

1. Introduction

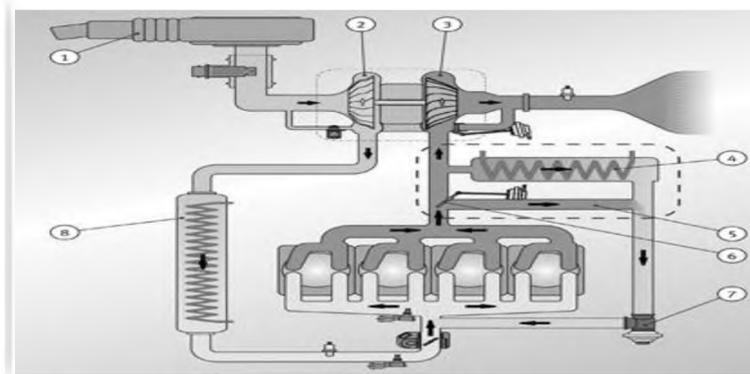
Turbo-supercharging is a process which is used to improve the performance of an engine by increasing the specific power output. In a conventional engine, supercharger functions as a compressor for the forced induction of the charge taking mechanical power from the engine crankshaft. The increased mass flow rate of air provides excess oxygen for complete combustion of fuel that would be available in a naturally aspirated engine. This allows more work to be done during the cycle thus increasing the overall power output of the engine. The general rule of thumb is that, not accounting for temperature-induced power losses, a turbo will increase horsepower by about 7 percent per pound of boost over a naturally aspirated configuration, and a supercharger will increase it by 5 or 6 percent per pound of boost. So for a boost pressure of 0.5 bar the power output of the engine can be increased to 150%.

Variable geometry turbine (VGT) has potential for improving part-load performance of the turbo charging system. It involves mechanical linkage to vary the angle of incidence of the turbine inlet guide vanes. The main problem is the fouling of the adjustable guide vanes by unburned fuel components and cylinder lubricating oil. Another drawback of VTG is that the extra mechanism adds to the cost of the turbochargers. Two-stage turbo charging is another concept which has often been mooted in the past when available turbochargers appear to be reaching their limits in efficiency and pressure ratio. Higher overall turbocharger efficiencies can be reached with two stages because it is possible to have inter-cooling between the two stages thereby reducing the compression work needed in second turbocharger stage. The major drawback of two-stage turbo charging is the complex arrangement of air and exhaust ducts [1]. In spite of several advantages, there are demerits of using turbo charging such as turbo lag, production cost, running cost, etc.

Exhaust Gas Recirculation (EGR) is a process which reduces the NO_x produced by engine because of supercharging. A widely adopted route to reduce NO_x by recirculating a controllable proportion of the engine's exhaust back into the intake air. Exhaust gas recirculation (EGR) is an effective method to reduce NO_x from diesel fuelled engines because it lowers the flame temperature and the oxygen concentration in the combustion chamber. Agarwal et al [2] conducted an experiment to investigate the effect of exhaust gas recirculation on the exhaust gas temperatures and exhaust opacity. It is seen that the exhaust gas temperatures reduce drastically by employing EGR. This indirectly shows the potential for reduction of NO_x emission. Thermal efficiency and brake specific fuel consumption are not affected significantly by EGR. However particulate matter emission in the exhaust increases, as evident from smoke opacity observations. Turbo charging along with EGR system is shown in Fig.1. The system becomes more complex and result in increased production and maintenance cost.

2. Scope of the Present Work

In this study, supercharging and exhaust gas recirculation for NO_x reduction are achieved using a jet compressor by re-circulating the exhaust gas. Jet compressor uses a jet of primary fluid to induce a peripheral secondary flow often against back pressure. Expansion of primary jet produces a partial vacuum near the secondary flow inlet creating a rapid re-pressurization of the mixed fluids followed by a diffuser to increase the pressure to the jet compressor exit value. In the jet compressor supercharging, the exhaust gas is used as the motive stream and the atmospheric air as the propelled stream. When high pressure motive stream from the engine exhaust is expanded in the nozzle, a low pressure is created at the nozzle exit. Due to this low pressure, atmospheric air is sucked into the expansion chamber of the compressor, where it is mixed and pressurized with the motive stream.



1. Air Filter
2. Turbocharger (Compressor)
3. Turbocharger (Turbine)
4. EGR Cooler
5. Bypass duct
6. Bypass flap
7. EGR Valve
8. Charge air cooler

Fig.1 Block diagram of the cooled exhaust gas recirculation system

The pressure of the mixed stream is further increased in the diverging section of the jet compressor. A percentage volume of the pressurized air mixture is then inducted back into the engine as supercharged air and the balance is let out as exhaust gas.

A back pressure valve is fixed to maintain the required boost pressure for the engine. Before inducting the gas mixture into the engine, it is filtered and cooled to the required inlet temperature of the engine. Thus, supercharged gas air mixture with required boost pressure and temperature is supplied to the engine which contains a maximum of 40% of exhaust gas. Combining the two processes not only saves the mechanical power required for supercharging but also dilutes the constituents of the engine exhaust gas thereby reducing the emission and the noise level generated from the engine exhaust. Further as there are no moving parts in jet compressor, production and maintenance costs are less when compared to conventional system. Fig.2 shows the schematic layout of an IC engine turbo-supercharger using a jet compressor.

3. Design of Jet Compressor

The geometrical design parameters of the jet compressor were obtained by solving the steady state Navier-Stokes equations as well as the equation of mass and energy transport for compressible flows.

$$\frac{\partial}{\partial x_i} \rho u_i = 0 \quad \dots\dots (1)$$

$$\frac{\partial}{\partial x_i} \rho u_i u_j = \rho g_i - \frac{\partial P}{\partial x_j} + \frac{\partial}{\partial x_i} (\tau_{ij} - \rho u_i u_j) \quad \dots\dots (2)$$

$$\frac{\partial}{\partial x_i} \rho C u_i T = \frac{\partial}{\partial x_i} (\lambda \frac{\partial T}{\partial x_i}) - \rho C u_i T + \mu \Phi \quad \dots\dots (3)$$

Using the theoretical design parameters of the jet compressor, a CFD analysis using the commercial software (FLUENT) was made to evaluate the performance of the jet compressor for the application of supercharging an IC engine. This evaluation turned out to be an efficient diagnostic tool for determining performance optimization and design of the jet compressor.

