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Withdrawal Behaviour Through a Single Round Hole In a Cross Flow

A. K. M. Sadrul Islam' .I. J. McGuirk"

ABSTRACT

Experimental measurements of drawdown, the critical withdrawal rate and the mean temperature profiles in the near field region of a withdrawal system in a two-layered thermally stratified cross-flow are presented. Flow visualisation is also reported which provides detail information on the structure and dynamics of the warm/cold water interfaces for different in take flow rates. The present experimental data on the critical intake flowmte are comparedwith the measurements ofGoldring(1984).The drawdown results indicate aninverse dependency on density gradient unlike the case without cross flow.

NOMENCLATURE

D	hydraulic diameter, 4* area / wetted perimeter
DD	drawdown fraction, $(T_o - T_2)/(T_1 - T_2)$
DD_m	peak drawdown fraction
Fr_o	Inlet Froude number, $U/(g'H)^{1/2}$
g	gravitational acceleration
g'	reduced gravitational acceleration, $g\,\Delta\rho/\rho_1$
h_1	depth of warm water layer
X,Y,Z	cartesian coordinates measured from the axis of the intake hole
Φ	normalised temperature, $(T - T_2)/(T_1 - T_2)$
ρ_1	density of warm water
ρ_2	density of cold water
$\Delta\rho$	density difference, $(\rho_2 - \rho_1)$
ν	kinematic viscosity
h_2	depth of cold water layer
H	$h_1 + h_2$
Q_o	intake flow rate
Q_o^c	critical intake flow rate
U	cross flow velocity
Re	Reynolds number, UD/ν
T	mean temperature of the field
T_1	temperature of warm water
T_2	temperature of cold water
T_o	mean temperature of the intake flow

INTRODUCTION

In natural water bodies density stratification is caused by the presence of a varying temperature with depth due to variation of observed solar radiation, also by a vertical profile in the concentration of

dissolved and suspended solids. This naturally occuring stratification is sometimes enhanced by the rejection of large volumes of waste heat from costal power stations in the from of a warm water discharge which tends to form a low den-sity floating surface layer. This tlpe of stratification is usually of stable nature and is important in several flow problems of engineering interest, e.g. management systems for water quality control in reservoirs, waste or warm water discharges into natural water bodies and cooling water intake from reserviors, seas and rivers. In many of these flows the density variation combines with gravity to produce buoyancy effecs which can crucially influence the fluid dynamic behaviour. The buoyancy force inhibits vertical motion and in some engineering problem, specially the management of water quality of reservoirs and the intake of cooling watcr, the flow is coming from a spatially limited sele,ctive layer or region. This phenomena is known as selective withdrawal Selective withdrawal phenomenon has often been studied for two dirnensional laminar flow cases. Brooks and Koh (1969) and Imberger (1 980) have provided a very good review of these cases. In practice, specially for the intake of cooling water, this phenomena is of threedimensional (3D) nature, and needs an extensive study. This paper deals with this type of 3D problem. A single round hole is considered for the intake geometry to withdraw selectively cold water from a two layered thermally stratified cross flow.

EXPERIMENTALI SETUPAND PROCEDURE

.

The expriments were carriedout in aflumeof 0.4m width, 0.5m deprh and Bm length. The withdrawal hole was situated on the smooth wooden bottom surface at the symmetry axis of the flume, 2.5m downstream of the entry section. The warm and cold water were innoduced into the flume from trvo consfant head ianks and the heights of the warm and cold water layers were maintained by means of a splitter plate. Figure 1 shows a schematic view of the experimental flume.

Goldring (1984) obtained experim-ental correlations for critical drawdown conditions for different cross-flow and intake hole diameter conditions. In the present study, Goldring's expgriment was extended with single round hole to study the mixing behaviour in the near field region of the intake system. The diameter of the hole was d=42.5 mm and the cross-flcw velocity was U=31.25 mm/sec for both layers. Seven experimental runs were made !o map the temperature field at two Reynolds numbers, 8330 and 9850 and the inlet Froude number, Fr. ranges from 0.72 to I .3. Table 1 gives the derails of the parameters of these seven experimental runs.

