPM)	± 5.9863795
NO_x (PPM)	± 49.10203162

It is important to mention that the subsequent testing data may fall outside the uncertainty margins depending on the magnitude of the improvement to the engine. As the engine parameter changes during the experiment, the results could be affected as well. The parameter changes in the subsequent test may be significant enough to surpass the uncertainty associated with the previous dataset. In addition, it is generally not suitable to include all subsequent data together for a single value of uncertainty margin as the measurement after improvement may have different sources and characteristics leading to incorrect representation of the overall uncertainty. Therefore, in reporting the

measurement uncertainty, each subsequent dataset should be considered for calculation. This includes the subsequent test that improves the performance and emissions. In estimating the measurement uncertainty as shown in Table 5.12, the following expression can be used for calculations:

- **Uncertainty in the Brake Power (kW), $u_e(BP) = \pm 0.0342$ kW**

$$u_c(BP) = BP \cdot \sqrt{\left(\frac{u(BP)}{BP}\right)^2 + \left(\frac{u(W)}{W}\right)^2 + \left(\frac{u(N)}{N}\right)^2} \quad (5.1)$$

Where,

u_c = Combined (relative) uncertainty

BP = Mean of the BP repeated measurements in kW (3.525 kW)

W = Load applied in N (49.13 N using 5kg calibrator)

N = Speed of engine in rpm (at 3600 rpm)

$$u_e(BP) = k \times u_c(BP) \quad (5.2)$$

Where,

u_e = Expanded uncertainty

k = Confidence factor (1.96 at 95% Confidence Level)

Using the standard uncertainty, u formula provided in Section 3.4.7, below estimated uncertainties are obtained at a 95% Confidence Level.

- Standard uncertainty of Brake Power readings, $u(BP) = \pm 0.0085$ kW
- Standard uncertainty of Load applied, $u(W) = \pm 0.131$ N
 - Assuming normal distribution; ± 0.2561 N /1.96
- Standard uncertainty of Speed, $u(N) = \pm 12.275$ rpm
 - Assuming normal distribution; ± 24.058 rpm /1.96

The standard uncertainties are calculated from both Type A and B evaluations that are obtained from the repeated readings and the information from the typical uncertainties, specifications, accuracies or errors provided by the manufacturer or from the certification. For Type B uncertainties, the standard uncertainty is calculated in the assumption of either normal, rectangular or triangular distribution. These standard uncertainties are further calculated using Equation (5.1) and (5.2). to obtain the combined and expanded uncertainty for Brake Power. Expanded uncertainty, $u_e(BP)$ is the final uncertainty for Brake Power. The same method is used for all the uncertainties.

- **Uncertainty in the Brake Torque (Nm), $u_e(BT) = \pm 0.215$ Nm**

$$u_c(BT) = BT \cdot \sqrt{\left(\frac{u(BT)}{BT}\right)^2 + \left(\frac{u(B)}{B}\right)^2 + \left(\frac{u(V)}{V}\right)^2} \quad (5.3)$$

Where,

BT = Mean of the BT repeated measurements in Nm (9.4 Nm)

B = Linearity of the weighing balance in g (5000.7 g)

N = Vernier caliper measurement in cm (20 cm)

$$u_e(BT) = k \times u_c(BT) \quad (5.4)$$

Where,

u_e = Expanded uncertainty

k = Confidence factor (1.96 at 95% Confidence Level)

Using the standard uncertainty u formula, the below-estimated uncertainties are obtained at a 95% Confidence Level. These uncertainties are further calculated using Equation (5.3) and (5.4) to obtain the combined and expanded uncertainty for Brake Torque. Expanded uncertainty, $u_e(BT)$ is the final uncertainty for Brake Torque.

- i. Standard uncertainty of BT readings, $u(BT) = \pm 0.110 \text{ Nm}$
 - ii. Standard uncertainty of weighing balance, $u(B) = \pm 0.115 \text{ g}$
 - Assuming rectangular distribution; $\pm 0.2 \text{ g} / \sqrt{3}$
 - iii. Standard uncertainty of vernier calliper, $u(V) = \pm 0.00051 \text{ cm}$
 - Assuming normal distribution; $\pm 0.001 \text{ cm} / 1.96$
- **Uncertainty in the Fuel Mass Flow Rate, $u_e(\dot{m}) = \pm 0.0724 \text{ g/kWh}$**

$$u_c(\dot{m}) = \dot{m} \cdot \sqrt{\left(\frac{u(\dot{m})}{\dot{m}}\right)^2 + \left(\frac{u(l)}{l}\right)^2 + \left(\frac{u(t)}{t}\right)^2} \quad (5.5)$$

Where,

$\dot{m}$ = Mean of the fuel mass flow rate repeated measurements in kg/h

(0.64616 kg/h)

l = Volume of the drained fuel (2 ml)

t = Time of the drained fuel (9.376 seconds at 3600 rpm)

$$u_e(\dot{m}) = k \times u_c(\dot{m}) \quad (5.6)$$

Where,

u_e = Expanded uncertainty

k = Confidence factor (1.96 at 95% Confidence Level)

Using the standard uncertainty u formula, the below-estimated uncertainties are obtained at a 95% Confidence Level. These uncertainties are further calculated using Equation (5.5) and (5.6) to obtain the combined and expanded uncertainty for Brake Torque. Expanded uncertainty, $u_e(\dot{m})$ is the final uncertainty for fuel mass flow rate.