The jet compressor performance is mainly affected by turbulent mixing, energy consumption in the suction of the propelled stream and the friction losses. Optimizing nozzle geometry enhances the tangential shear interaction between the propelled and the motive fluids so that they completely mix inside the throat. However, experiments have shown that nozzle design doesn't influence much the overall performance of the jet compressor apart from affecting the motive fluid velocity. Care should be taken in the position of the nozzle which alters the turbulence mixing and indirectly affects the entrainment ratio. Throat length and diameter also contribute much to the performance of the jet compressor.

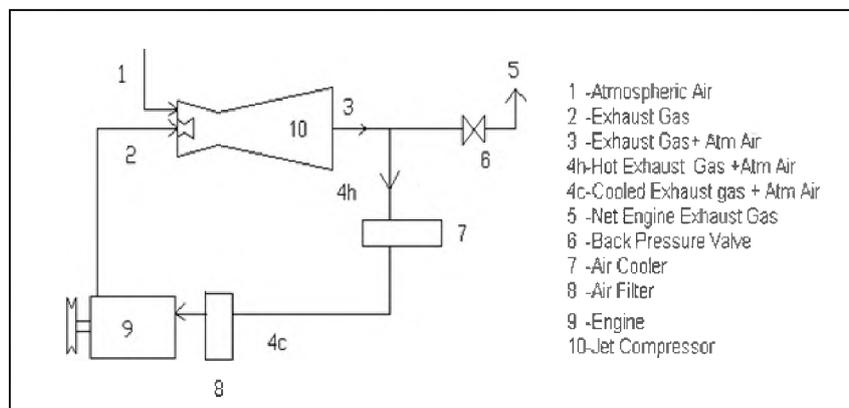


Fig.2 Schematic view of a turbo-supercharger using jet compressor with forced draft.

The throat should be sufficiently long to develop a uniform velocity before the flow enters the diffuser section thus reducing the energy losses with better pressure recovery [3]. Optimal throat diameter affects the entrainment ratio that is achievable [4]. Smaller throat diameter creates a huge change in the entrainment ratio by choking whereas a larger diameter makes the flow leak back into the system. Divergence angle and the length also contribute much to the performance of the jet compressor. Even though larger divergent length favours the pressure recovery, the optimum recommended length is twice the throat diameter. In 1951, Holton [5] showed that the entrainment ratio is a function of molecular weight of the fluid and the operating temperature but independent of pressure and the jet compressor design.

To enhance the jet compressor performance, understanding the flow field mechanism inside the jet compressor is much useful. The flow velocity distribution indicates the degree of mixing between the motive and the propelled stream and the quantity of entrained fluid. When the motive stream velocity exceeds the speed of the sound, shock waves are created inside the compressor. The shock waves convert the velocity into pressure but in an inefficient manner. Apart from this the shock waves interact with the boundary layer formed along the jet compressor wall exposing the flow to a strong inviscid-viscous interaction limiting the exit or the discharge pressure. This reduces the maximum pressure lift ratio and the jet compressor performance significantly. To overcome this problem Constant Rate of Momentum Change (CRMC) method proposed by Eames, 2002 [6] was used. This method eliminates the shock waves created at the diffuser by allowing the momentum of the flow to change at a constant rate as it passes through the diffuser passage by gradually raising the static pressure from entry to exit, thus avoiding the total pressure loss due to shock waves encountered in the conventional diffusers. The CRMC method based jet compressor gave a remarkable improvement in the entrainment ratio and the pressure lift ratio. Figure.3 shows the flow chart for designing of jet compressor using CRMC. The procedure to find various

geometrical design parameters of jet compressors is given in Appendix-A. Figure.4 shows the comparison between the diffuser shapes of conventional and CRMC method jet compressor.

4. Numerical Analysis of Jet Compressor

The flow field inside the jet compressor before entering the supercharger has been simulated using FLUENT software. The simulated results have helped in understanding the local interactions between the two fluids, and recompression rate which in turn resulted in a more reliable and accurate geometric design and operating conditions of the jet compressor. Many numerical studies about supersonic ejectors have been reported since 1990's in predicting ejector performance and providing a better understanding of the flow and mixing processes within the ejector [7-10], pump [11] and in mixing processes [12]. Simulations were carried out with structured quadrilateral mesh of size 0.25 mm, and a converged solution was obtained. Table.1 shows the details of the flow domain meshing and Fig. 5 shows the meshed geometry of the 2D jet compressor. The jet compressor developed using gambit consists of a primary nozzle, secondary nozzle, diffuser and a storage chamber. Table.2 describes the various parameters used for simulation in FLUENT (*CFD* modeling). From CFD analysis the flow analysis such as velocity of flow (Fig.6), static pressure inside jet compressor (Fig.7) and static pressure raise along the axis of jet compressor (Fig.8) was studied. The effect of varying the input and output properties of jet compressor was studied in detail. Effect of diffuser pressure on the performance of jet compressor is given in Fig. 9. The figure shows the variation of ratio of actual to the designed diffuser pressure with entrainment ratio, where the entrainment ratio is found to be constant for a lower pressure ratio and then decreases for higher pressure ratios. This could be due to the energy loss during the mixing of primary and secondary fluids.