The water temperature was measured by means of nine miniature bead (1 .5 rnm dia) thermistors ®S Data I5L-142) calibrated to an accuracy of + 0.01oC; six

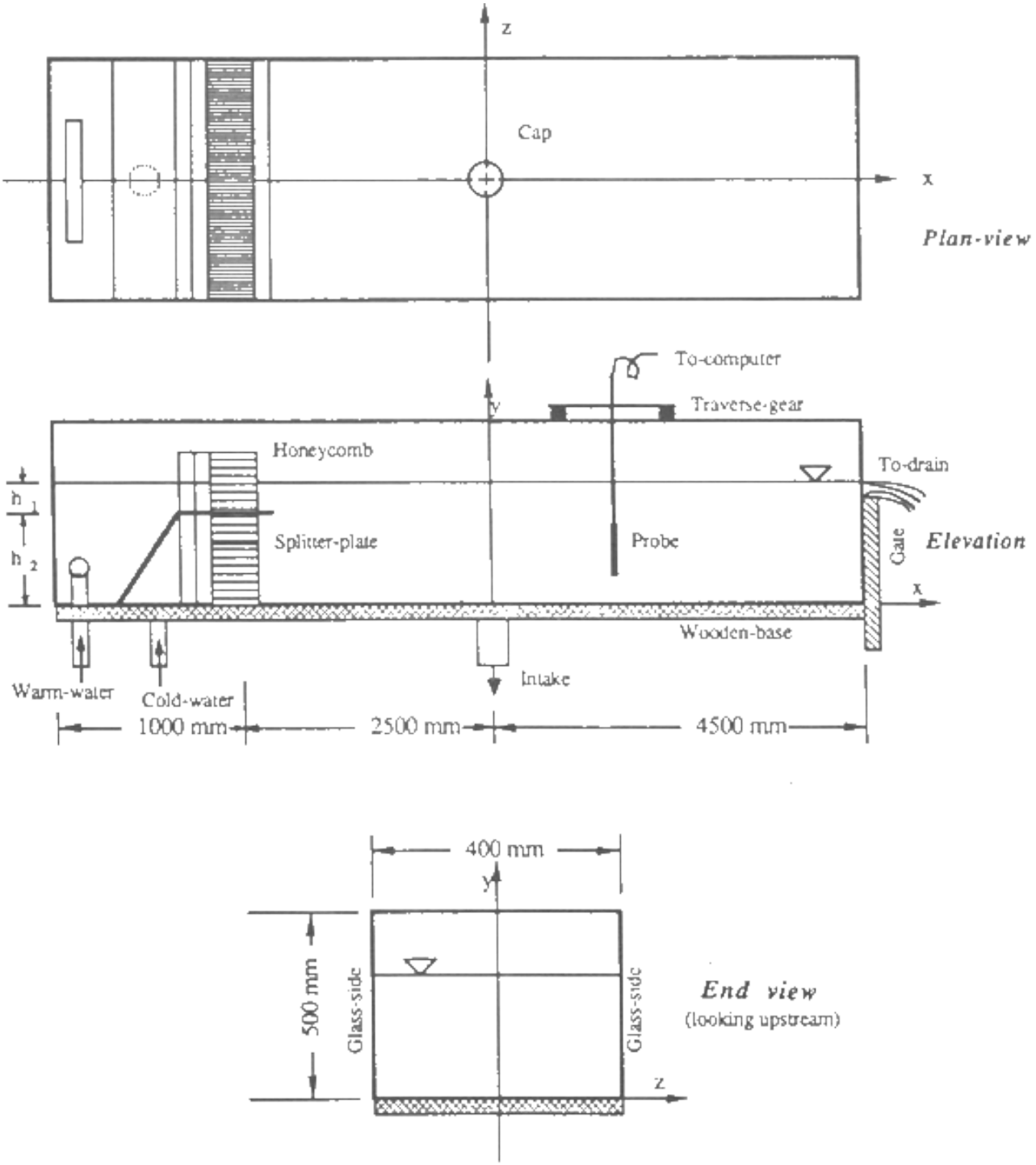


Fig. 1 The cross-flow flume.

