

# Cyclone Intake Manifold Effects on Marine Diesel Combustion Using CFD

Fathur Rahman<sup>1\*</sup>, Semin<sup>2</sup>, Alfiy Alfatarizqi<sup>3</sup>

(Received: 30 July 2026 / Revised: 30 July 2026 / Accepted: 31 July 2026 / Available Online: 24 September 2026)

**Abstract**—Improving the air-fuel mixing process remains a practical route for increasing the combustion quality of naturally aspirated marine diesel engines. This study evaluates a passive cyclone installed in the intake manifold of a single-cylinder Yanmar TF-85 MH engine using three-dimensional computational fluid dynamics and experimental validation. Twenty-seven cyclone geometries were generated from combinations of vane height, vane length, and twist angle, and simulations were performed at 1,600, 1,800, and 2,000 rpm. The numerical model employed an RNG k-epsilon turbulence formulation, moving-valve and piston domains, a chemical-kinetics combustion solver, and a B40 surrogate composed of methyl decanoate and n-dodecane. The optimum geometry, designated GD 11560, increased turbulent kinetic energy from 330.35 to 367.34 m<sup>2</sup>/s<sup>2</sup>, or 11.20%, while reducing the inducted air mass by 4.12%. The indicated mean effective pressure increased by 1.88% at 1,600 rpm and 0.36% at 1,800 rpm, but decreased by 0.87% at 2,000 rpm. Maximum combustion temperature and pressure rose by 2.43% and 1.08%, respectively, at 1,600 rpm. Validation errors remained below 5%. The cyclone therefore improves low-to-medium-speed combustion through stronger axial-tangential motion, although its flow restriction becomes important at high speed.

**Keywords**—air-fuel mixing; effective pressure; marine propulsion; turbulence.

\*Corresponding Author: 6019241015@student.its.ac.id

## I. INTRODUCTION

Marine diesel engines remain central to small-vessel propulsion and auxiliary power because they combine high compression efficiency, robust torque characteristics, and long service life. Their performance, however, depends strongly on a process that occupies only a brief interval of the engine cycle: the evaporation, entrainment, and mixing of injected fuel with hot compressed air. A weakly organized intake flow can leave locally fuel-rich regions, delay oxidation, and prolong late combustion. These effects reduce the fraction of released chemical energy converted into useful indicated work and can increase incomplete-combustion products [1], [2].

Air motion in a direct-injection diesel cylinder is not merely a transport mechanism. Swirl, tumble, squish, and small-scale turbulence determine how the spray plume encounters oxygen, how quickly droplets break up and evaporate, and how heat release develops around top dead center. Swirl is especially influential because its angular momentum may survive the intake and compression strokes, then interact with the spray during the premixed and mixing-controlled phases. Earlier work has shown that organized rotational motion can improve mixture preparation, but excessive intensity may also accelerate wall heat transfer or restrict the amount of inducted air [3]. The design problem is consequently an optimization problem, not a search for the largest possible swirl value.

A number of engineering routes have been used to improve diesel combustion. High-pressure common-rail injection, variable valve timing, turbocharging, exhaust-energy recovery, and sophisticated combustion-chamber geometries can all produce substantial benefits. Nevertheless, many of these solutions require expensive hardware, electronic control, or major modification of an existing marine engine [4], [5]. A passive intake device is attractive in this context because it can be fitted with limited intervention, has no moving actuator, and uses the incoming flow itself to generate rotational motion.

The intake port and manifold geometry govern the compromise between flow capacity and swirl generation. Variable swirl control valves, helical and tangential ports, and guide vanes have demonstrated that redirecting air near the valve can intensify rotational motion [6], [7]. Guide-vane studies further indicate that vane number, length, and inclination influence both the in-cylinder flow structure and the associated pressure loss [8], [9]. Numerical studies of high-viscosity and alternative-fuel applications also underline the importance of maintaining turbulent mixing when the fuel is more difficult to atomize [10].

A cyclone intake insert is a compact form of a static guide-vane device. The vanes impose a tangential velocity component on an otherwise predominantly axial flow, causing the streamlines to spiral before the air crosses the intake valve. Experiments and simulations on small engines have reported improvements in torque or flow organization after cyclone installation [11]–[15]. Yet results are not universally favorable. The solid vane area occupies part of the duct, increases wetted surface, and introduces turning losses. At sufficiently high engine speed, the resulting reduction in volumetric efficiency may offset the gain in mixture quality.

The literature still leaves two connected gaps. First, many previous investigations examine a single cyclone shape or evaluate engine outputs without resolving the complete intake-to-cylinder flow field. Second, studies that screen a broad geometry matrix and then connect

---

Fathur Rahman, Department of Marine Engineering, Institut Teknologi Sepuluh Nopember, Surabaya, 60111, Indonesia. Email: 6019241015@student.its.ac.id

Semin, Department of Marine Engineering, Institut Teknologi Sepuluh Nopember, Surabaya, 60111, Indonesia. Email: semin@its.ac.id

Alfiy Alfatarizqi, Department of Marine Engineering, Institut Teknologi Sepuluh Nopember, Surabaya, 60111, Indonesia. Email: alfiyalfatarizqi.research@gmail.com

turbulent kinetic energy, inducted mass, indicated mean effective pressure, maximum temperature, and maximum pressure remain limited for a naturally aspirated marine diesel engine. A geometry may look effective in a local velocity contour yet perform poorly when the mass-flow penalty is included. A credible assessment therefore requires a coupled reading of flow and combustion metrics, supported by validation against measured engine data.

This study addresses those gaps through three-dimensional computational fluid dynamics of a Yanmar TF-85 MH single-cylinder marine diesel engine. Twenty-seven cyclone geometries are screened by varying vane height, vane length, and twist angle. The best candidate is subsequently compared with the baseline manifold at 1,600, 1,800, and 2,000 rpm. The research objectives are to determine how the cyclone changes the intake and in-cylinder flow distribution, quantify its influence on indicated mean effective pressure, and explain how the modified flow affects peak combustion temperature and pressure. The contribution lies in revealing the speed-dependent trade-off between turbulence enhancement and intake restriction, rather than treating swirl generation as an unconditional performance gain.

