INTAKE VALVE MODELLING AND STUDY OF THE SUCTION AIR PRESSURE AND VOLUMETRIC EFFICIENCY IN A FOUR STROKE INTERNAL COMBUSTION ENGINE

Mohammad Syed Ali Molla[1*], Mohd Sapuan Salit, Mohd Megat Hamdan Bin Megat Ahmed, Fuad Abas[2] and Waqar Asrar[3]

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Abstract

The air pressure inside a cylinder on the suction stroke is the measure of volumetric efficiency and air quantity drawn on the suction stroke of the engine. Intake air pressures inside a cylinder have been computed in each degree of the suction stroke and exhaust stroke with seven different intake valve diameters from 14 mm to 32 mm, with increments of 3 mm in diameter in a four stroke internal combustion engine. The results of the computational work show that cylinder pressure on the suction stroke starts decreasing from the suction valve full opening timing at 200^o crank angle after the top dead centre (TDC at 180^o) until the middle of the suction stroke at 255^o crank angle for all the intake valve diameters, and again cylinder pressure increases until the suction valve starts closing at crank angle 340^o. The effects of the intake valve diameter on cylinder pressure on the suction stroke are also found comparable between the suction valve opening and closing period. The increase of intake valve diameter provides a significant decrease of cylinder pressure until middle of the suction stroke at crank angle 255^o, when the piston moves downward, but these pressure differences again reduces until the suction valve is completely closed. The air pressures inside the cylinder on the suction stroke with different intake valves become nearly the same when the suction valves completely close. Thus the necessity for increasing the intake valve diameter is not an important factor to increase volumetric efficiency, although the manufacturers are at present considering to increase the number of intake valves as in a 12 or 16 valve engine instead of an 8 valve engine in a four cylinder SI engine.

Keywords: Intake valve, suction air pressure and volumetric efficiency

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[1] *Department of Mechanical Engineering, Khulna University of Engineering and Technology (KUET), Khulna - 920300, Khulna, Bangladesh. Email mmsali03@yahoo.com & pmsali@me.kuet.ac.bd*

[2] *Department of Mechanical and Manufacturing Engineering, University Putra Malaysia, 43400, Serdang, Selangor, Malaysia*

[3] *Faculty of Engineering, International Islamic University, Malaysia Jalan Gombak, 53100 Kuala Lumpur, Malaysia*

[4] *Corresponding author*

Introduction

In a naturally aspirated engine, air flows from the atmosphere to the engine cylinder due to the negative pressure inside the cylinder on the suction stroke when the piston moves downwards. Though this negative pressure helps in drawing air from the atmosphere inside the cylinder on the suction stroke, it lowers volumetric efficiency of the cylinder. If negative pressure prevails inside a cylinder on the suction stroke after the suction valve closes, the quantity of air entering inside the cylinder is less than the theoretical air quantity that could be reserved inside the cylinder at the atmospheric pressure. This negative pressure can affect the volumetric efficiency of the engine.

Intake valve can be modelled in different ways to investigate fluid flow behavior and air fuel mixing. In some engine, intake air valve is slanted on engine head so that intake air may enter in the combustion chamber tangentially to increase swirl in suction stroke and for better air fuel mixing. In this model the clearance volume and swept volume of engine cylinder were kept constant and only intake valve diameter was increased to increase the flow area to investigate its effects on suction air property.

Air and fuel mixing on the suction stroke of a SI engine depends on the physical and chemical properties of the fuel and on the flow property of the air and fuel. Air fuel mixing characteristic depends on air velocity and degree of turbulence.

Several investigators computed the flow property of air in different situations along with the experimental work.

Nicolao *et al.* (1996) modelled the volumetric efficiency (η_v) represents a measure of the effectiveness of an air pumping system, and is one of the most commonly used parameters in the characterization and control of four stroke internal combustion engines. Physical models of volumetric efficiency η_v require the knowledge of some quantities usually not available in normal operating conditions. Hence, a purely black-box approach is often used to determine the dependence of η_v upon the main engine variables, like the crankshaft speed and the intake manifold pressure. Various black-box approaches for the estimation of η_v are reviewed, from parametric (polynomial-type) models, to non-parametric and neural techniques, like additive models, radial basis function neural networks and multi-layer perceptions. The benefits and limitations of these approaches have been examined and compared. The problem considered there can be viewed as a realistic benchmark for different estimation techniques.