- i. Standard uncertainty of fuel mass flow rate readings, $u(\dot{m}) = \pm 0.0161$ kg/h
- ii. Standard uncertainty of fuel drained volume, $u(l) = \pm 0.1$ ml
 - Assuming triangular distribution; $(\pm 0.1 \text{ ml Tol.})/\sqrt{6} \times 50$
 - Assuming triangular distribution with 27°C working temperature (Coefficient of expansion at 20°C $\text{EX} = 4 \times 10^{-6}$); $7 \times 4 \times 10^{-6} / 27\sqrt{6}$.
- iii. Standard uncertainty of fuel drained time, $u(t) = \pm 0.115$ seconds
 - Assuming rectangular distribution; ± 0.2 seconds $/\sqrt{3}$
- **Uncertainty in the Brake Specific Fuel Consumption, $u_e(\text{bsfc}) = \pm 36.064$ g/kWh**

$$u_c(\text{bsfc}) = \text{bsfc} \cdot \sqrt{\left(\frac{u(\text{bsfc})}{\text{bsfc}}\right)^2 + \left(\frac{u(\text{BP})}{\text{BP}}\right)^2 + \left(\frac{u(\dot{m})}{\dot{m}}\right)^2} \quad (5.7)$$

Where,

bsfc = Mean of the bsfc repeated measurements in g/kWh (9241.2 g/kWh)

BP = Brake Power readings (3.524 kW)

$\dot{m}$ = Mass flow rate of fuel in kg/h (0.646 kg/h)

$$u_e(bsfc) = k \times u_c(bsfc) \quad (5.8)$$

Where,

u_e = Expanded uncertainty

k = Confidence factor (1.96 at 95% Confidence Level)

Using the standard uncertainty u formula, the below-estimated uncertainties are obtained at a 95% Confidence Level. These uncertainties are further calculated using Equation (5.7) and (5.8) to obtain the combined and expanded uncertainty for Brake Torque. Expanded uncertainty, $u_e(bsfc)$ is the final uncertainty for Brake Torque.

- i. Standard uncertainty of BSFC readings, $u (bsfc) = \pm 12.119 \text{ g/kWh}$
- ii. Standard uncertainty of BP, $u (BP) = \pm 0.0175 \text{ kW}$
 - Assuming normal distribution; $\pm 0.0342 \text{ kW} / 1.96$
- iii. Standard uncertainty of mass flow rate of fuel, $u (\dot{m}) = \pm 0.0370 \text{ kg/h}$
 - Assuming normal distribution; $\pm 0.0724 \text{ kg/h} / 1.96$

5.8 Summary

In this chapter, the effect of base-swirl and parametric-swirl air intake is discussed in detail. It can be concluded from the detailed analysis that the formation of swirl airflow with the effect of ram caused by the blade revolution and parametric combinations led to higher engine performance, better combustion and reduction of exhaust emissions. This is mainly due to the increase in fuel-air mixture homogeneity and higher mixing rate. The improvement in the homogeneity and fuel-air mixing rate results in better combustion efficiency which slightly reduces the ignition delay, improves

combustion timing and produces more rapid energy release. This improvement led to a better utilization of the available fuel-air mixture and finally generated higher cylinder pressure on the power stroke which consequently delivered higher brake power, torque and brake mean effective pressure.

The effect of ram-swirled air intake is more obvious at part load to high load conditions. While maintaining a brake power slightly above the normal intake at low load conditions, greater performance at higher load gives the upper hand when additional power is needed. The experimental results have also shown that the emissions of NO_x are decreased substantially at high and low load conditions, while the pollutants resulting from incomplete combustion such as CO and UHC are significantly reduced at part load to high load conditions with the use of a parametric-swirl device. The improvement in the engine performance and emissions further validates the analysis done in the parametric simulation, where the ANOVA test has statistically concluded the significance of the studied factors in affecting the parametric swirl intensity.

Lastly, the measurement of uncertainties should be linked to the significance and reliability of the outcomes by comparing the uncertainty margin to the size of the experimental outcome. The uncertainty can also assess the sensitivity of the outcomes to the changes in the experimental conditions or the measurement techniques. By clearly identifying and discussing the sources of uncertainty in the measurement process, the overall reliability of the outcomes can be assessed and understood well.