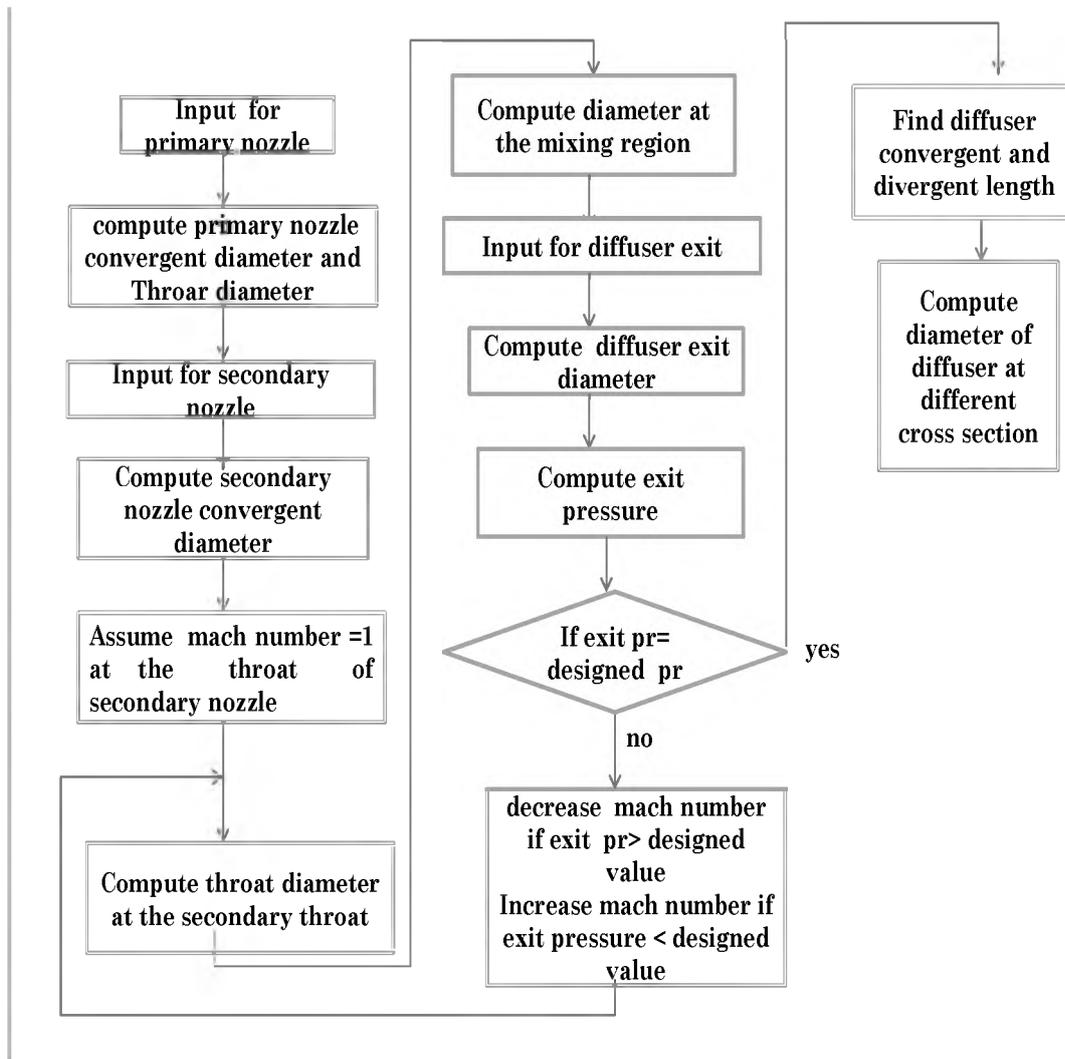


Fig. 3 Flow chart for design of jet compressor using CRMC method

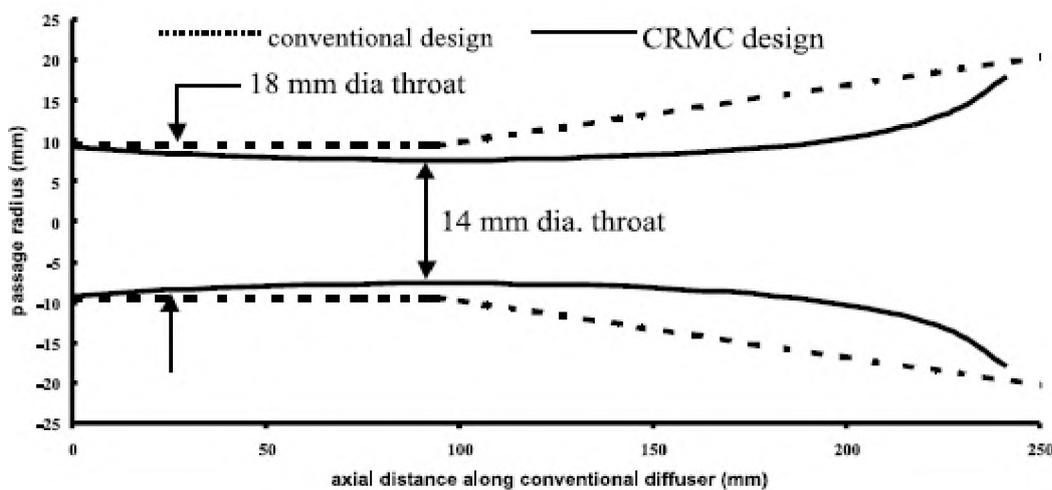


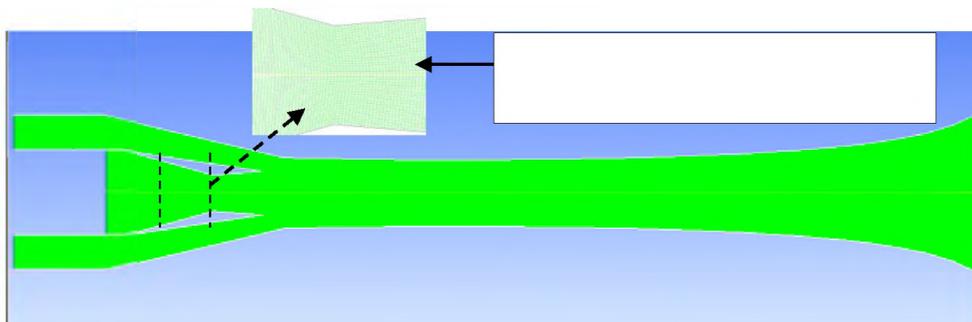
Fig. 4 Comparison of jet compressor diffuser profile between conventional and CRMC design

Table. 1 Details of the flow domain meshing

Type of meshing	Elements of meshing	Interval size	No. of zones	No. of cells	No. of nodes
Structured Map	Quadrilateral	0.25	9	54314	55367

Table.2 Various parameters used for simulation in FLUENT(*CFD modeling*)

Model type	Two-dimensional axi-symmetric model
Numerical solver	Conventional equation (segregated solver)
Turbulence model	Standard $k-\varepsilon$ model
Discretization technique	Finite volume
Discretization scheme	
Pressure	Standard scheme
Pressure-velocity coupling	SIMPLE
Boundary condition	
Propelled-stream inlet	Inlet mass flow rate
Motive-stream inlet	Inlet pressure
Diffuser exit	Pressure outlet



*The insert shows the uniform type of quadrilateral structured mesh used to mesh the jet compressor

Fig.5 Meshed model of the jet compressor.