Arslan *et al.* (2001) studied the effects of valve timing on volumetric efficiency of an IC engine. In the study, the effect of valve timing, which is a dynamic parameter, has been investigated. Beside the original one, thirteen new camshafts have been produced. New camshafts are then produced by shifting the valve-open period. All camshafts are used to determine volumetric efficiency.

Hatamura, *et al.* (2001) optimized intake air throat diameter to maintain volumetric efficiency in the high-speed range and maximize intake airflow velocity in the low- and mid-speed ranges, thus enhancing volumetric efficiency in all speed ranges with the output performance in high-speed range being ensured. The valve angle was then narrowed for optimization and the combustion chamber made more compact (reduction of surface/volumetric ratio) to reduce cooling loss for higher thermal efficiency, without relinquishing adequate throat diameter. Squish area, which creates turbulence of the air/fuel mixture (squish) during the compression stroke, was given around the valves to ensure high volumetric efficiency and high combustion speed. As a result, the combustion chamber was a compact pentroof with intake and exhaust valve angles of 27 degrees. Squish area to bore area ratio was 17.3 percent and the squish clearance was 0.68 mm. The throat diameter was 28.5 mm on the intake side; and 25 mm on the exhaust side.

In designing the intake-port of an internal combustion (IC) engine, its curvature ratio is one of the important parameters. Thus, curved duct flows (U-bend flows) of Chang *et al.* (1983) and

Cheah *et al.*(1994) are considered as fundamental test cases prior to computations of the IC engine port-cylinder flows of Kawazoe (1993).

Celik *et al.* (2000) investigated the preliminary results of some of a few kinds of effort in large eddy simulation (LES) of engine flows to predict turbulent fluctuations, and the statistics of turbulence quantities inside IC engine cylinders. The computed velocity fluctuations, correlation coefficients and energy spectra of turbulent fluctuations were compared to experimental results. The predictions seemed to extend well into the inertial range of turbulence and depicted a good qualitative agreement with measurements. The results also showed light into the mechanisms by which turbulence may be generated by the piston-bowl assembly.

A measurement technique, like laser Doppler Anemometry, provides instantaneous data, for a generic variable like velocity. These data, in the spatial location at the crank angle may be represented as the sum of a deterministic unsteady ensemble-averaged component and random unsteady fluctuations component about the ensemble-averaged values, Boretti and Nebuloni (1993). Turbulent parameters computed by Boretti and Nebuloni, (1993) with the models for the small-scale fluctuations used in computer codes for engine flow studies applications for both gasoline and diesel engines and made comparison with available experimental data .

Sher, *et al.* (1988) presented an experimental study of the flow pattern inside a model cylinder of uniflow-scavenged two-stroke engine. The velocity fields as well as the turbulent parameters have been mapped under steady-flow conditions with the aid of a hot wire anemometry technique.

Trigui *et al.* (1999) presented a methodology, which combined, steady state CFD techniques with robust numerical optimization tools to design, rather than just to evaluate the performance of IC engine ports and chambers. Haworth and Jansen (2000) reported the advance in physical models, numerical methods, and computational power together have brought large-eddy simulation (LES) to the point where it warrants serious consideration for computing in-cylinder turbulent flows. Gosman *et al.*(1990) used the SPEED_HC code for in-cylinder flow simulations during compression and expansion strokes for four valves and five valves gasoline engines. This code solves the three-dimensional, unsteady, density weighted averaged, Eulerian conservation equations for mass, momentum, energy and species. Reitz and Diwakar (1986) involved conservation equations for k and ε using the previously defined eddy-viscosity formulations. Zhao and Ladommatos (1998) reviewed the optical diagnostics for in-cylinder mixture formation measurements in IC engine. Chant *et al.* (1998) reported that reverse engineering is used to determine the inlet port geometry of a diesel engine. Choi and Guezennec (1998) studied the measurements of the instantaneous in-cylinder flow fields in a water analog engine simulation rig by Particle Tracking Velocimetry (PTV). Shudo (2000) made series of research analysis on the thermal efficiency of an internal combustion engine fuelled with hydrogen, especially influence of the degree of constant volume and the cooling loss. Lumley (2001) described the early work of Ricardo, in which squish, is used in flat-head engines to generate turbulence levels comparable to those in overhead-valve engines, leading to rapid flame propagation, and suppressing knock. Suga *et al.* (1995) has reported from Heat Transfer Laboratory of Toyota Central Research and Development Laboratory that the three-equation cubic k-ε–A_2 model proposed by Craft *et al.* (1997) has been evaluated turbulent flows pertinent to engineering applications, especially in the automobile industry. The k-ε–A_2 cubic EVM of Craft *et al.* (1997) may be suitable for complicated industrial flows, since it is one of the rare models totally free from the topographical parameters, though it requires to solve the third transport equation for A2 which is the second invariant of the anisotropic Reynolds stress tensor. In fact, the k-ε–A_2 model showed encouraging validation results Craft *et al.* (1997). However, all of the flows tested were essentially two-dimensional.