## II. METHOD

### A. Research Design and Engine Configuration

The investigation combined numerical simulation with experimental validation. Dimensional measurements were obtained from the intake manifold, valves, piston, cylinder head, and exhaust passage of the test engine in the Marine Power and Propulsion Laboratory at Institut Teknologi Sepuluh Nopember. The measured geometry was reconstructed in a computer-aided design environment and imported into ANSYS Forte. Two primary configurations were analyzed: the original intake system, referred to as the baseline, and the same system equipped with a passive cyclone insert.

TABLE 1.  
ENGINE, MANIFOLD, AND CYCLONE SPECIFICATIONS

| Parameter                | Value                          |
|--------------------------|--------------------------------|
| Engine type              | 1-cylinder, 4-stroke DI diesel |
| Displacement             | 493 cm <sup>3</sup>            |
| Bore x stroke            | 85 x 87 mm                     |
| Compression ratio        | 18:1                           |
| Connecting rod           | 143 mm                         |
| Rated output             | 8.5 hp at 2,200 rpm            |
| Intake diameter          | 34.6 mm                        |
| Intake length            | 66.7 mm                        |
| Cyclone diameter         | 34.6 mm                        |
| Cyclone thickness/length | 2/30 mm                        |

The engine was a naturally aspirated, four-stroke, direct-injection Yanmar TF-85 MH. It has one cylinder, a swept volume of 493 cm<sup>3</sup>, an 85 mm bore, an 87 mm stroke, a 143 mm connecting rod, and a compression ratio of 18:1. The intake manifold inner diameter was 34.6 mm, its wall thickness was 2.8 mm, and its measured length was 66.7 mm. Injection began at 18 degrees before top dead center through four 0.22 mm

nozzle holes with an included spray angle of 150 degrees. The main engine and intake dimensions are summarized in Table 1.

### B. Cyclone Geometry and Screening Matrix

The cyclone was designed as a cylindrical insert containing internal vanes. Its outside diameter matched the intake passage so that the device could be installed without changing the original runner. Three geometric variables were combined in a full factorial matrix: vane height of 5, 8, and 11 mm; vane length of 30, 40, and 50 mm; and twist angle of 30, 45, and 60 degrees. These levels generated 27 designs. The code GD 11560 identifies the candidate with an 11 mm vane height, a 50 mm vane length, and a 60-degree twist angle.

Geometry screening was conducted at 2,000 rpm because the high intake demand exposes both the swirl-generating capability and the restriction caused by the insert. Each candidate was assessed using volume-averaged turbulent kinetic energy and in-cylinder air mass. Turbulent kinetic energy was used as an indicator of the velocity fluctuations available for mixing, while inducted mass represented the breathing penalty. The three geometries with the highest turbulent kinetic energy were examined in greater detail through contours and streamlines. The candidate showing the strongest useful turbulence was then carried into the combustion simulations at all three engine speeds.

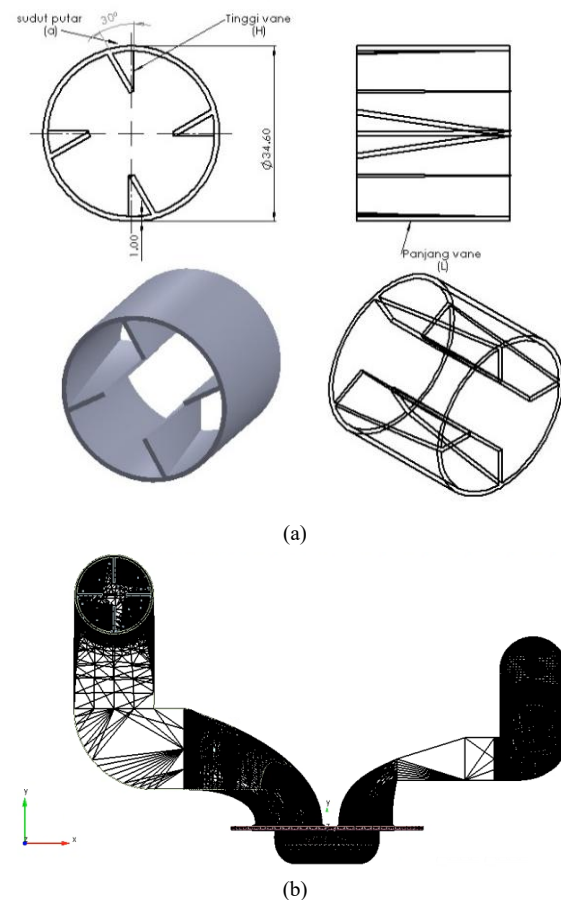


Figure 1. Cyclone geometry (a) and computational mesh used in the intake-to-cylinder domain (b).

### C. Computational Domain, Mesh, and Physical Models

The moving-domain simulation covered the inlet, intake runner, intake valve, cylinder, exhaust valve, and exhaust port. Automatic mesh generation was applied

using a global size of 0.2 cm. Local refinement was specified at one-quarter of the global size around the inlet, intake valve, exhaust valve, cyclone surfaces, and cylinder volume, while the general wall region used one-half of the global size. This refinement strategy concentrated cells in zones where valve motion, vane-induced turning, spray development, and near-wall gradients were expected to be strongest. Figure 1 shows the cyclone model and representative mesh.

Unsteady Reynolds-averaged Navier-Stokes equations were solved with the RNG k-epsilon turbulence model. The formulation provides transport equations for turbulent kinetic energy and its dissipation rate and is frequently used for strongly strained internal flows. The local turbulent kinetic energy was interpreted using (1), where  $u'$ ,  $v'$ , and  $w'$  denote fluctuating velocity components.

$$k = 1/2 (u'^2 + v'^2 + w'^2) \quad (1)$$

Mass conservation was enforced through the transient continuity equation, while momentum conservation resolved pressure, viscous, and inertial effects in the curved intake passage and moving cylinder. The total-energy equation accounted for convection, compression work, wall heat transfer, and chemical heat release. These coupled equations were discretized on the deforming mesh as the valves and piston moved through the engine cycle. The pressure-velocity solution was advanced in crank-angle increments until the monitored residuals and integrated cylinder quantities exhibited stable convergence.

The RNG k-epsilon closure was selected because the flow contains rapid strain, streamline curvature, valve-jet acceleration, and vane-induced rotation. It remains a Reynolds-averaged approximation and does not resolve every instantaneous eddy. Accordingly, the contour analysis emphasized robust differences between identically meshed configurations rather than treating small local fluctuations as exact physical measurements. The same numerical controls were retained for the baseline and every cyclone candidate so that geometric effects were not confounded by changes in solver settings.