CHAPTER 6

CONCLUSIONS AND RECOMMENDATIONS

As mentioned in the early chapters, the CI engine plays a significant role in the energy sector which includes the transportation and industrial sectors, however, the stricter emission standards imposed globally are the current challenges for any country. As the internal combustion engine's contribution is substantial to economic growth, continuous improvement to the internal combustion engines is vital for the sustainability of energy. After completing the experimental work of the CI engine operated with a swirl air intake manifold device, several important findings were discovered. These discoveries provide further opportunities for continuous improvement to the engine performance and emissions, where the improvement is not only limited to CI engines but also to the other modes of internal combustion engines.

6.1 Summary of Findings

- **Effect of Helical Air Intake Device Parameters to the Air Flow Characteristics and Swirl Intensity**

The swirl air intake model for use in a single-cylinder diesel engine has been simulated to study the effects of the parameters design on the swirl air flow characteristics. The development of the device in two phases provides conclusive findings towards the effective formation of ram-swirled air induction.

In general, the simulation works in the first and second phases were done to observe the swirl intensity, flow velocity, pressure difference, turbulence kinetic energy, turbulence energy dissipation and other flow characteristics that may be affected by the change of input parameters. This includes the change in the

swirl parameter combinations and mesh types. The first phase CFD simulation provides the finding and suggestion for a better base design in progression to the second phase simulation, where the swirl formation was found to be better on six (6) blades with 0.25 and 0.5 revolutions considering the trend in the TKE and TED characteristics in every design.

In the second phase of CFD simulation, a parametric simulation study was done by employing 0.5 revolutions as a constant helical blade parameter to investigate the effect of changing the 'ram' parameters to the swirl intensity and flow characteristics. The second phase simulation found that the ram effect led to a substantial increase in the mass flow rate of air, velocity and pressure difference in the parametric-swirl air intake model. The simulation also suggested that the TKE characteristics became higher at an increasing device length and draft angle, indicating a significant increase of kinetic energy in the flow due to the effect of the ram pressure.

Further analysis of the coefficient of determination R^2 value found that the draft angle was highly correlated to the increase of TKE compared to the length which had a low correlation, suggesting the draft angle had a stronger relationship to the increasing trend of TKE in the parametric-swirl model. The findings also showed a variation in the swirl intensity with the trend of low-swirl intensity being found to be affected by the greater length, while the larger draft angle was seen to increase the swirl intensity constantly. This relationship can be used as a guiding factor in the parameter selection for subsequent tests. The optimum parametric combination as provided by the simulation results should be 50mm length and 18° draft angle. However, due to the size of the

test engine intended for the verification stage, thus, the suggested combination for verification, among other combinations, was 70mm in length and 18° draft angle.

- **Effect of Ram Swirled Helical Air Intake Device on Engine Performance and Emissions**

The selected swirl air intake models were tested in the experimental stage to verify their effectiveness in improving the performance, combustion and emissions. The experiment was conducted in two parts of testing. This involved the testing of the three main prototypes of different helical revolutions which were previously simulated in the first phase simulation.

Nine (9) experimental runs were conducted according to the Taguchi Orthogonal Array design and each run was done with the observation of the brake power, torque, brake specific fuel consumption, emissions and other standard engine parameters. The first part of testing provides the suggestion for a better base-swirl design in progression to the second part of testing, where the engine performance was found to be better on the swirl air intake with 0.25 and 0.5 revolutions compared to the results of baseline (normal intake) diesel engine by 2.9 to 5.3% at various engine speeds.

In the second part of the test, 0.75 revolution was not considered for further testing due to a lower maximum brake power at 3600 rpm compared to normal intake. Although the swirl intensity for this model was the highest as discovered by CFD simulation, however further analysis has shown that this model also had the highest air restriction as presented by the highest turbulence energy dissipation (TED) with slightly lower turbulence kinetic energy (TKE) compared

to 0.5 revolution. These were causing the ineffectiveness of inducted air charge especially at higher engine speeds. Maintaining the test conditions similar to the first part of testing, two sets of prototypes comprised of 0.25 and 0.5 revolutions were tested for further verification. Each of the sets consists of three major parametric combinations that were selected as the prototypes based on the highest and smallest possible geometries that are suitable for the engine inlet size.

The second part of testing on the diesel engine operated with the parametric-swirl device resulted in a significant improvement to the maximum brake power by approximately a 6% increase compared to the results of the baseline engine. At higher engine load conditions, the brake power increased by a maximum of 17.1 to 18.4% higher than the baseline engine. Greater performance at higher load conditions while maintaining the brake power slightly above the normal intake at low load conditions gave the upper hand to the engine operated with the parametric-swirl device as soon as the additional power was required at any range of speeds. Improvement in the exhaust emissions was also discovered, where the NO_x was substantially decreased at high and low load conditions by 47%, while the other air pollutants such as CO and UHC were also reduced by 71.2% and 40.7% respectively at part load to high load conditions. This is due to more complete combustions by the improvement of fuel-air mixing homogeneity.