Yang *et al.* (1997) have simulated the flow and turbulence in an IC engine cylinder using the Reynolds stress turbulence closure model. Finally, simulations were applied to IC engine like geometries. The results showed that the Reynolds stress model predicted additional flow structures and yielded less diffusive profiles than those predicted by an eddy-viscosity model

The typical shortcomings are found in stagnation flows, swirling flows, flows driven by turbulence and flows near curved boundaries. In order to remove some of these with a reasonable extra load, the use of a nonlinear stress-strain relation has been focused on, particularly in the last decade (e.g. Myong and Kasagi, 1990; Rubinstein and Barton, 1990; Gatski and Speziale, 1993; Shih *et al.* 1995. Most of them are quadratic eddy viscosity turbulence models (EVMs) and many of the low Reynolds number versions, however, still employ a parameter of wall distance for introducing near wall effects (e.g. Myong and Kasagi, 1990; Abe *et al.,* 1997. Such a topographical parameter is hard to be defined in complicated flow fields and thus undesirable to be used in the model expressions.

In the present investigation, the transport properties of the fluid flow on the suction stroke of an IC engine have been computed for seven different engine intake valve diameters from 14 mm to 32 mm. The engine speed was 1,000 rpm and flow was assumed to be turbulent and finite volume technique was adapted to this simulation work. The present article of research work has discussed the cylinder pressure at different crank angles with seven different intake valve diameters on the suction stroke of an IC engine. The effects of intake valve diameter on cylinder pressure and volumetric efficiency on the suction stroke of an IC engine have been investigated.

Mathematical Modelling

In the flow field, the mass and momentum conservation equations solved for general incompressible and compressible fluid flows and a moving coordinate frame Navier-Stokes equations are in tensor notation, Launder and Spalding (1974); Rodi (1979); Warsi (1981); E1 Tahary (1983).

$$\frac{1}{\sqrt{g}}\frac{\partial}{\partial t}(\sqrt{g}\rho)+\frac{\partial}{\partial x_i}(\rho\tilde{u}_j)=s_m \qquad (1)$$

$$\frac{1}{\sqrt{g}}\frac{\partial}{\partial t}(\sqrt{g}\,\rho\, u_i)+\frac{\partial}{\partial x_j}(\rho\,\tilde{u}_j\, u_i-\tau_{ij})=-\frac{\partial p}{\partial x_i}+s_i \qquad (2)$$

Constitutive Relation for Newtonian Laminar flow

$$\tau_{ij}=2\mu\, s_{ij}-\frac{2}{3}\mu\left(\frac{\partial u_k}{\partial x_k}\right)\delta_{ij} \qquad (3)$$

Where, μ is the fluid viscosity, δ_{ij} the Kroneker delta, which is unity when i = j and zero otherwise and s_{ij} is the rate of stress tensor given by

$$s_{ij}=\frac{1}{2}\left(\frac{\partial u_i}{\partial x_j}+\frac{\partial u_j}{\partial x_i}\right)$$

For Newtonian turbulent flow

$$\tau_{ij}=2\mu\, S_{ij}-\frac{2}{3}\mu\left(\frac{\partial u_k}{\partial x_k}\delta_{ij}-\overline{\rho u_i' u_j'}\right) \qquad (4)$$