TABLE 2.  
PRINCIPAL CFD AND COMBUSTION SETTINGS

| Item                    | Setting                               |
|-------------------------|---------------------------------------|
| Solver                  | ANSYS Forte, transient                |
| Turbulence              | RANS RNG k-epsilon                    |
| Global mesh             | 0.2 cm                                |
| Local mesh              | 1/4 at valves, cyclone, cylinder      |
| Inlet                   | 330 K; 101,325 Pa                     |
| Wall temperatures       | 500-550 K                             |
| Fuel surrogate          | 0.4 methyl decanoate + 0.6 n-dodecane |
| Injector                | 4 x 0.22 mm; 150 degrees              |
| Injection timing        | 18 degrees BTDC                       |
| Engine speeds           | 1,600; 1,800; 2,000 rpm               |
| Fuel/cycle at 1,600 rpm | 11.103 mg (2 kW); 16.095 mg (3 kW)    |
| Fuel/cycle at 1,800 rpm | 12.265 mg (2 kW); 17.706 mg (3 kW)    |
| Fuel/cycle at 2,000 rpm | 13.576 mg (2 kW); 17.939 mg (3 kW)    |

Mesh refinement near the cyclone and valve curtain was particularly important. The vane wake begins as a thin shear layer, while the valve curtain converts the intake stream into a high-speed annular jet. If these regions are under-resolved, the model can artificially dissipate the rotational motion before it reaches the cylinder. Local sizing therefore linked the swirl-generating device to the in-cylinder volume with a continuous refined path, while coarser cells were retained in less sensitive portions of the external ports to control computational cost.

The intake boundary was set to 330 K and 101,325 Pa. The intake port was initialized at 330 K, the exhaust port and both valves at 550 K, the cylinder head and liner at 500 K, and the piston at 550 K. The outlet used a static-pressure condition. Valve timing followed the measured Yanmar configuration, with the intake valve opening 12 degrees before top dead center and closing 44 degrees after bottom dead center, while the exhaust valve opened 40 degrees before bottom dead center and closed 18 degrees after top dead center.

B40 fuel was represented by a surrogate mixture of 40% methyl decanoate and 60% n-dodecane by mass. Spray breakup was treated using Kelvin-Helmholtz and Rayleigh-Taylor mechanisms, and chemical kinetics were enabled to resolve ignition delay and heat release. The injected mass was specified from the experimental operating data: 11.103 and 16.095 mg per cycle at 1,600 rpm, 12.265 and 17.706 mg at 1,800 rpm, and 13.576 and 17.939 mg at 2,000 rpm for the 2,000 and 3,000 W load points, respectively. Table 2 condenses the principal numerical settings.

#### D. Validation and Performance Metrics

The combustion model was validated using indicated mean effective pressure. Experimental brake mean effective pressure was converted to indicated mean effective pressure by adding friction mean effective pressure estimated from the engine operating condition [20]. The purpose of this comparison was not to force point-by-point identity but to verify that the model reproduced the magnitude and speed trend of the measured cycle work.

The absolute percentage error was calculated using The Willan's line method [21]. A prediction error below 5% was adopted as evidence that the model was sufficiently accurate for comparative assessment of the baseline and cyclone configurations. The final interpretation considered turbulent kinetic energy, inducted air mass, flow topology, indicated mean effective pressure, maximum gas temperature, and maximum cylinder pressure together. This multi-metric approach was essential because an insert can strengthen turbulence while simultaneously reducing the air supplied to the cylinder.

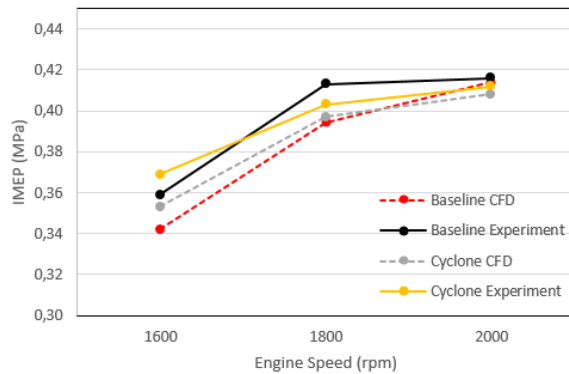
### III. RESULTS AND DISCUSSION

#### A. Model Validation

The simulated indicated mean effective pressure followed the experimental trend for both configurations. In the baseline case, the numerical values were 0.342, 0.394, and 0.414 MPa at 1,600, 1,800, and 2,000 rpm, whereas the corresponding experimental values were

0.359, 0.413, and 0.416 MPa. The absolute errors were therefore 4.83%, 4.54%, and 0.35%. For GD 11560, the numerical values were 0.353, 0.397, and 0.408 MPa, compared with experimental values of 0.369, 0.403, and 0.412 MPa. The errors were 0.04%, 0.01%, and 0.08%.

Figure 2 and Table 3 show that the numerical model slightly underpredicted indicated mean effective pressure but preserved the direction of change across engine speed. The growing error at higher speed is physically plausible because friction, leakage, wall heat transfer, and cycle-to-cycle variation become harder to represent with fixed or simplified submodels. Even so, all errors remained below 5%, supporting use of the model for relative comparisons. The agreement also indicates that the selected spray, combustion, and turbulence formulations represented the first-order behavior of the test engine.