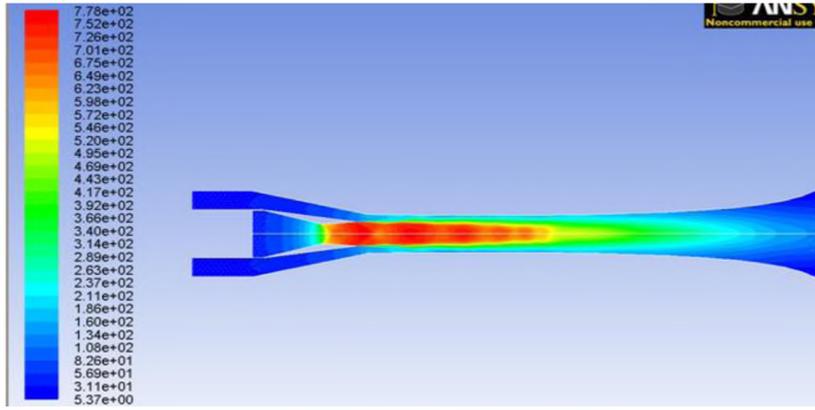


Fig 6 Velocity of flow inside jet compressor

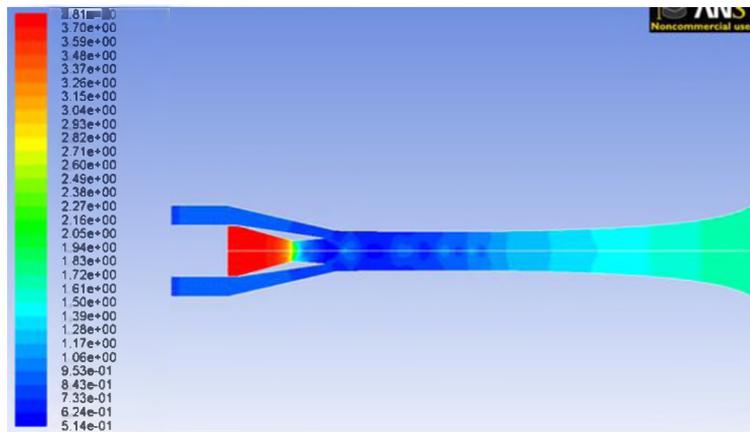


Fig 7 Static pressure inside jet compressor

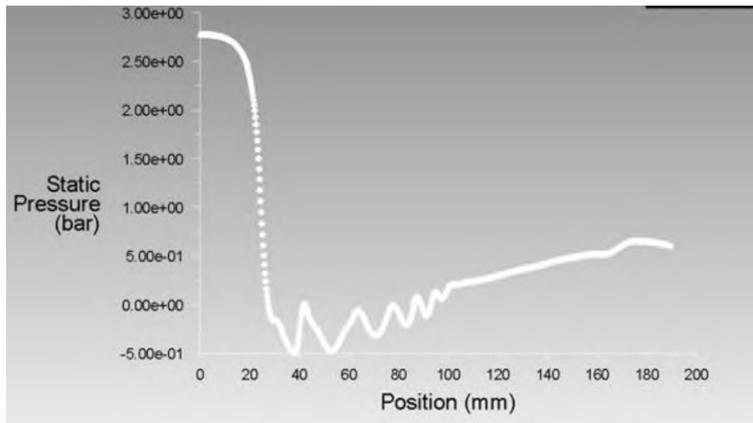


Fig 8 Static pressure along axis of jet compressor

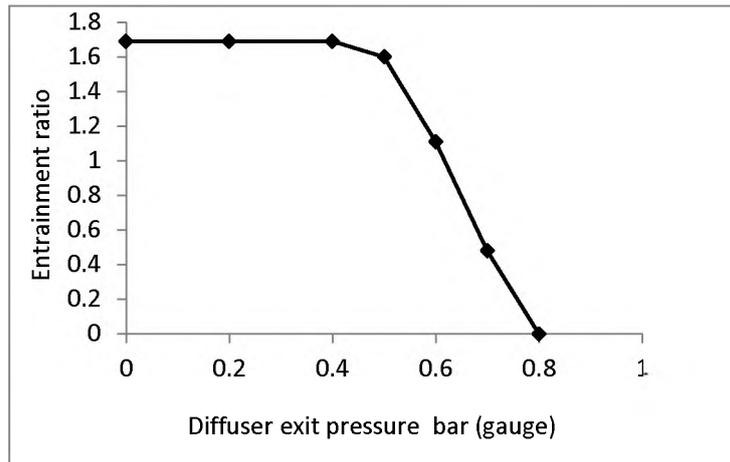
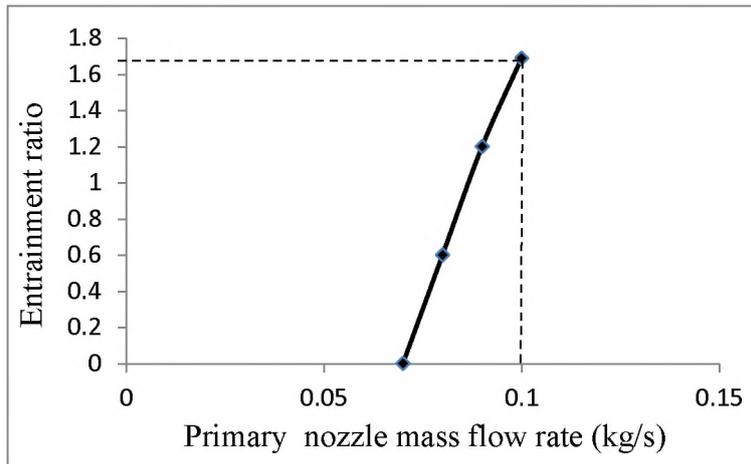


Fig. 9 Effect of diffuser exit pressure on the entrainment ratio of jet compressor

A fluent simulation study was made on a jet compressor designed for the conditions given in Table.3. Figure.10 shows the effect of primary fluid mass flow rate on the entrainment ratio of the jet compressor. It shows that, the entrainment ratio varies linearly with the mass flow rate and below 0.07kg/s, the entrainment ratio is zero which results in reverse flow. The same trend is observed for primary fluid pressure, temperature on the entrainment ratio of the jet compressor and they are shown in Figs.11&12 respectively. The above results indicate that the entrainment ratio of a jet compressor depends on the operating conditions given in Table.3 and varies when the engine operating conditions were changed.