Where S_{ij} is average strain rate and average turbulent momentum flux (average Reynold's stress) quantity $-\overline{\rho u_i' u_j'}$ is accompanied by so called turbulent model, which provide this unknown by expression of fluctuation in terms of mean component. To do so one has to rely on experimental data and knowledge obtained from direct numerical simulation. No single model can expect to reproduce well effects of turbulence on mean flow in all practical applications. The most popular turbulence models are eddy viscosity models, which postulate an analogy between turbulent and viscous diffusion (Boussinesq eddy viscosity hypothesis). K-ε model with wall function has been used for this prediction work. Particularly k-ε model is used for high Reynolds number to fully turbulent incompressible or compressible flow, Launder and Spalding (1974), and it allows to some extent for buoyancy effects Rodi (1979); Warsi (1983); E1 Tahry (1983)

Engine details and Flow Condition:

Engine bore............................=>	62 MM
Stroke length..........................=>	70 MM
Engine speed.......................... =>	1,000 rpm
Number of cells..................... =>	582
Number of Boundary faces.... =>	1,424
Transient Flow (G. rate)......... =>	Start from time step = 0
Solution procedure................ =>	Piso
Number of time steps............ =>	360
Time step size........................=>	DT= 1.666600E -04s
Post data frequency................=>	5
Solve.....................................=>	U, V, P, TE, ED, T, VIS, DEN,
Fluid flow.............................. =>	Turbulent compressible K-Eps
Characteristic length..............=>	1.600E-02 m
Reference pressure................ =>	PREF= 1.000E +05 Pa
Reference temperature...........=>	TREF= 2.730E+02 K
Molecular viscosity............... =>	Constant - MU = 1.810E -05 Pas
Density..................................=>	Ideal Gas : Background
Fluid molw.............................=>	2.896E+01
Specific heat.......................... =>	Constant - C = 1.006E+ 03 J/kgK
Conductivity.......................... =>	Constant - K = 2.637E- W/mK
Turbulent prandtl Number...... =>	Prtur = 9.000E-01

Combustion Chamber Geometry and Modetry and Modelling Strategy

The inside cylinder diameter has been taken 62 mm and engine compression ratio is 7.5: 1 considering usual SI engine specification and piston speed is 1,000 rpm for investigating fluid flow property at idling speed. For reference datum of crank angle, the bottom dead centre when exhaust valve opens has been assumed to be at 0.0 degree in exhaust stroke and top dead centre at the start of suction stroke has been assumed at 180 degree.

Modelling Strategy

This intake valve modeling was prepared with seven different engine intake valve diameters of 14 mm, 17 mm, 20 mm, 23 mm, 26 mm, 29 mm and 32 mm. Movement of the piston and valves of IC engine were produced by cell removal and cell addition. The mesh motion can be formulated in this problem by two conceptual steps. The first deals with connectivity changes i.e. cell removal, cell addition, reconnection etc. which is defined as ëeventí. The second step is to specify the grid vertex position as a function of time by supplying a set of grid manipulation commands to be executed at each time step.

The initial mesh contained all the cells that are used in the analysis of interest. The overlapped cells at valve bottom position and valve top position should be initially represented in the grid construction. When cells are added (activated) they are still deemed to be reconnected to the neighbors and follow the process of deactivation by removal of the cell from the neighbors. Thus sequencing constraints of activation and deactivation are followed. If any part of the solution domain becomes separated from the rest of the flow field during transient phases, (e.g. intake ports after the intake valve is closed) the cell material type there must be changed. So all material types are defined in the initial set-up.

Results and Discussions

The cylinder pressures in each degree of exhaust and suction stroke have been predicted with seven different engine intake valve diameters (14 mm, 17 mm, 20 mm, 23 mm, 26 mm, 29 mm and 32 mm) and these are depicted in Figure 1 to Figure 7.

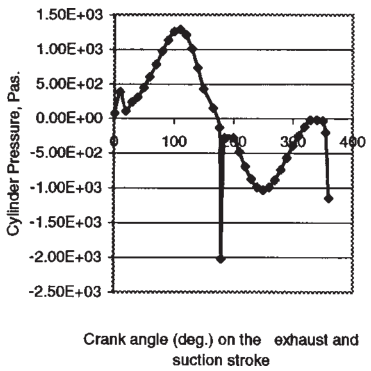

Figure 1 **Cylinder Pressure Vs. Crank angle on the exhaust and suction stroke with 14 mm intake valve diameter**