**Figure 2.** Comparison of simulated and experimental indicated mean effective pressure.

TABLE 3.  
IMEP VALIDATION RESULTS

| Case     | rpm  | CFD   | Exp.  | Error |
|----------|------|-------|-------|-------|
| Baseline | 1600 | 0.342 | 0.359 | 4.83% |
| Baseline | 1800 | 0.394 | 0.413 | 4.54% |
| Baseline | 2000 | 0.414 | 0.416 | 0.35% |
| GD 11560 | 1600 | 0.353 | 0.369 | 0.04% |
| GD 11560 | 1800 | 0.397 | 0.403 | 0.01% |
| GD 11560 | 2000 | 0.408 | 0.412 | 0.08% |

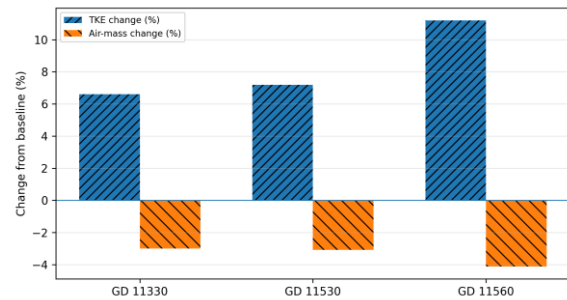
### B. Geometry Screening and the Turbulence-Mass Trade-off

All cyclone designs altered the intake flow, but the magnitude of the response depended on vane geometry. The baseline turbulent kinetic energy was 330.3493 m<sup>2</sup>/s<sup>2</sup> with an inducted air mass of 0.24618 g. GD 11330 increased turbulent kinetic energy to 352.1917 m<sup>2</sup>/s<sup>2</sup>, equivalent to 6.61%, while reducing air mass by 3.00%. GD 11530 produced 354.0897 m<sup>2</sup>/s<sup>2</sup>, a 7.19% increase, with a 3.08% mass reduction. GD 11560 reached 367.3446 m<sup>2</sup>/s<sup>2</sup>, an 11.20% increase, but its inducted mass fell to 0.23604 g, a 4.12% reduction.

The screening results reveal a clear design trade-off. Taller and longer vanes penetrate farther into the central flow and expose a larger surface to the incoming air. A 60-degree twist redirects a greater fraction of axial momentum into tangential momentum, strengthening the rotating structure. The same features also increase blockage, skin friction, and form drag. GD 11560 was selected because its turbulence gain was disproportionately larger than the additional mass-flow

loss relative to the other high-performing candidates. It was not a loss-free design, but it created the most energetic and persistent mixing field among the 27 alternatives.

This result agrees with the broader observation that intake devices should be evaluated simultaneously through swirl or turbulence and flow coefficient. Studies of swirl-control valves and guide vanes similarly report that stronger rotational motion is accompanied by reduced flow capacity when the opening or passage area becomes too restrictive [6]–[10]. Figure 3 makes that compromise visible: the selected design occupies the upper-left performance frontier in which turbulence gain is high but air-mass loss has not yet become catastrophic. A single metric would have hidden this balance.



**Figure 3.** Turbulent kinetic energy gain and inducted-air-mass penalty of the leading cyclone geometries.

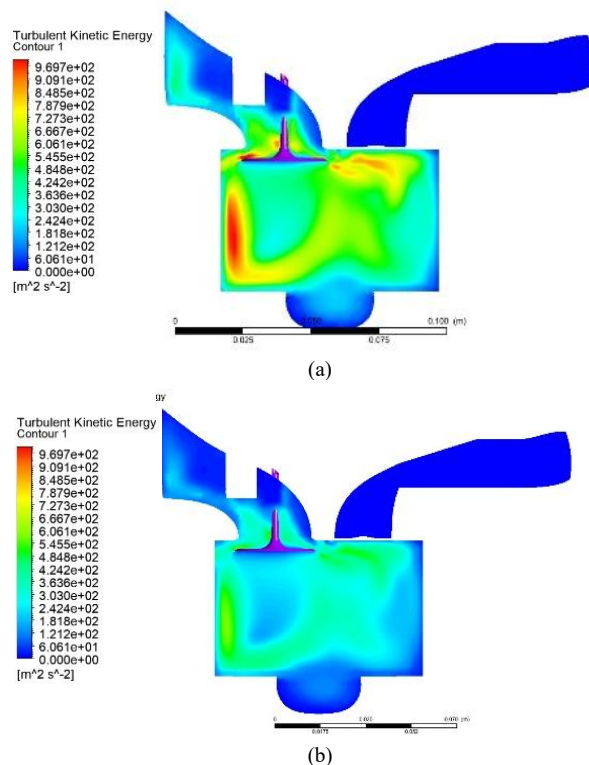
TABLE 4. |  
CFD SCREENING RESULTS FOR ALL CYCLONE GEOMETRIES AT 2,000 RPM

| Model    | TKE    | TKE change | Air mass | Mass change |
|----------|--------|------------|----------|-------------|
| Baseline | 330.35 | -          | 0.24618  | -           |
| GD 05330 | 346.81 | +4.98%     | 0.24082  | -2.18%      |
| GD 05345 | 347.13 | +5.08%     | 0.24058  | -2.27%      |
| GD 05360 | 347.87 | +5.30%     | 0.24048  | -2.32%      |
| GD 05430 | 347.85 | +5.30%     | 0.24090  | -2.14%      |
| GD 05445 | 348.03 | +5.35%     | 0.23905  | -2.90%      |
| GD 05460 | 348.10 | +5.37%     | 0.23874  | -3.02%      |
| GD 05530 | 348.72 | +5.56%     | 0.24085  | -2.17%      |
| GD 05545 | 348.95 | +5.63%     | 0.24022  | -2.42%      |
| GD 05560 | 326.22 | -1.25%     | 0.23790  | -3.36%      |
| GD 08330 | 350.54 | +6.11%     | 0.23963  | -2.66%      |
| GD 08345 | 350.54 | +6.11%     | 0.23963  | -2.66%      |
| GD 08360 | 348.69 | +5.55%     | 0.23963  | -2.66%      |
| GD 08430 | 349.76 | +5.88%     | 0.23997  | -2.52%      |
| GD 08445 | 349.78 | +5.88%     | 0.23963  | -2.66%      |
| GD 08460 | 348.06 | +5.36%     | 0.23945  | -2.73%      |
| GD 08530 | 349.78 | +5.88%     | 0.24006  | -2.49%      |
| GD 08545 | 350.93 | +6.23%     | 0.23940  | -2.75%      |
| GD 08560 | 348.73 | +5.56%     | 0.23845  | -3.14%      |
| GD 11330 | 352.19 | +6.61%     | 0.23881  | -2.99%      |
| GD 11345 | 350.26 | +6.03%     | 0.23886  | -2.97%      |
| GD 11360 | 350.40 | +6.07%     | 0.23867  | -3.05%      |
| GD 11430 | 351.81 | +6.50%     | 0.23895  | -2.94%      |
| GD 11445 | 351.05 | +6.27%     | 0.23884  | -2.98%      |
| GD 11460 | 349.03 | +5.66%     | 0.23843  | -3.15%      |
| GD 11530 | 354.09 | +7.19%     | 0.23860  | -3.08%      |
| GD 11545 | 352.03 | +6.56%     | 0.23680  | -3.81%      |
| GD 11560 | 367.34 | +11.20%    | 0.23604  | -4.12%      |

The complete matrix also shows that the geometric response was not monotonic. GD 05560, for example, reduced turbulent kinetic energy below the baseline even though it imposed a 3.36% air-mass penalty. Its combination of comparatively short vane height, long vane length, and steep twist likely produced dissipation and blockage without establishing a coherent large-scale vortex. In contrast, the 11 mm vane group generally maintained higher turbulent kinetic energy because its tips intercepted a more energetic central portion of the intake stream. This contrast confirms that vane area alone does not determine performance; the interaction among penetration depth, turning angle, and available development length is decisive.