6.2 Recommendations for Future Works

- **Expansion of Swirl Air Intake Manifold Device Testing at a Different Mode and Application.**

Evaluation in a different mode of combustion in an HCCI or SI engine is highly recommended to expand the contribution of the swirl air intake manifold device to promote better engine operation, improved performance and reduced emissions. This device was fully compatible with a CI engine operation and proven to increase the overall engine performance by means of better fuel-air mixing in the cylinder. Since internal combustion engine modes like HCCI and SI are also dependent on the homogeneity of the fuel-air mixture to have a better combustion efficiency, the ram-swirled air intake charge is the practical solution for engine inefficiencies due to incomplete combustion and may have a significant effect on the overall engine performance and operating range. The ram-swirled air intake device also can contribute to future improvement of the engine performance with reduced exhaust emissions that can be further accomplished by various kinds of alternative fuels such as biodiesel made from palm oil, corn oil, sunflower oil, algae oil and cottonseed.

Apart from biofuels as an alternative fuel in IC engines, hydrogen has also been known as an excellent alternative fuel due to its clean properties and does not produce carbon dioxide (CO_2), CO, HC and PM. However, because hydrogen is unstable to be used as a single fuel in HCCI engines, most studies use it as an additive to control the ignition timing or increase the engine performance. The benefits of all these developments that may be potentially obtained from the CI engine give a strong indication that the current

improvement to the compression ignition mode of the engine is relevant and significant as the groundwork of other developments.

It is also highly recommended to have more variation of swirl air intake structure design in future works, where the test in different sizes of engines can be done to cover both automotive and industrial engines. Since the internal combustion engine is currently used as the main powering unit in the industrial sector, it is important to assess the device in a different application to evaluate the full potential of the device. This is because the engine can be operated under a different working environment and requirements with various types of fuels which include diesel-biodiesel blends to consider the total operation costs and limitations by the local regulations. For the transportation sector, further testing on the real driving condition is necessary for consideration of the actual driving pattern, where external factors such as the weather, surrounding air temperature, road conditions and driving behaviour will be further considered.

- **The Effect of Engine and Air Intake Temperature on the Swirl Intake Charge**

The future work is required to determine the effect of the interaction of the engine or air intake temperature with the swirl intake air on the performance of the engine. Engine or air intake temperature shall be set as a controlled factor to consider the effect of air intake temperature on the behaviour of the engine operated with a swirl air intake device. Based on the previous analysis and findings, the effect of the swirl airflow for the base-swirl air intake device was inconclusive at the higher load conditions due to the inconsistency in the data trend. This could be affected by the engine or air intake temperature at full load conditions. To have a conclusive result, this may require detailed

investigations at the specific engine or intake temperature level, for example, according to specific engine load, which is not covered in this study.

The previous chapter briefly explained that a controlling measure of the engine temperature at a higher engine load was always needed due to the rapid temperature increase. As shown by the experimental setup, the tested diesel engine was an air-cooled type generator set engine, which apparently was insufficient to maintain the normal engine temperature at higher load operations. In general, air-cooled systems are mainly used for smaller, lightweight engine applications such as bikes and scooters. It is often used in the engine design for an electric generator. Therefore, testing in water-cooled engines is more appropriate to see the effect of the swirling airflow at full load conditions, considering the inconsistency in the results at higher engine loads due to the variation of the engine operating temperature.

- **The Real-time Swirl Airflow Characteristics Evaluation**

The future study of the swirling airflow in the engine air intake system can be further upgraded by determining the swirl behaviour in steady flow test equipment and in actual engine performance tests in real time. Fabrication of a steady flow instrument and a special apparatus are essential to observe the flow characteristics and change in the swirl intensity in actual conditions. This can provide direct confirmation of the airflow data and the swirl formation generated by the swirl device in comparison to the CFD simulation results.