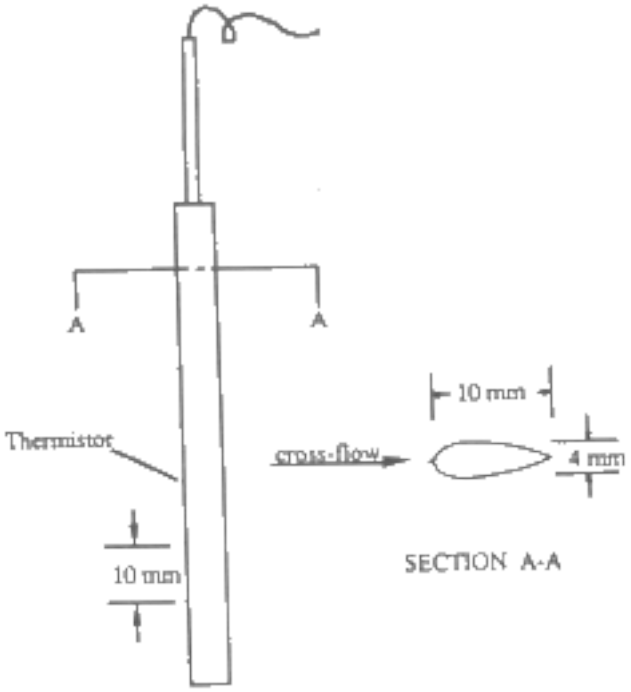


Fig. 2 (a) The probe for field temperature measurements.

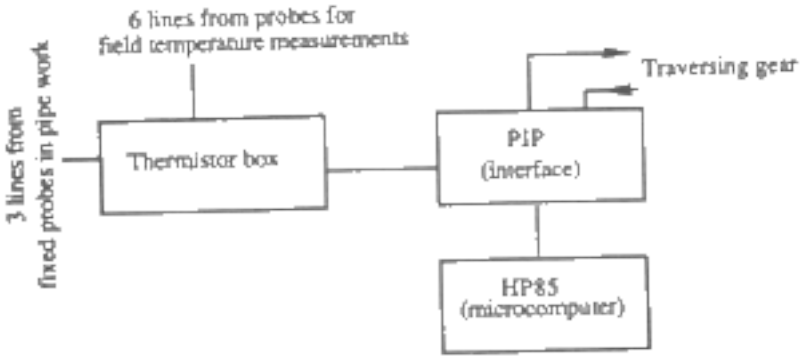


Fig. 2 (b) Block diagram for electrical connection of thermistor probes and traversing gear to the microcomputer.

of which was for fi eld temperature mapping. The therm istor box which converts the millivolt (mv) signal to an equivalent resistance and then an anologue signal which was digitised by an interface (Plant Intedace Peripheral) and sent to a Hewlett Packard HP85 microcomputer where an equivalent temperature reading ("c) was reco-rded. Fig. 2(b) illusfiates these connections by a block diagram. The field temperature thermistors were fixed on a small (10mm x 5mm) stream linedplastic rod, (fig. 2a) which was traversed by a traversing gear monitored by the microcomputer. The time taken by the system for sampling readings for the nine thermistors and converting the readings into temperature ('c)

was approximately 1 second.

The temperature field in the wake of the hole was observed to fluctuats due to the turbulence created by the flow disturbance caused by the intake system. The amplitude of these fluctuations increascd with the intake flow rate. To averaging out this fluctuations, different sample sizes ranging from 30 to 300 depending upon the flow condition were taken.