### C. Flow-Field Transformation

The contour comparison in Figure 4 confirms that the change was not confined to the immediate wake of the vanes. In the baseline manifold, low turbulent kinetic energy dominated the runner, and only a localized increase appeared as the flow accelerated through the valve curtain. The absence of an upstream rotational device allowed the stream to retain a mainly axial character. Once GD 11560 was installed, higher turbulent kinetic energy emerged from the cyclone outlet, remained visible through the curved passage, and intensified at the valve gap before spreading across the cylinder.

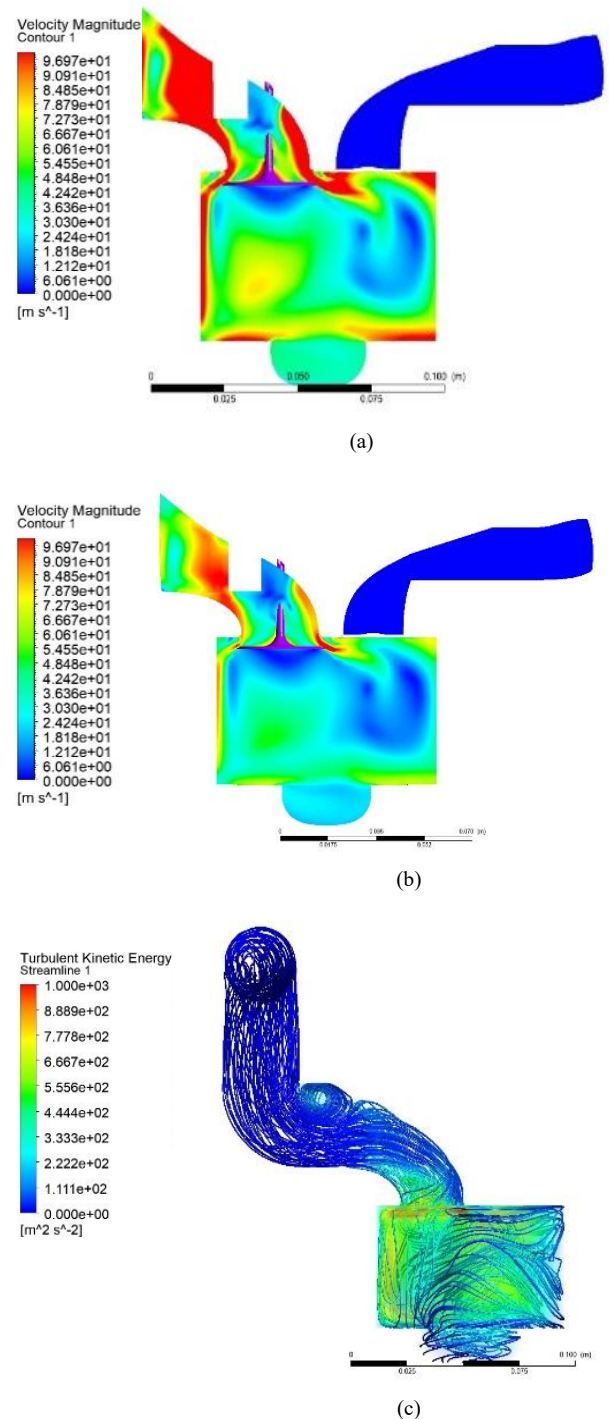


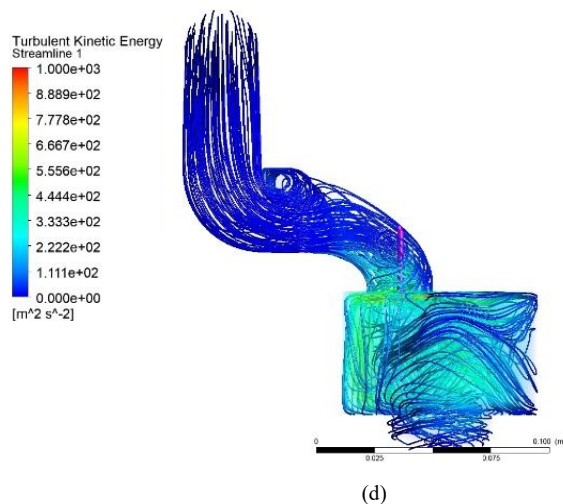
**Figure 4.** Turbulent kinetic energy contours for GD 11560 (1) and the baseline (2) configuration.

The selected cyclone generated a pronounced high-energy region from the valve curtain toward the opposite cylinder wall. This distribution indicates that the tangential component did not decay completely before the flow entered the chamber. Instead, the jet crossed the valve, impinged on the wall, and rolled into a larger rotating structure. The flow therefore shifted from a predominantly axial jet to a combined axial-tangential

motion. Such a structure increases the relative velocity between air and fuel droplets and repeatedly transports oxygen into spray-rich regions.

Velocity and streamline visualizations in Figure 5 support the same interpretation. The baseline case showed comparatively straight paths and a broader low-velocity zone. GD 11560 produced denser spiral streamlines and a valve-jet velocity approaching 97 m/s. Regions between approximately 48 and 85 m/s extended more deeply through the cylinder, reducing the area occupied by nearly stagnant air. The selected design thus improved not merely the peak velocity but the spatial reach of active flow. This point matters because effective diesel mixing depends on how much of the chamber participates in entrainment, not only on a narrow high-speed core.





**Figure 5.** Velocity contours for GD 11560 (a), baseline (b) and streamlines illustrating the axial-tangential flow induced by GD 11560 (c), Baseline (d).