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APPENDICES

APPENDIX A:

Engine Exhaust Gas Temperature Data

Table A.1: Test Raw Data of the Exhaust Gas Temperature (EGT) Against Speed

Speed	rpm	1800	2000	2200	2400	2600	2800	3000	3200	3400	3600
Unit	-	°C	°C	°C	°C	°C	°C	°C	°C	°C	°C
Normal Intake	T1	464.4	482.4	480.7	516.4	504	499.5	459.9	472.5	481.1	460.4
	T2	457.3	467.1	507	519.1	489	513	453.8	465.8	475.3	440.6
	T3	467	492.2	503.7	493.6	435.4	498.1	451.5	464.7	467	444.6
Helical-Base 1-D1	T1	461.1	485.2	484.2	519.5	510.5	518.5	467.1	474.3	490.9	479.9
	T2	458.3	481.8	483.8	516.1	505.1	513.9	456.2	466.3	473.7	433.6
	T3	467.3	469.9	501.5	492.2	538.7	512.5	453.3	465.3	468.6	444.4
Helical-Base 1-D2	T1	467.2	472.2	502.6	495.2	535	508.5	449.1	460.7	468	442.3
	T2	463.5	477.2	504.1	499.2	541.6	503.7	456.7	466.3	476.1	438.3
	T3	462.3	482.7	504.3	498.1	541.8	504	456.6	468.3	475.5	451.4
Helical-Base 1-D3	T1	459.4	459.8	493.6	501.9	527.3	470.3	443.4	455.4	467.8	460.7
	T2	456.3	461.5	484.8	514.7	495.6	491.6	444.8	458.4	469.4	449.8
	T3	473.7	485.2	505.3	509.1	537	496.6	449.8	464.74	477	476.4
Helical-Base 2-D1	T1	443.4	449.5	477.9	473.9	509.5	477.9	419	432.7	435.5	421.7
	T2	439.5	446.7	478.8	467.4	503.6	465.6	421.4	434.1	438.1	409.6
	T3	469.5	483.6	488.7	578.9	521.9	512.7	451.7	461.6	471.6	473.8
Helical-Base 2-D2	T1	411	425.9	423.4	458.3	441.7	448.4	382	391.9	402.8	403.3
	T2	426.9	419.8	467.5	443.4	490.6	448.4	410.8	423.4	429.4	412.1
	T3	416.9	431.7	439.2	445.7	437.1	431.3	382.9	391.5	395.8	404.8
Helical-Base 2-D3	T1	440.6	446.2	474.3	472.3	496.1	470.9	413.1	427.6	438.5	433.4
	T2	433.4	459.1	462.9	492.6	489.3	461.8	407.5	420.8	431.3	429.2
	T3	439.6	448.3	468.1	496.1	494.4	472.4	412.1	425.6	436.4	435.5
Helical-Base 3-D1	T1	467.8	486.9	502	503.9	533.5	501.1	448.7	458.1	475	471.8
	T2	459.8	466.4	477.5	522.3	501.7	497	451.1	463.8	470.4	450.2
	T3	466.8	470.6	505.1	487.6	534.4	496	445.2	460.5	467.7	438.9
Helical-Base 3-D2	T1	467.5	487.9	485	522.7	496.8	489.9	443.6	455.2	461.4	416.5
	T2	453.9	459.4	497.6	525.5	521.7	511.3	442.1	451.4	463.3	465.6
	T3	456.2	485.1	491.3	521.1	511.6	511.8	435.9	436.3	468.2	476.4
Helical-Base 3-D3	T1	471.2	487.1	491.1	524.1	508.3	518.5	436.3	448.4	456.8	426.9
	T2	453	459.5	497	498.1	523.5	487	457.3	436.1	472.8	459.8
	T3	453	463.2	499.6	505.1	508.5	496.1	407.7	453.3	467.2	473
Helical-P 1	T1	420.2	440.2	473.3	460.4	502.8	457.1	411.6	420	428.5	414.1
	T2	433.4	462	469.6	441.7	490.3	438	412.1	431.5	457.1	418.2
	T3	425.3	444	479.8	479.3	491.5	447.2	403.1	411.8	424.7	425.8
Helical-P 2	T1	432.4	441.8	480.6	460.9	500.7	458.5	414.8	430.6	471.9	459.9
	T2	429.2	459.8	457	501	459.1	436.9	414.4	427.3	439.4	446.4
	T3	436.4	476.2	467.8	484.1	510.3	468.7	412.2	425.2	429.3	415.6
Helical-P 3	T1	439.4	452.4	488.1	480.8	509.9	457.5	398.1	419.6	426.6	409.7
	T2	435.1	454.8	479.8	474.6	510.6	467.8	409.2	416.7	428.2	435.8
	T3	436.7	471.3	466.2	505	484.1	477.1	414.9	425.2	437.1	461.8
Helical-P 4	T1	415.3	429.2	458.8	445.6	477.3	417.5	398.1	408.4	411.2	400.9
	T2	415.1	445.6	434.3	479.2	450.4	435.8	397.29	407.2	413.8	389.8
	T3	411.7	443.4	441	485.7	482	438.2	390.7	393.4	413.7	409.5
Helical-P 5	T1	425.6	426.6	461.5	438.6	478.6	436.6	405.2	416	418.1	409
	T2	420.1	424.1	458.1	447.1	480.7	447.9	406.7	417.7	420.3	387.3
	T3	416	433.5	464.7	444.8	474.5	426.7	412.8	419.4	423.8	412.1
Helical-P 6	T1	414.6	453.1	455.2	484	484.1	439.4	407.3	412.9	399.5	396
	T2	431.2	460.4	460	474.9	504.7	462.4	411.6	424.5	429.9	422.7
	T3	416.5	455.7	462	445.4	479.3	443.5	408.2	415.2	419.7	408.7