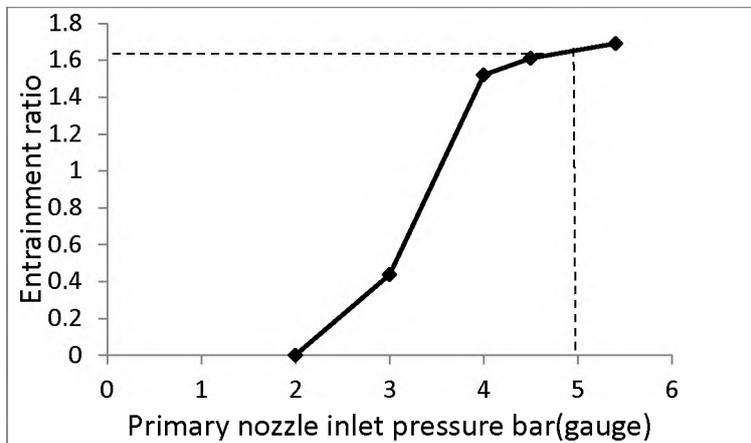
Table.3 Operating conditions for design of jet compressor

Primary nozzle inlet	Secondary nozzle	diffuser
Pressure =5 bar(abs)	Pressure =1 bar(abs)	Pressure= 1.5 bar(abs)
Temperature= 1300 K	Temperature= 300 K	Mass flow rate=0.266kg/s
Mass flow rate =0.1kg/s	Mass flow rate=0.166kg/s	



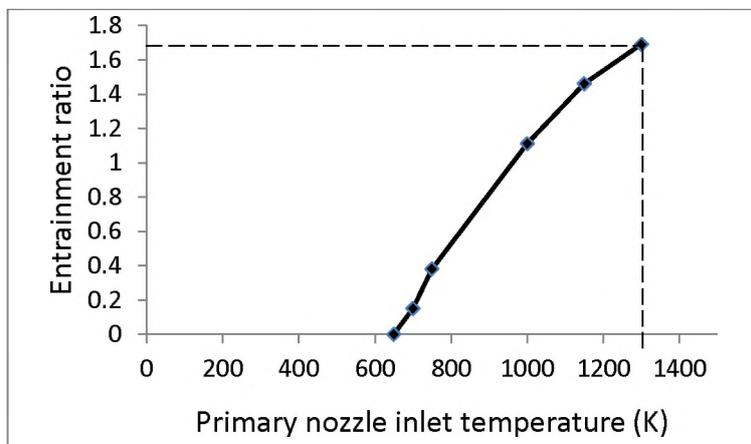
Note: *The dashed line gives the maximum entrainment ratio when the engine was operated at the design primary nozzle mass flow rate of 0.1 kg/s.

Fig.10 Effect of primary nozzle mass flow rate on the effect of entrainment ratio



Note: *The dashed line gives the maximum entrainment ratio when the engine was operated at the design inlet pressure of 5 bar(gauge).

Fig.11 Effect of primary nozzle inlet pressure on jet compressor for a fixed diffuser outlet Pressure



Note: *The dashed line gives the maximum entrainment ratio when the engine was operated at the design temperature of 1300 K.

Fig. 12 Effect of primary nozzle inlet temperature on the entrainment ratio.

5. Design of Jet Compressor for Supercharging

The geometrical parameters of the exhaust gas driven Jet compressor are designed for engine's maximum power output. Exhaust gas mass flow rate and pressure are maximum for that condition. Engine exhaust gas back pressure affects the performance of the engine. Exhaust gas pressure should not exceed atmospheric pressure as it degrades scavenging process of the engine. Hence pressure in primary nozzle that handles exhaust gas should be below atmospheric pressure. Exhaust gas entering at atmospheric pressure should leave the primary nozzle at a pressure less than atmospheric pressure to entrain secondary fluid (atmospheric air). So a divergent primary nozzle is used for this purpose. The grid view of the jet compressor connected to engine exhaust manifold is shown in Fig.13.

The input parameters for the jet compressor are exhaust gas mass flow rate, pressure and temperature and output parameters of the jet compressor are diffuser outlet pressure(boost pressure) and entrainment ratio (%EGR) . These parameters are not constant and varies depending upon the engine speed and power output. The designed jet compressor should be able to match with the supercharging requirement for all engine speeds and power outputs.

Simulation studies on the jet compressor have to be carried out to find the feasibility of jet compressor as super charger for a wide range of engine operation. The input parameters of the jet compressors for different loads of the engine are to be determined. For this purpose a computer code is developed to study the various engine parameters of super charged diesel engine [13]. The code is written based on two zone model with gas exchange and heat transfer process. In two zone model, the cylinder has two zones, one the unburned zone and the other burned zone of the working fluid. These zones are actually two distinct thermodynamic systems with energy and mass interactions between themselves and their common surroundings, the cylinder walls. The

mass-burning rate (or the cylinder pressure), as a function of crank angle, is then numerically computed by solving the energy balance equation obtained from applying the first law to the two zones separately. Further friction is taken into account using empirical relations. For a given engine specifications, the shaft power, exhaust gas pressure, exhaust gas temperature and gas mass flow rate are determined using the code. The code was also run to study the variation of input parameters viz. boost pressure, %EGR, air fuel ratio, engine speed. The results obtained from the code were validated with the results obtained by conducting the performance test on the engine under normal operation conditions.

6. Results and Discussion

A simulation study of jet compressor fitted with diesel engine is carried out to study the EGR requirement for various power output. The natural aspirated engine is chosen for study and its specifications are given in Table. 4. The overall power output of the engine is 24 kW with 8 kW output per cylinder. Simulation is carried out on a single cylinder diesel engine. Due to supercharging, maximum power output of the engine has increased from 8kW to 12kW and boost pressure is increased to 0.5 bar (gauge).