The flow field also illustrates why geometric optimization is necessary. Intense local recirculation may dissipate energy without contributing to spray entrainment, and very aggressive turning can reduce the total charge mass. GD 11560 retained a coherent spiral rather than producing only isolated chaotic eddies. The visual evidence is consistent with the measured rise in volume-averaged turbulent kinetic energy. Similar behavior has been reported for guide vanes and modified ports, where a properly oriented device extends rotational motion into the cylinder and improves fuel-air interaction [8]–[12].

#### D. Combustion Response and Engine-Speed Dependence

The stronger mixing field produced a consistent rise in maximum combustion temperature. At 1,600 rpm, the baseline maximum was 1,934.93 K and the cyclone value was 1,981.97 K, an increase of 2.43%. At 1,800 rpm, the maximum rose from 1,992.48 to 2,031.04 K, or 1.94%. At 2,000 rpm, it increased from 2,020.77 to 2,058.54 K, or 1.87%. The increase at every speed indicates that the device promoted faster and more complete release of the available fuel energy during the resolved cycle.

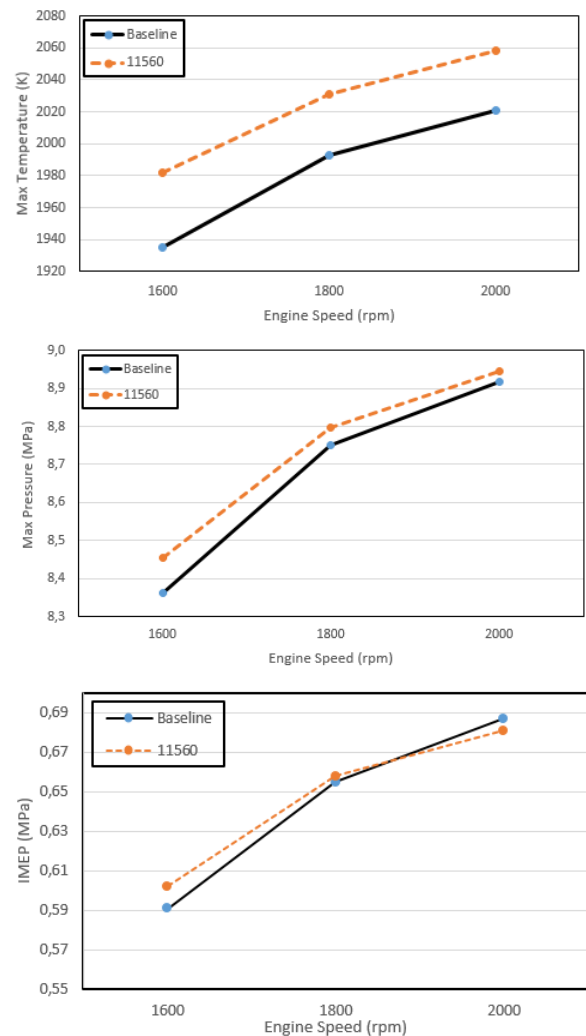
Maximum cylinder pressure followed the thermal trend. The pressure increased from 8.36 to 8.45 MPa at 1,600 rpm, from 8.75 to 8.79 MPa at 1,800 rpm, and from 8.918 to 8.944 MPa at 2,000 rpm. These changes correspond to 1.08%, 0.46%, and 0.29%. The larger low-speed response suggests that, when the time available for intake and mixing is relatively long, the cyclone can establish a coherent swirl structure before injection and intensify the premixed portion of combustion. The resulting earlier heat release raises both temperature and pressure near top dead center.

Indicated mean effective pressure did not follow a uniformly positive trend. GD 11560 increased it by 1.88% at 1,600 rpm and 0.36% at 1,800 rpm. At 2,000 rpm, however, indicated mean effective pressure decreased by 0.87%. This reversal is the most important engineering finding of the study. Peak temperature and pressure can still rise while integrated cycle work falls slightly if the inducted mass is reduced, heat release shifts unfavorably, or pumping losses increase. The

4.12% loss of air mass measured during screening becomes more influential as the engine demands a larger volumetric flow rate.

Figure 6 and Table 5 summarize this speed dependence. The cyclone behaves as a useful mixing device at low and medium speed, where its turbulence benefit dominates. At high speed, the vanes become an appreciable restriction. This behavior explains why passive swirl devices often show strong benefits in a limited operating region rather than across an entire engine map [11]–[15]. The result also cautions against selecting a cyclone using peak pressure alone. Indicated work, airflow, and the phase of combustion must be examined together.

The temperature rise has an additional implication. Improved oxidation may reduce soot and unburned hydrocarbons, but higher local temperature can increase thermal nitrogen-oxide formation. Emissions were outside the present simulation scope, so the hotter combustion field should not automatically be described as environmentally superior. A subsequent optimization should include nitrogen oxides, soot, carbon monoxide, and fuel consumption in a multi-objective framework. The best design for indicated work may not be the best design for regulated emissions.



**Figure 6.** Effects of GD 11560 on indicated mean effective pressure, maximum temperature, and maximum cylinder pressure.

TABLE 5.  
COMBUSTION RESPONSE OF GD 11560 RELATIVE TO BASELINE

| rpm  | IMEP   | Max. T | Max. P |
|------|--------|--------|--------|
| 1600 | +1.88% | +2.43% | +1.08% |
| 1800 | +0.36% | +1.94% | +0.46% |
| 2000 | -0.87% | +1.87% | +0.29% |

For marine application, the passive insert offers practical simplicity, but operational matching is essential. Small workboats and generator sets that spend substantial time near 1,600-1,800 rpm may benefit from the improved indicated output. An engine routinely operated near 2,000 rpm may require a shorter vane, smaller twist angle, bypass passage, or variable device to preserve flow capacity. The present results therefore support a calibrated cyclone rather than a universal one-size-fits-all insert.

#### *E. Mechanistic Interpretation and Comparison with Prior Work*

The simultaneous rise in turbulent kinetic energy and fall in inducted mass clarifies the mechanism more effectively than either result in isolation. The cyclone does not create energy; it redistributes the momentum of the incoming charge. Part of the axial kinetic energy is converted into tangential motion, and part is dissipated through separation, wall shear, and wake formation behind the vanes. Useful conversion increases the velocity fluctuations and organized rotation that support spray entrainment. Irrecoverable dissipation and blockage reduce the amount of air available for combustion. The optimum geometry is therefore the one that directs momentum into a persistent cylinder-scale structure with the smallest possible loss of flow capacity.