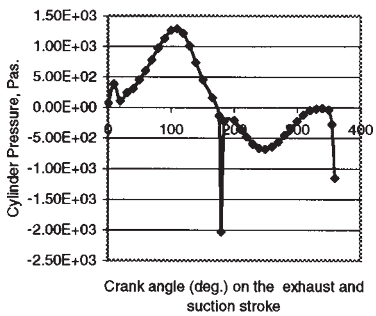

Figure 2 **Cylinder Pressure Vs. Crank angle on the exhaust and suction stroke with 17 mm intake valve diameter**

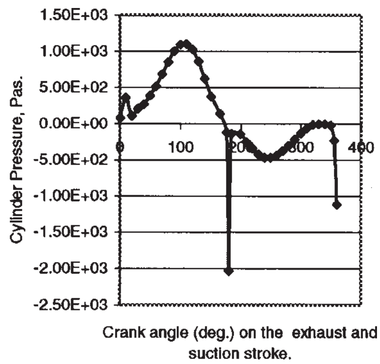

Figure 3 **Cylinder Pressure Vs. Crank angle on the exhaust and suction stroke with 20 mm intake valve diameter**

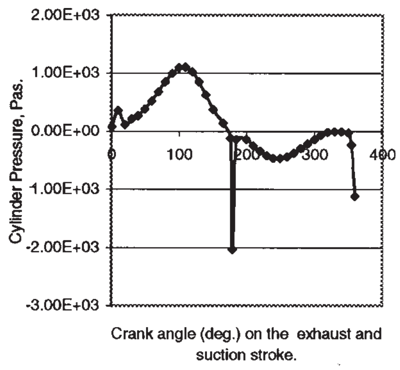

Figure 4 **Cylinder Pressure Vs. Crank angle on the exhaust and suction stroke with 23 mm intake valve diameter**

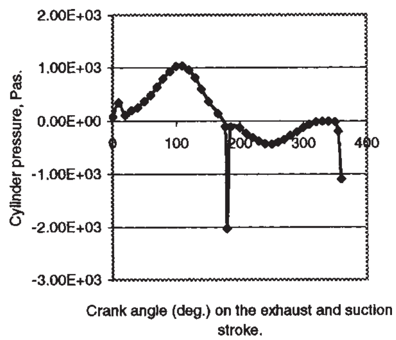

Figure 5 **Cylinder Pressure Vs. Crank angle on the exhaust and suction stroke with 26 mm intake valve diameter**

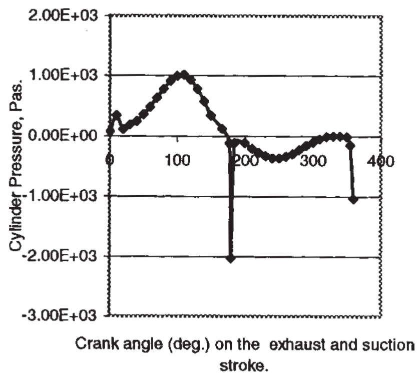

Figure 6 **Cylinder Pressure Vs. Crank angle on the exhaust and suction stroke with 29 mm intake valve diameter**

As reference datum of crank angle, the exhaust stroke is from bottom dead centre (0.0 ^{o}) to top dead centre (180^o) and suction stroke is from the top dead centre 180^o to 360^o. The exhaust valve opens during 0.0^o to 20^o in exhaust stroke and suction valve opens during 180^o to 200^o of suction stroke.

The pressure variations in exhaust and suction stroke have been shown in all figures. But the pressure variation in suction stroke (180^o to 360^o) is main interest of analysis in this paper. It is found from the above figures that the cylinder pressure instantly decreases and increases at the intake valve opening timing (180^o to 200^o). This pressure drop may be for short delay of suction valve opening response or very small suction valve opening depending on cam profile when piston moves downward from top dead center (TDC). When the intake valve fully opens and the piston moves at a faster speed towards middle of suction stroke, cylinder air pressure decreases with piston speed. The cylinder pressure attains a peak negative value near the middle of the suction stroke (250^o to 255^o). As the piston passes through the middle of the suction stroke, the negative peak pressure inside the cylinder reduces towards the atmospheric pressure with the advancement of the piston towards bottom dead center. The cylinder pressure becomes atmospheric when intake valve starts closing at 340^o. All the figures show similar nature of variation of pressure inside the cylinder in the suction stroke but the magnitudes of variation are different as discussed below.