Primary nozzle is connected to the engine cylinder using exhaust manifold. Since the simulation is carried for a multi cylinder engine the average mass flow rate per cylinder of the exhaust gas is considered as the mass flow rate of primary fluid. Using engine simulation code the exhaust gas properties just before the exhaust valve opening are determined, which will be input parameters for simulating the performance of jet compressor. Theoretical performance analysis of jet compressor is done and a performance map of jet compressor for wide range of operation is drawn. Figure.14 shows the flow chart to create performance map of jet compressor using computer engine simulation code and fluent. A performance map of jet compressor is shown in

Fig.15. It shows that the variation of engine power output for the percentage of EGR admitted is different for different boost pressures. This implies that the power output is a function of both the percentage of exhaust gas circulated and the boost pressure developed inside the engine. For low range of boost pressures 0.9 to 1.1 bar, the power output increases with increase in percentage EGR. However, for boost pressures greater than 1.1 bar, the engine power output increases with decrease in percentage EGR. This is due to the fact that at low boost pressure the jet compressor was operated in off-design conditions. For a given percentage of exhaust gas re-circulation, the optimum boost pressure and the maximum power output can be determined from the performance map.

Table.4 Engine specifications for experimental setup

Engine make	Kirlosker H394 (air cooled) naturally aspirated diesel engine
No of cylinders	3
Swept volume	2900 cc
Max power and Speed	23.5kW and 1500 rpm

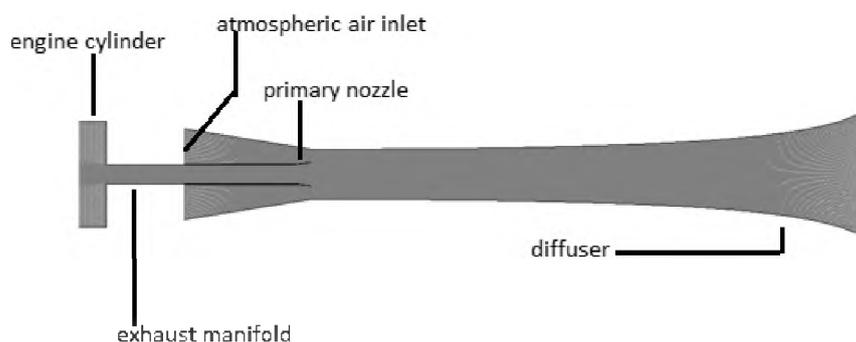


Fig. 13 Grid view of the jet compressor connected to engine exhaust manifold

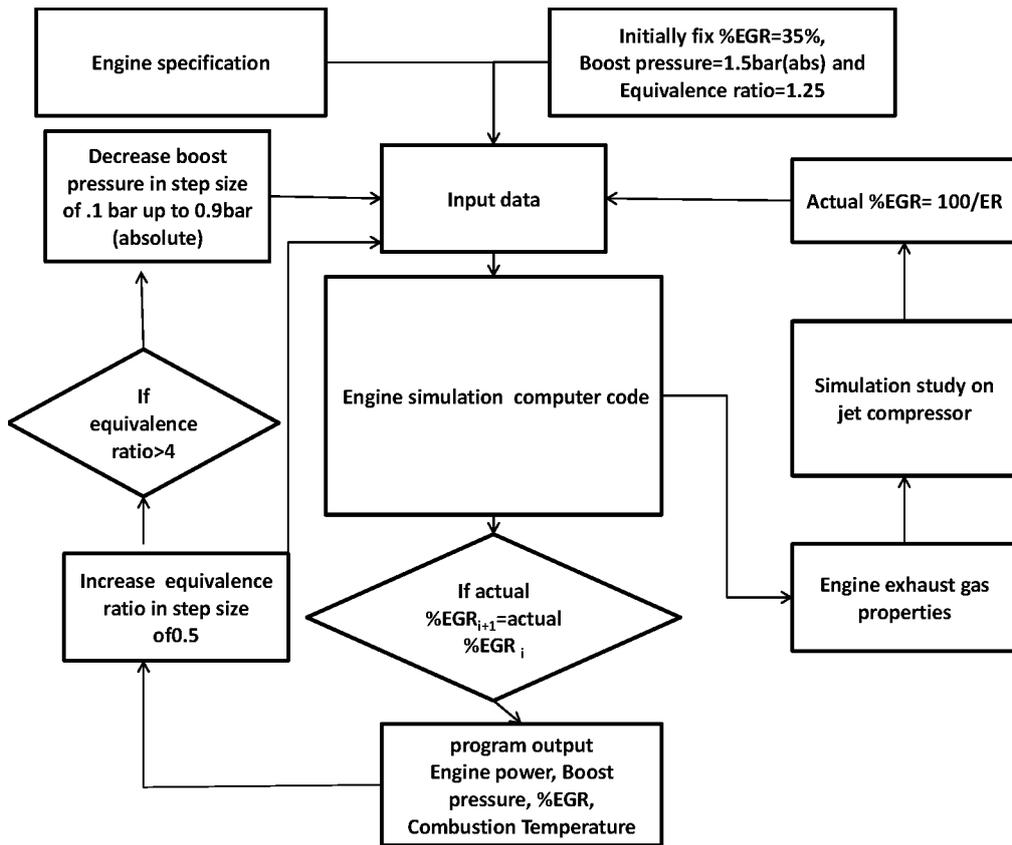


Fig. 14. Flow chart to create performance map of jet compressor using computer engine simulation code and FLUENT

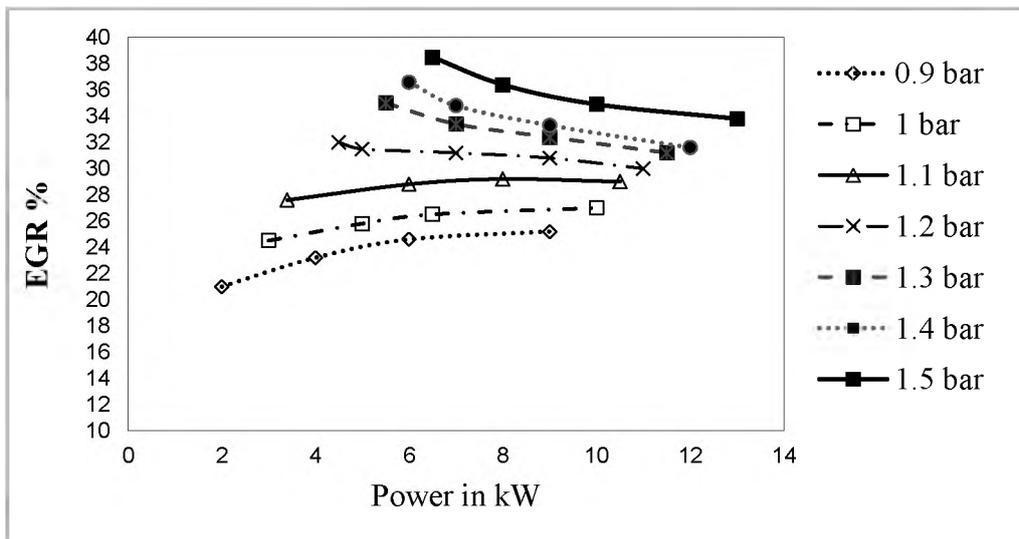


Fig. 15 Performance map of jet compressor for different boost pressure and engine power output



Fig.16 Experimental setup of EGR run jet compressor fitted to a diesel engine.