A simple screening ratio derived from the present data illustrates this point. Dividing the percentage turbulent-kinetic-energy gain by the absolute percentage air-mass loss gives approximately 2.21 for GD 11330, 2.33 for GD 11530, and 2.72 for GD 11560. This ratio is not proposed as a universal cyclone index because it ignores combustion phasing and pressure loss, but it helps explain why GD 11560 remained the preferred candidate despite having the largest mass penalty. Its additional turbulence gain was proportionally greater than the associated loss of charge mass. GD 05560, by contrast, would have a negative value because it consumed flow capacity while reducing turbulent kinetic energy.

The speed response can also be interpreted through the time available for flow development. A complete four-stroke cycle lasts about 75 ms at 1,600 rpm, 66.7 ms at 1,800 rpm, and 60 ms at 2,000 rpm. As speed rises, the cyclone must redirect a larger instantaneous volume flow within a shorter interval. The vane wake and tangential structure have less time to organize before the intake valve closes, while the pressure difference required to move the charge through the restricted passage grows. At 1,600 rpm, the longer time scale permits the rotational motion to support mixing without an overwhelming breathing penalty. At 2,000 rpm, restriction becomes more influential and the indicated-work benefit disappears.

This interpretation is consistent with intake-port and guide-vane research. Kim et al. found that port

arrangement and valve-seat flow direction strongly influence the swirl that survives into the cylinder [6]. Fan et al. showed that restricting a swirl-control valve increases swirl intensity but can sharply reduce the flow coefficient [7]. Saad, Bari, and Hossain demonstrated that guide vanes can improve in-cylinder motion for biodiesel operation, while their numerical work also emphasized the need to preserve adequate airflow [8], [9]. The current results extend that literature by screening a larger passive-cyclone matrix and linking the flow trade-off directly to validated indicated mean effective pressure.

The findings also complement previous cyclone studies on smaller engines. Numerical and experimental reports have shown that a turbo cyclone can alter manifold velocity and, under selected conditions, improve torque or power [12]-[16]. Those studies often focus on a limited number of devices or external engine outputs. The present intake-to-cylinder analysis shows why an apparently beneficial device may lose effectiveness as speed rises. The internal flow remains more energetic, and peak combustion quantities can still increase, yet the integrated indicated work may decline when the charge-mass deficit becomes dominant. This distinction helps reconcile mixed performance results in the earlier literature.

The B40 fuel representation strengthens the practical relevance of the mixing analysis. A biodiesel-containing blend generally has physical and chemical properties different from conventional petroleum diesel, and its heavier components can make evaporation and spray-air interaction more demanding. The cyclone-induced turbulence increases relative motion around the droplets and can shorten the distance over which rich pockets persist. Nevertheless, the present simulation used a two-component surrogate rather than a complete real-fuel composition. The predicted temperature, ignition behavior, and emissions chemistry should therefore be interpreted at the level of comparative engine behavior rather than exact molecular detail.

Peak temperature and pressure require similarly careful interpretation. Their increases indicate stronger or earlier heat release, but they do not by themselves prove an improvement in useful work. Indicated mean effective pressure integrates the pressure-volume loop over the cycle, so it is sensitive to combustion timing, expansion work, pumping loss, and the amount of trapped charge. A narrow increase in peak pressure can coexist with a smaller cycle integral. This is exactly what occurred at 2,000 rpm, where maximum pressure rose by 0.29% but indicated mean effective pressure fell by 0.87%. The result demonstrates the danger of optimizing a combustion device against a single peak value.

From a design perspective, the cyclone should be treated as an operating-range component. A fixed insert is most appropriate when the engine spends a predictable share of its duty cycle at speeds where mixing limits combustion more than flow capacity limits filling. For vessels with broader speed variation, a removable cartridge, bypass path, or variable-vane arrangement may be more appropriate. The cyclone could also be integrated with a manifold section that gradually recovers static pressure after the vanes, reducing

separation before the valve. Such changes would preserve the low-speed mixing advantage while mitigating the high-speed restriction observed here.

The study therefore suggests a staged engineering workflow. Candidate geometries should first be screened using turbulent kinetic energy, air mass, and pressure loss. The survivors should then be evaluated through full-cycle combustion for indicated work, heat-release

#### F. Limitations and Further Development

The numerical model was validated through indicated mean effective pressure, but direct optical or particle-image-velocimetry measurements of the flow field were not available. Turbulent kinetic energy and streamline structures should therefore be understood as model-supported interpretations rather than direct measurements. Mesh refinement was concentrated in critical zones, yet a formal grid-convergence index was not reported. Future work should quantify discretization uncertainty and compare more than one turbulence closure.

The cyclone position was fixed in the present geometry. Moving the device closer to the intake valve may preserve angular momentum, whereas moving it upstream may reduce local blockage and permit a more developed spiral. Additional loads, injection strategies, and transient operating points should also be tested. Finally, coupling the combustion model with a complete emissions mechanism and measuring fuel consumption would allow a multi-objective design that balances indicated work, volumetric efficiency, nitrogen oxides, soot, and installation practicality.

#### IV. CONCLUSION

The computational and experimental results demonstrate that a passive cyclone can substantially reorganize the intake flow of a naturally aspirated marine diesel engine. Among 27 candidates, GD 11560, with an 11 mm vane height, 50 mm vane length, and 60-degree twist, generated the strongest useful axial-tangential motion. It increased turbulent kinetic energy by 11.20%, extended the high-velocity and rotating flow deeper into the cylinder, and raised indicated mean effective pressure by 1.88% at 1,600 rpm and 0.36% at 1,800 rpm. The same geometry reduced inducted air mass by 4.12%, and this penalty became dominant at 2,000 rpm, where indicated mean effective pressure decreased by 0.87% despite small increases in maximum temperature and pressure. The main design implication is that cyclone effectiveness depends on engine speed and cannot be inferred from turbulence alone. A successful marine application must balance mixing enhancement against flow restriction and should be matched to the engine operating profile. The validated CFD model, with errors below 5%, provides a credible basis for that optimization, although future work should include mesh uncertainty, device position, wider load conditions, fuel consumption, and regulated emissions.