The variation of cylinder pressures with different intake valve diameters as shown in Figure 1 to Figure 7, have been compared in Figure 8 where it is found that pressure variation is very prominent and distinguishable with intake valve diameter in the middle of the suction stroke but it becomes insignificant at the end of the suction stroke. A larger vacuum or higher negative peak pressure (-1,030 Pascal) is predicted inside the cylinder with the smaller intake valve diameter of 14 mm in the middle of the suction stroke. While the lower negative peak pressure inside the piston (-282 Pascal) is predicted with larger intake valve diameter. Thus a lower vacuum in the cylinder is predicted with larger intake valve diameter while a higher vacuum inside the cylinder predicted with a smaller intake valve diameter during the middle position (250^o to 255^o) of the suction stroke. However, as the piston passes the middle of the suction stroke, the cylinder pressure difference starts reducing. The pressure difference becomes 20 Pascal near valve closing timing (340^o) between higher (32 mm) and smaller (14 mm) intake valve diameters. At end of suction stroke the pressure difference becomes negligible.

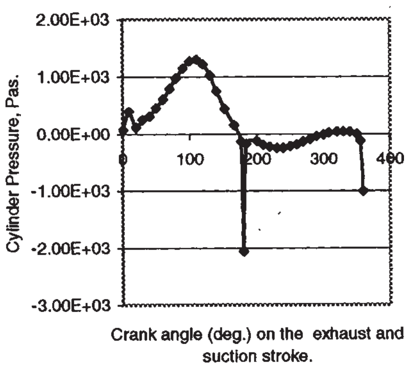

Figure 7 Cylinder Pressure Vs. Crank angle on the exhaust and suction stroke with 32 mm intake valve diameter

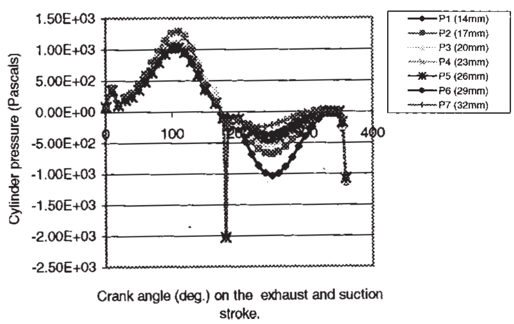

Figure 8 Comparison of cylinder pressures for sever different intake valve diameters on the exhaust and suction stroke

Thus the effect of increasing the intake valve diameter on the cylinder pressure as well as volumetric efficiency is found to increase initially until the middle of the suction stroke but this effect becomes negligible at the end of the suction stroke.

Conclusions

The following conclusions are made from this computational work.

(i) The cylinder pressure falls initially from the suction valve opening to the middle of the suction stroke and then the cylinder pressure increases until the suction valve is completely closed in all seven different intake valve diameters.

(ii) The suction air pressure inside a cylinder and its peak is found to increase with the increase of the intake valve diameter until the middle of the suction stroke but its effects diminish at the end of the suction stroke.

(iii) Thus the cylinder pressure and volumetric efficiency are found to vary with the engine intake valve diameter initially but its overall effect at the end of the suction stroke becomes negligible.

(iv) The intake valve diameter is the measure of flow area and this flow area also depends on the number of intake valve. Thus the result of the present investigation can help manufacturers to reduce the intake valve size and intake valve number in a four stroke IC engine.

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Appendix

CI	Compression ignition
SI	Spark ignition
g	Standard gravitational constant
k	Turbulent kinetic energy (TE)
p	Piezometric pressure = $p_s - \rho_o g_m X_m$
P_s	Static pressure
g_m	Gravitational field components
s_i	Momentum source components
s_{ij}	Rate of stress tensor
S_{ij}	Mean strain rate tensor
s_m	Mass source
t	Time
u_i	Absolute velocity component in direction x_i
u_j	Absolute velocity component of fluid in direction x_l
$\tilde{u}_j$	$u_j - u_{cj}$, relative velocity between fluid and local (moving) coordinate frame that moves with velocity u_{cj}
u_{cj}	Velocity of local (moving) coordinate frame.
x_i	Coordinate (i = 1,2,3)
x_j	Corrdinate (J = 1,2,3)
δ_{ij}	Kroneker delta = 1 when i=j and zero otherwise
ρ	Density
μ	Fluid viscosity
τ	Shear stress
τ_{ij}	Stress tensor components
η_V	Volumetric efficiency

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