APPENDIX B:

Engine Load Data

Table B.1: Test Raw Data of the Engine Load Against Speed

Speed	rpm	1800	2000	2200	2400	2600	2800	3000	3200	3400	3600
Unit	-	%	%	%	%	%	%	%	%	%	%
Normal Intake	T1	89	73	66	54	49	44	38	38	37	36
	T2	91	75	60	53	50	45	39	38	38	38
	T3	99	68	64	57	50	46	40	39	39	38
Helical-Base 1-D1	T1	88	75	64	53	49	44	38	38	38	36
	T2	91	75	66	54	51	46	38	38	37	36
	T3	92	84	64	60	51	47	40	39	39	38
Helical-Base 1-D2	T1	100	85	69	62	51	48	40	40	39	38
	T2	97	80	67	58	50	46	39	39	39	38
	T3	97	84	68	61	50	47	40	39	38	37
Helical-Base 1-D3	T1	94	81	65	56	48	44	38	37	37	36
	T2	93	81	68	54	51	45	39	38	38	37
	T3	99	84	69	60	51	46	39	38	38	38
Helical-Base 2-D1	T1	93	80	63	57	49	46	38	38	37	36
	T2	98	80	66	56	49	46	39	38	38	37
	T3	89	75	66	54	50	44	38	38	37	36
Helical-Base 2-D2	T1	92	82	71	59	54	49	41	40	40	38
	T2	98	84	64	59	50	46	39	38	38	37
	T3	81	69	58	49	47	46	39	38	38	37
Helical-Base 2-D3	T1	100	79	66	58	51	46	39	38	37	35
	T2	98	82	69	55	52	45	38	38	37	36
	T3	98	83	74	59	55	48	40	40	39	38
Helical-Base 3-D1	T1	97	76	62	56	50	46	39	39	39	38
	T2	85	74	66	53	50	46	39	39	39	38
	T3	99	85	67	60	52	46	40	39	39	38
Helical-Base 3-D2	T1	98	79	70	57	54	46	39	38	38	36
	T2	93	86	76	60	54	46	39	39	37	36
	T3	99	77	68	55	52	46	38	38	38	37
Helical-Base 3-D3	T1	100	84	72	57	54	47	39	38	37	37
	T2	93	80	65	55	48	44	36	37	37	36
	T3	98	82	63	55	51	46	38	39	39	38
Helical-P 1	T1	100	85	67	59	51	47	39	39	38	36
	T2	100	83	69	61	52	46	39	39	36	38
	T3	100	83	71	59	52	45	39	38	37	36
Helical-P 2	T1	100	83	67	59	51	46	39	39	35	35
	T2	100	85	72	56	53	46	39	38	37	38
	T3	99	85	68	57	50	46	38	38	37	36
Helical-P 3	T1	100	88	76	64	53	47	40	38	37	36
	T2	100	84	71	61	53	47	39	39	38	36
	T3	98	89	71	57	53	48	39	40	39	36
Helical-P 4	T1	94	79	64	58	49	45	39	38	37	36
	T2	97	80	71	56	52	45	39	38	37	36
	T3	98	86	73	54	49	45	38	37	37	36
Helical-P 5	T1	94	82	66	59	50	45	40	39	38	38
	T2	96	83	69	57	51	46	40	39	38	38
	T3	93	83	66	57	51	45	40	40	39	39
Helical-P 6	T1	96	90	61	58	51	46	40	39	39	37
	T2	99	87	71	59	51	46	38	38	37	36
	T3	98	83	67	58	51	47	40	39	39	38

APPENDIX C:

Air-Fuel Ratio Measurement Results

The air-fuel ratio is the ratio of the mass of air to the mass of fuel being supplied to the engine. It is typically expressed as parts of air to one part of fuel, where the typical ratio is 14.5:1 for a stoichiometric air-fuel ratio in a diesel engine.

The air-fuel ratio for diesel engines is typically not controlled as precisely as in a gasoline engine as the diesel engines often operate with a relatively high excess of air resulting in a lean air-fuel mixture (above 14.5:1) (Vinod Raj & Dheikshith Raj, 2023). This is mainly because the engine relies on the self-ignition properties of the fuel and the heat generated by the air compression rather than relying on a stoichiometric air-fuel ratio for combustion. The primary control of the air-fuel ratio is achieved through the fuel injection system.