#### ACKNOWLEDGEMENTS

The authors acknowledge the Marine Power and Propulsion Laboratory, Department of Marine Engineering, Institut Teknologi Sepuluh Nopember, for access to the Yanmar TF-85 MH engine, measurement

phasing, and thermal loading. Finally, the most promising designs should be tested experimentally over the intended marine duty cycle with fuel consumption and emissions measurements. This sequence prevents a visually dramatic swirl field from being mistaken for an efficient engine solution and directs computational effort toward configurations with a defensible balance of fluid-dynamic and thermodynamic performance.

facilities, and experimental data used for numerical validation.

#### REFERENCES

- [1] J. B. Heywood, *Internal Combustion Engine Fundamentals*, 2nd ed. New York, NY, USA: McGraw-Hill Education, 2018.
- [2] S. Jafarmadar and M. Mansoury, "Exergy analysis of air injection at various loads in a natural aspirated direct injection diesel engine using multidimensional model," *Fuel*, vol. 154, pp. 123-131, 2015, doi: 10.1016/j.fuel.2015.03.026.
- [3] P. G. Hill and D. Zhang, "The effects of swirl and tumble on combustion in spark-ignition engines," *Prog. Energy Combust. Sci.*, vol. 20, no. 5, pp. 373-429, 1994.
- [4] H. Aghaali and H.-E. Angstrom, "A review of turbocompounding as a waste heat recovery system for internal combustion engines," *Renew. Sustain. Energy Rev.*, vol. 49, pp. 813-824, 2015, doi: 10.1016/j.rser.2015.04.144.
- [5] Varun, P. Singh, S. K. Tiwari, R. Singh, and N. Kumar, "Modification in combustion chamber geometry of CI engines for suitability of biodiesel: A review," *Renew. Sustain. Energy Rev.*, vol. 79, pp. 1016-1033, 2017, doi: 10.1016/j.rser.2017.05.116.
- [6] Y. Kim, Y. Han, and K. Lee, "A study on the effects of the intake port configurations on the swirl flow generated in a small D.I. diesel engine," *J. Therm. Sci.*, vol. 23, no. 3, pp. 297-306, 2014, doi: 10.1007/s11630-014-0710-x.
- [7] Y. Fan, D. Liu, and L. Wang, "3D numerical simulation on the variable swirl intake process of diesel engine," *Appl. Mech. Mater.*, vol. 273, pp. 143-147, 2013, doi: 10.4028/www.scientific.net/AMM.273.143.
- [8] I. Saad, S. Bari, and S. N. Hossain, "In-cylinder air flow characteristics generated by guide vane swirl and tumble device to improve air-fuel mixing in diesel engine using biodiesel," *Procedia Eng.*, vol. 56, pp. 363-368, 2013, doi: 10.1016/j.proeng.2013.03.133.
- [9] S. Bari and I. Saad, "CFD modelling of the effect of guide vane swirl and tumble device to generate better in-cylinder air flow in a CI engine fuelled by biodiesel," *Comput. Fluids*, vol. 84, pp. 262-269, 2013, doi: 10.1016/j.compfluid.2013.06.011.
- [10] M. F. Hamid, M. Y. Idroas, S. Saad, and M. K. Abdullah, "Numerical investigation of fluid flow and in-cylinder air flow characteristics for higher viscosity fuel applications," *Processes*, vol. 8, no. 4, Art. no. 439, 2020, doi: 10.3390/pr8040439.
- [11] Y. Z. Guo, H. L. Kui, C. S. Shao, Z. Z. Wang, and Y. Liu, "Flow characteristics of a dual-intake port diesel engine with guide vanes," *J. Appl. Fluid Mech.*, vol. 15, no. 3, pp. 709-722, 2022, doi: 10.47176/jafm.15.03.33207.
- [12] R. T. K. Raj and R. Manimaran, "Effect of swirl in a constant speed DI diesel engine using computational fluid dynamics," *CFD Lett.*, vol. 4, pp. 214-224, 2012.
- [13] Sarjito, S. P. Pratama, Wijianto, and Subroto, "Computational fluid dynamic analysis of turbo cyclone and intake manifold spacer on Honda Supra Fit," *JTTM J. Terap. Tek. Mesin*, vol. 3, no. 1, pp. 9-18, 2022, doi: 10.37373/jttm.v3i1.161.
- [14] D. F. Sinaga, Sanuri, and A. Zuhdi, "Pengaruh perubahan bentuk intake manifold terhadap unjuk kerja motor diesel dengan metode simulasi," *J. Tek. POMITS*, vol. 3, no. 1, pp. 30-33, 2014.
- [15] T. S. Utomo, "Simulasi efek turbo cyclone terhadap karakteristik aliran udara pada saluran udara suatu motor bakar menggunakan computational fluid dynamics," *Rotasi*, vol. 8, no. 2, pp. 30-36, 2012, doi: 10.14710/rotasi.8.2.30-36.
- [16] Y. R. Fauzi, "Pengaruh penambahan turbo cyclone aksial terhadap aliran dan performa motor bakar," *Turbo J. Program Studi Tek. Mesin*, vol. 7, no. 1, pp. 25-31, 2018, doi: 10.24127/trb.v7i1.679.
- [17] N. H. Anand, A. A. Hairuddin, K. A. M. Rezali, M. S. M. Alias, and A. N. Nizarimi, "CFD based parametric study of swirl intake

- manifold geometry and its effect on airflow characteristics on CI engines,” Eng. Res. Express, vol. 7, no. 3, Art. no. 035564, 2025, doi: 10.1088/2631-8695/ae0000.
- [18] P. A. Lakshminarayanan and A. Kumar, Design and Development of Heavy-Duty Diesel Engines. Singapore: Springer, 2020.
- [19] P. J. Pritchard and J. C. Leylegian, Fox and McDonald’s Introduction to Fluid Mechanics, 8th ed. Hoboken, NJ, USA: Wiley, 2011.
- [20] C. R. Ferguson and A. T. Kirkpatrick, Internal Combustion Engines: Applied Thermosciences, 3rd ed. Hoboken, NJ, USA: Wiley, 2015.
- [21] A. J. Martyr and M. A. Plint, Engine Testing, 3rd ed. 2007.