A performance test was conducted on the engine fitted with the jet compressor (Fig.16) to compare the results with that of theoretical values obtained for different boost pressure and percentage of exhaust gas re-circulation. To conduct the test for different boost pressures, the engine was fitted with a back pressure butterfly valve at the outlet of the jet compressor diffuser section. The engine is loaded using an electrical resistance loading system. Using an orifice meter, the mass flow rate of the atmospheric air sucked in the secondary nozzle of the jet compressor was measured. The engine was run at full load of 35 kW with the absolute boost pressure set at 1.5 bar by adjusting the back pressure butterfly valve. At this condition the mass flow rates of the primary and secondary fluids were measured. From the measured mass flow rates, the entrainment ratio of the jet compressor was determined at full load condition.

The experiment is repeated for every reduction of 5kW load and their corresponding entrainment ratios of jet compressor were calculated. Using the determined entrainment ratios, the percentage of EGR admitted to the engine was determined. The entire procedure of the test was repeated by changing the boost pressure at the diffuser section of the jet compressor in terms of 0.1 bar. The determined percentage EGR for different loads and boost pressures were plotted to get the

performance map of the given three cylinder diesel engine fitted with the EGR run jet compressor.

Figure.17 shows the comparison of percentage of EGR as a function of engine power obtained from experiment and simulation for different boost pressures.

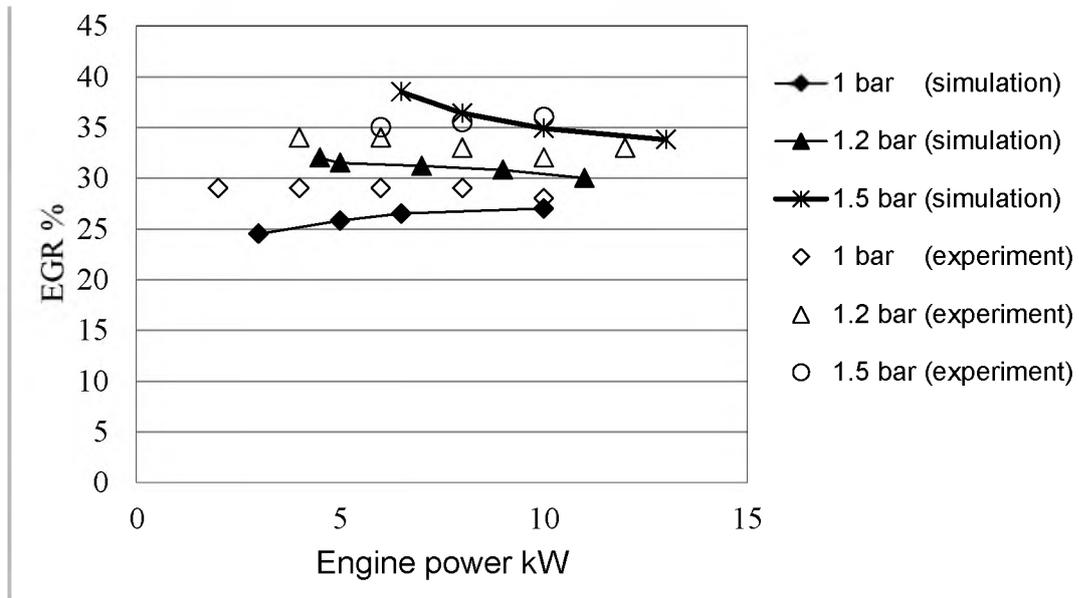


Fig.17. Comparison of simulated and experimental results of jet compressor supercharging.

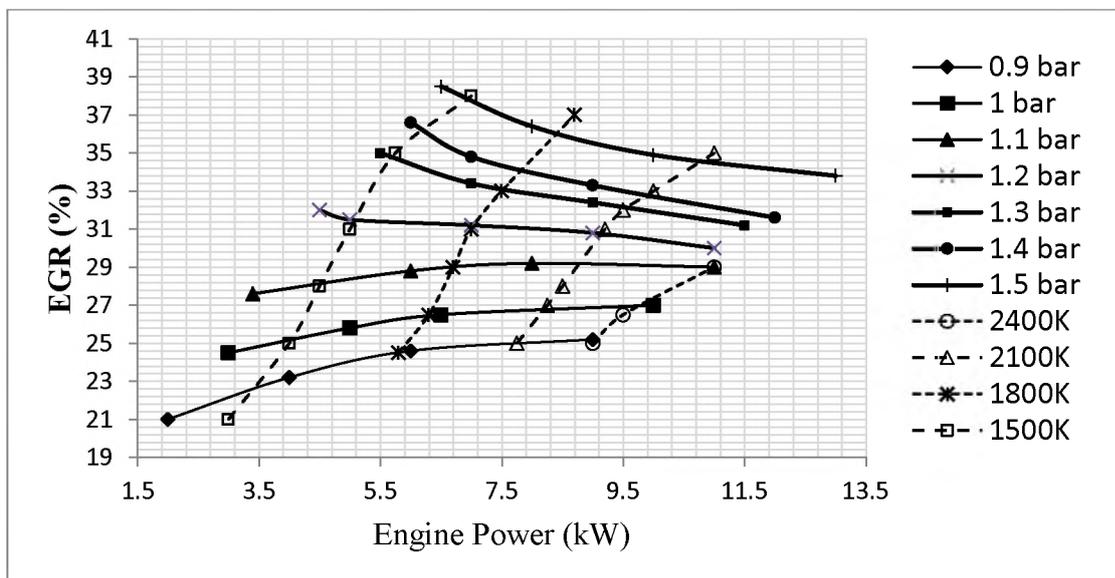


Fig.18 Performance map along with combustion temperature inside the cylinder

Emissions of NO_x from combustion are primarily in the form of NO. According to the Zeldovich equation, the generation of NO is limited based on the availability of oxygen present in air and the operating temperatures. At temperatures below 1300°C, the concentration of NO generated is low compared to the higher concentration (about 200,000 ppm) generated above 2,300°C [14]. This shows that NO_x emission from the engine is mostly controlled by the engine operating temperature. Experiments were conducted by fixing boost pressure with minimum combustion chamber temperature for different loads with and without exhaust gas re-circulation and the results obtained are shown in Fig .19. The measured level of NO_x was found to decrease much compared to a naturally aspirated engine without EGR. The reduction in the NO_x level is due to the percentage of exhaust gas used in the jet compressor to increase the engine power output.