FLOW VISUALISATION

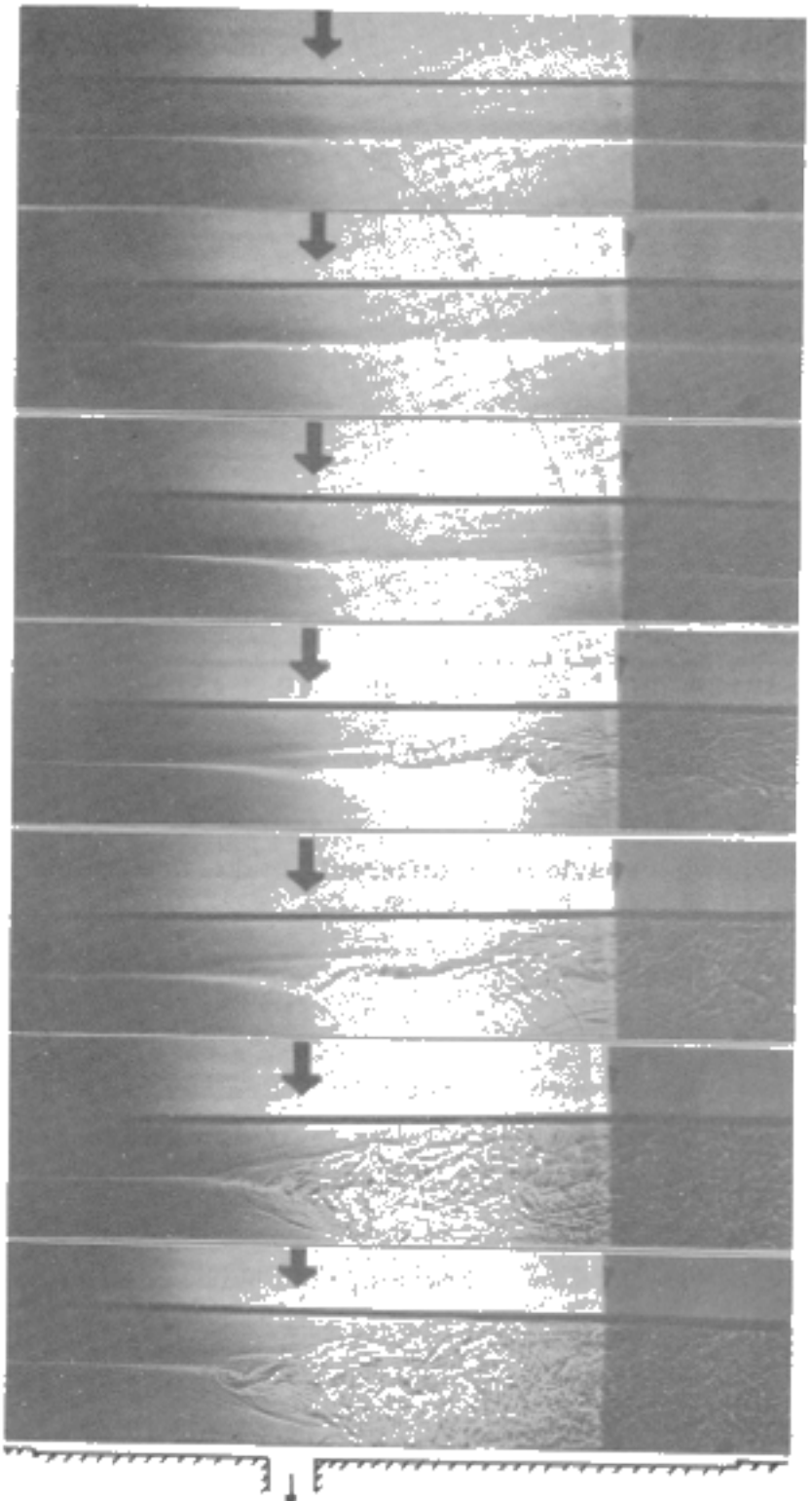
Shadowgraph technique was used to capture the features of the mixing and distur-bance created in the wake of the intake system (hole). Figure 3 shows photographs taken from shadowgraph images for no intake flow rate and six different intakes flow rates before and after the incipient (critical) drawdown occurs. It can be seen from the photographs that upstream of the hole the buoyancy has damped outall enlrainment, the interface thickness remains constant and the flow becomes laminar and two dimensional. As the intake flow rate increases the turbulence generated at the hole breaks up the interface and in the wake region the two. layers are mixed up and the mixed layer is pulled down towards the hole, causing a significant proportion of drawdown.

EXPERIMENTAL RESULTS AND DISCUSSION

The drawdown fraction (DD) is calculated with the tnean temperature of the intake flow. Since the temperature of the intake flow fluctuates and this fluctuation also depends on the intake flow rate, the mean temperature was obtained by averaging 100-600 samples depending on the intake flow condition. Figure 4 shows a typical drawdown behaviour at different intake flow rates. After drawdown onset the drawdown fraction increases linearly with Q" and then tends towards an asymptofic value. The critical intake flow rate, Qo was obtained by the projection of the linear portion of the drawdown curve to the DD = 0 line (after the practice of Jirka & Katavola, 1979 and Goldring 1984). The asymptotic (orpeak) value observed in the drawdown curve has a great practical importance in the context of intake system design. For example, for the case of fig. 4, for well mixeel conditions at the upstream the drowdown will be

$$DD_m = \frac{h_1}{h_1 + h_2} = 20\%$$

which gives a defining limit for selective withdrawal (see appendix for derivation). For DD > DD*, the intake systern selectively withdraws water from warm upper layer and for DD < DD* it withdraws selectively from the cold lower layer. The optimum system is therefore that which leads to a maximum reduction in the peak drawdown value below DD* . During the present experimental studies, the peak DD observed were always substantially bclow DD



Shadowgraph pictures for plain hole intake system at different intake flow rates. h_1 = 50 mm, h_2 = 80 mm, T_1 = 9.4°C, T_2 = 4.6°C