To further highlight, the fuel injection of the tested engine is not electronically controlled by the ECU programme. It is fully relying on manual fuel throttling without a system to receive feedback from the oxygen sensor. While it could be possible for the air-fuel ratio to be affected by the inducted air charge, it may not be too significant to change the overall profile out from the range determined by the fuel injection system. The air-fuel ratio for a full range of operations is presented in Figure C1 to C4. It can be observed that the air-fuel ratio for 0.25 and 0.5 revolutions are generally lower than the normal intake by up to 6.25% and 5.17% respectively, while 0.75 revolution is higher than the normal at most of the engine speeds by up to 4.57%. It is essential to highlight that the ratio is all above the stoichiometric ratio (14.5:1) based on the fact that diesel will always run under a lean mixture.

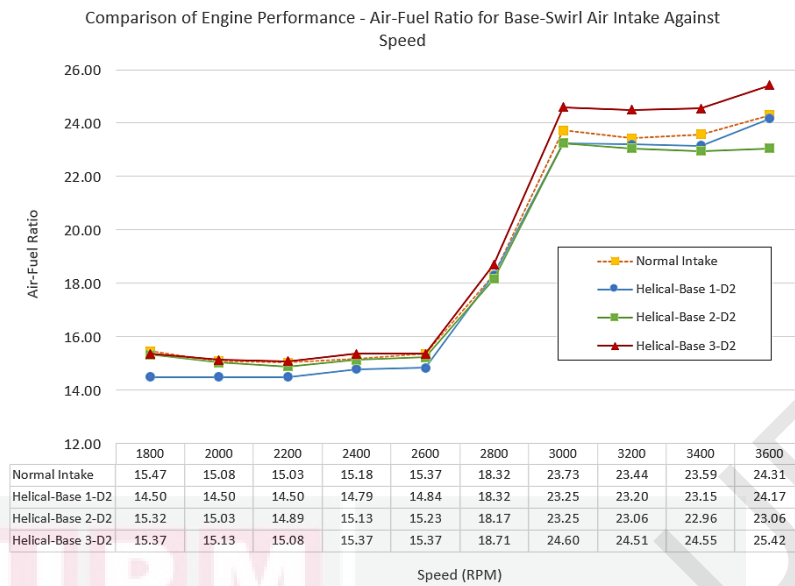


Figure C1: Comparison of Engine Performance – Air/Fuel Ratio for Base-Swirl Air Intake Against Speed

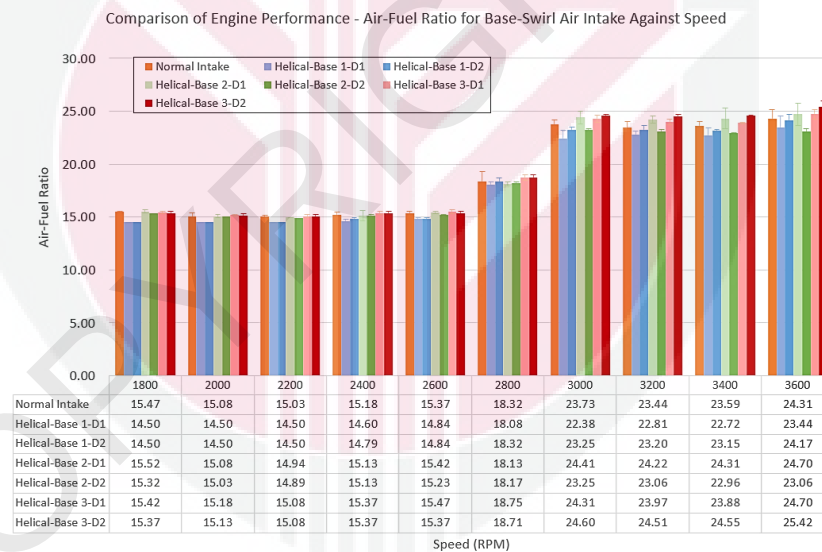


Figure C2: Comparison of Engine Performance – Air/Fuel Ratio for Base-Swirl Air Intake at Different Distances Against Speed

While the swirl intensity increases the rate of the fuel-air mixing for thorough mixing of fuel with available air, the greater mass of oxygen per unit volume of air increases the oxygen content in the cylinder, leading to more complete combustion. More complete combustion means more air is effectively used,

causing less residual air in the exhaust gases. This led to a slightly richer mixture as detected by the oxygen sensor compared to the base-swirl results. Nevertheless, this 'richer' mixture is still above the stoichiometric ratio and is thus considered a lean fuel-air mixture. It should indicate a normal operation for a CI engine.