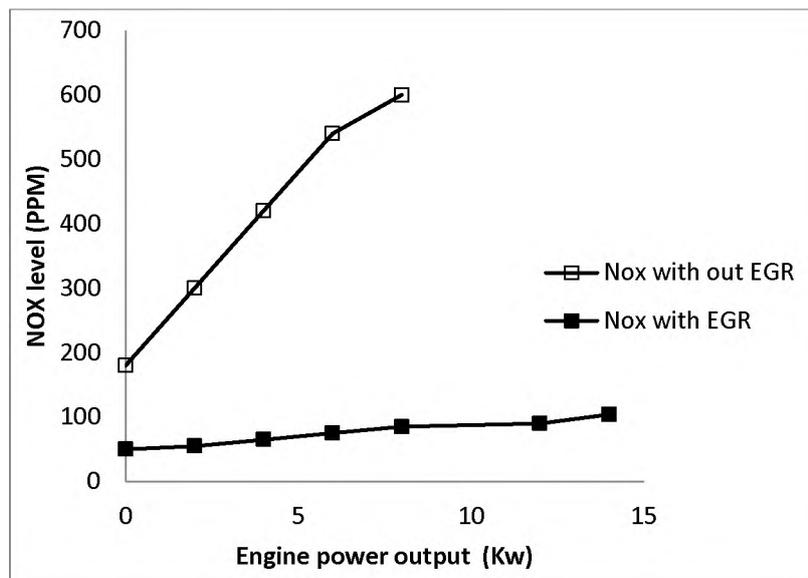


Fig.19 Comparison of NO_x level obtained from experiment with and without EGR.

7. Conclusions

In this study, a novel method of supercharging a diesel engine using the exhaust gas assisted jet compressor was analyzed both numerically and experimentally. The specifications of the jet compressor were determined using the constant momentum method. The engine operating

conditions were optimized using the available standard code for a given engine specifications. From the optimized engine operating conditions, the input parameters for the jet compressor were fixed. The performance of the jet compressor was then analyzed both using the commercially available software Fluent as well as experimentally. The results were compared and found to be closely matching. A performance map was drawn using which the optimum boost pressure and maximum entrainment ratio could be obtained for a given percentage of exhaust gas recirculation and power output. studies are also be made on thermodynamic aspect to improve the performance of the jet compressor used for supercharging as well as to reduce the NOx emission.

Nomenclature

τ_{ij}	symmetric stress tensor
$\rho u_i u_j$	Reynolds stress,
$\rho C_p u_i T$	turbulent heat flux
$\mu \Phi$	viscous dissipation A area(m^2)
C_p	specific heat capacity at constant pressure($J\ kg^{-1}K^{-1}$)
η	efficiency
γ	ratio of specific heat
c	velocity(ms^{-1})
D	diameter,(m)
F	force(N)
L	length (m)
L_D	length of diffuser
m	mass flow ($kgms^{-1}$)
M_a	Mach number
M_o	momentum ($kg\ m\ s^{-1}$)
P	static pressure (Pa)
R	individual gas constant ($J\ kg^{-1}K^{-1}$)
R_m	m_g/m_g entrainment ratio
T	static temperature (K)
T_O	total or stagnation temperature (K)
x	axial distance from diffuser entry

ρ	density (kgm^{-3})
θ	diffuser half-angle(deg)
β	constant
NO_x	oxides of Nitrogen

Subscripts

:	1	plane at entry to diffuser section
	D	diffuser
	DE	diffuser exit plane
	g	exhaust gas or primary flow condition
	NE	primary nozzle exit
	o	total or stagnation condition
	s	secondary flow stream
	x	co-ordinate along central axial of jet compressor

superscripts

*	Refers to critical condition of diffuser throat ($\text{Ma}=1$)
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LIST OF PUBLICATIONS OUT OF THIS RESEARCH WORK

I. PUBLISHED IN INTERNATIONAL / NATIONAL JOURNALS / CONFERENCES :

(a) International Journal :

1. "IC Engine Supercharging and Exhaust Gas Recirculation using Jet Compressor", **A. Kalaiselvane**, N.Alagumurthy, K.Palaniradja, G.S.Gunasegarane. *International journal of Thermal Science* 2010 Volume 14, Issue 4, Pages: 1027-1037.

(b) International Conferences

1. "Determination of Optimized Jet Compressor Parameters using DOE", **A.Kalaiselvane**, N.Alagumurthy, K.Pajaniradja, *International conference on challenges and applications of mathematics in science and technology*, National Institute of Technology, Rourkela, Jan 2010.
2. "Optimization of jet compressor parameters using DOE" **A.Kalaiselvane**, N.Alagumurthy, *International Conference on Advances in Industrial Engineering Applications*, Anna University, Jan 2010.

(c) National Conferences

1. "IC Engine Super Charging using Exhaust Gas Assisted Jet Compressor", **A.Kalaiselvane**, N.Alagumurthy, K.Pajaniradja, S.Sangeeth, *National conference on low carbon technologies in automobile*, Annamalai university, Feb 2009.

II. COMMUNICATED TO INTERNATIONAL JOURNALS :

1. "Performance improvement of jet compressor using forced draft system", **Kalaiselvane Adhimoulame**, Alagumurthy Natarajan, Palaniradja Krishnaraj, Gunasegarane Selvaraj G. *International journal of Aerospace* (under review).
2. "Experimental investigation on diesel engine supercharging and exhaust gas recirculation (EGR) using exhaust gas assisted jet compressor" ,**A. Kalaiselvane**, G. S. Gunasegarane, N. Alagumurthy, K.Palaniradja, *International journal of gas turbines and power* (ASME) (under review).