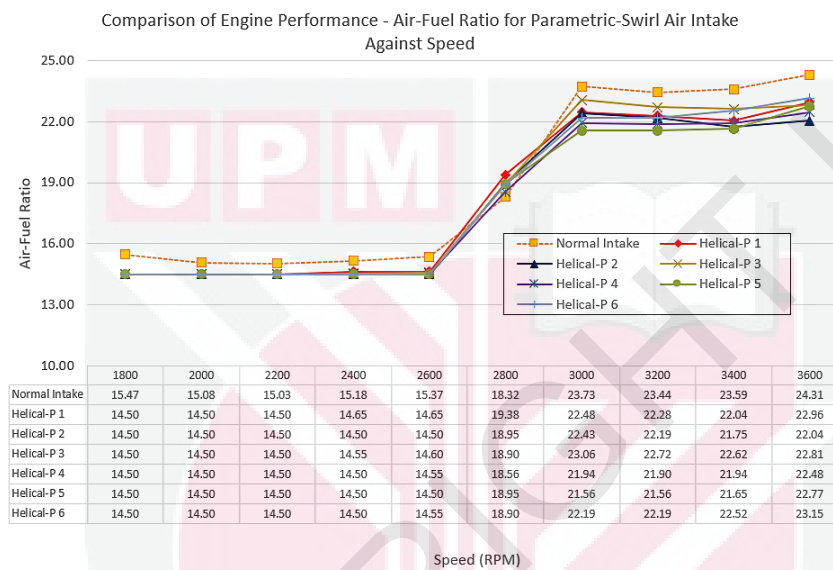


Figure C3: Comparison of engine performance – Air/Fuel Ratio for Parametric-Swirl Air Intake Against Speed

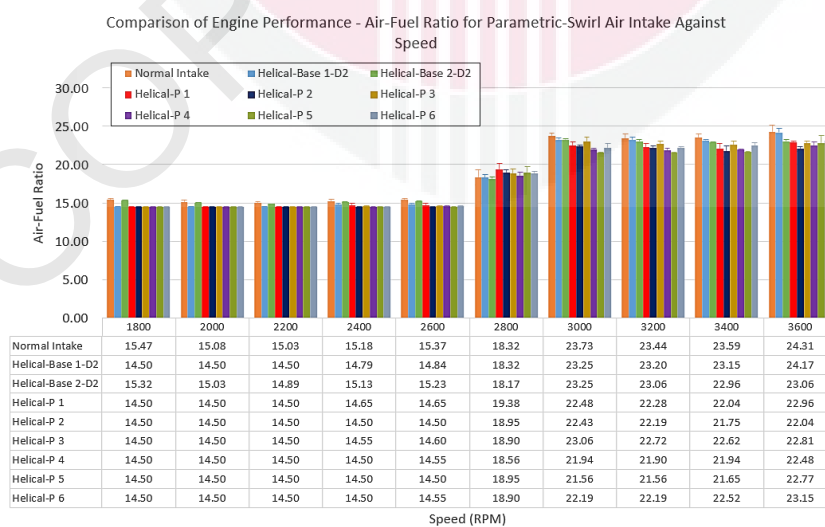


Figure C4: Comparison of engine performance – Air/Fuel Ratio for Base and Parametric-Swirl Air Intake Against Speed

APPENDIX D:

Simulation runs of the parametric study

Table D.1 shows the summary of the full simulation run planning based on the designated parameters set with the General Full Factorial Method ($L_{49}7^2$).

Table D.1: Full simulation runs of the parametric combination

Run	Draft Angle	Device Length	Responses
1	1.26	50	Studied responses (From CFD simulation post-processing data):
2	1.26	70	
3	1.26	80	
4	1.26	90	
5	1.26	100	
6	1.26	110	
7	1.26	120	
8	3	50	1. Velocity-Outlet 2. Swirl Number (Intensity) 3. Pressure-Inlet 4. Pressure Difference 5. Turbulence Kinetic Energy (TKE) 6. Turbulence Energy Dissipation (TED) 7. Mass Flow Rate of Air
9	3	70	
10	3	80	
11	3	90	
12	3	100	
13	3	110	
14	3	120	
15	6	50	
16	6	70	
17	6	80	
18	6	90	
19	6	100	
20	6	110	
21	6	120	
22	9	50	
23	9	70	
24	9	80	
25	9	90	
26	9	100	
27	9	110	
28	9	120	
29	13	50	
30	13	70	
31	13	80	
32	13	90	
33	13	100	
34	13	110	
35	13	120	
36	16	50	
37	16	70	
38	16	80	
39	16	90	
40	16	100	
41	16	110	
42	16	120	
43	18	50	
44	18	70	
45	18	80	
46	18	90	
47	18	100	
48	18	110	
49	18	120	

LIST OF PUBLICATIONS

Alias, M.S.M., Hairuddin, A.A., Hassan, M.K., Rezali, K.A.M. (2023). *An Air Intake Device*. (Patent Application No. PI2023007910) Malaysian Patent.

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