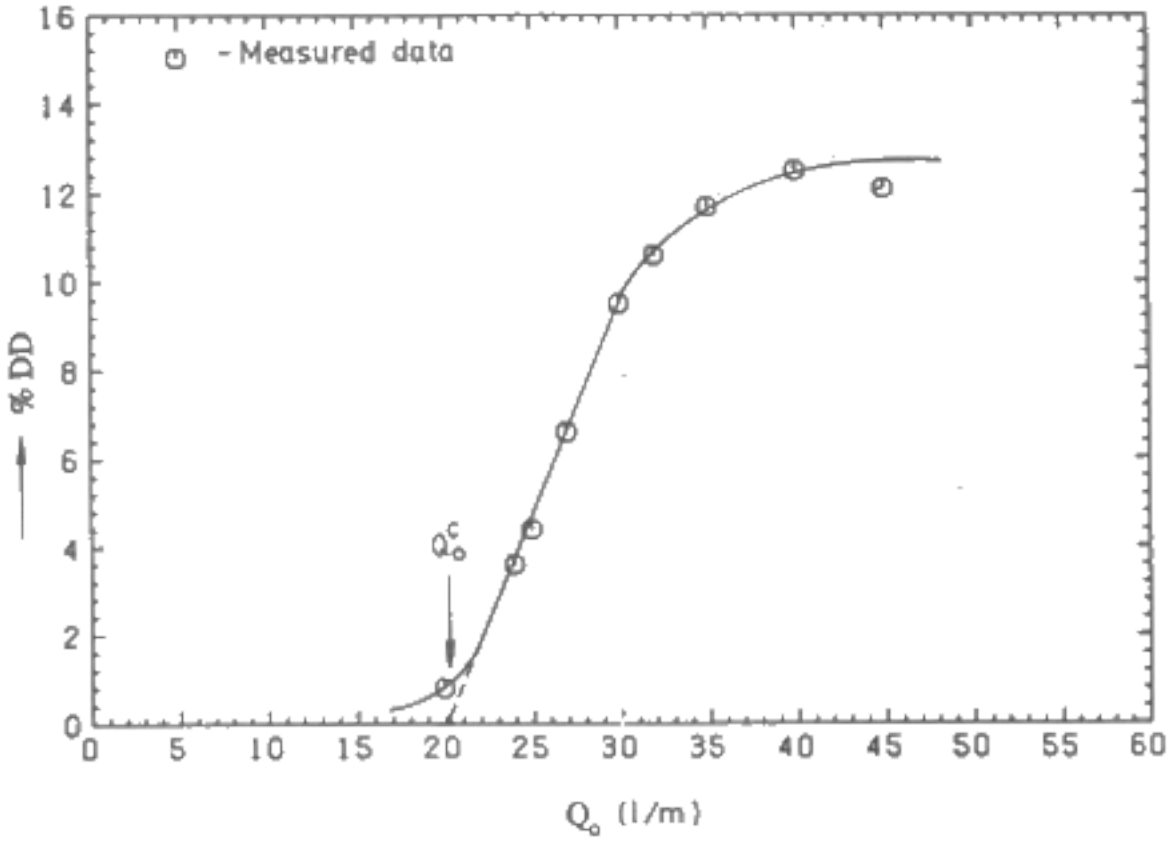


Fig. 4 : Drawdown behaviour at different intake flow rates for h_1 = 20 mm, h_2 = 80 mm, ΔT = 12°C, Fr_o = 0.72 and Re = 8330.

showing that in all cases the intake system selectively withdrew water from the cold lower layer. Goldring (1984) did however report a few cases where the maximum DD values were slightly greater than DD-. The critical drawdown results of the present investigation are compared with the correlation equation (Goblring 1984) in fig. 5.The present measurements agree quite well wittr the correlation equation (1).

$$\frac{h_2}{d}\left(\frac{U}{\sqrt{g'h_2}}\right)^{0.575} = 0.763\left(\frac{Q_o^c}{\sqrt{g'd^5}}\right)^{0.386} \quad (1)$$

Equation (1) rcveals that for the cross-flow cases the critical intake flow rates, Qo" shows an inverse proportionality with density difference Dr , which is in direct contrast to the belief that the buoyancy inhibits drawdown, although this belief is supported by the axisymmefric withdrawal from stagnant environment. (see, Craya 1949, Harleman et al 1959, Goldring 1981, Ivey & Blake 1985, McGuirk & Islam 1987).

The effect of Dr on drawdorvn at different supercritical iniake flow rates (Q. > Q.) are also presented in

figure 6. By increasing Dr, the drawdown is increased .

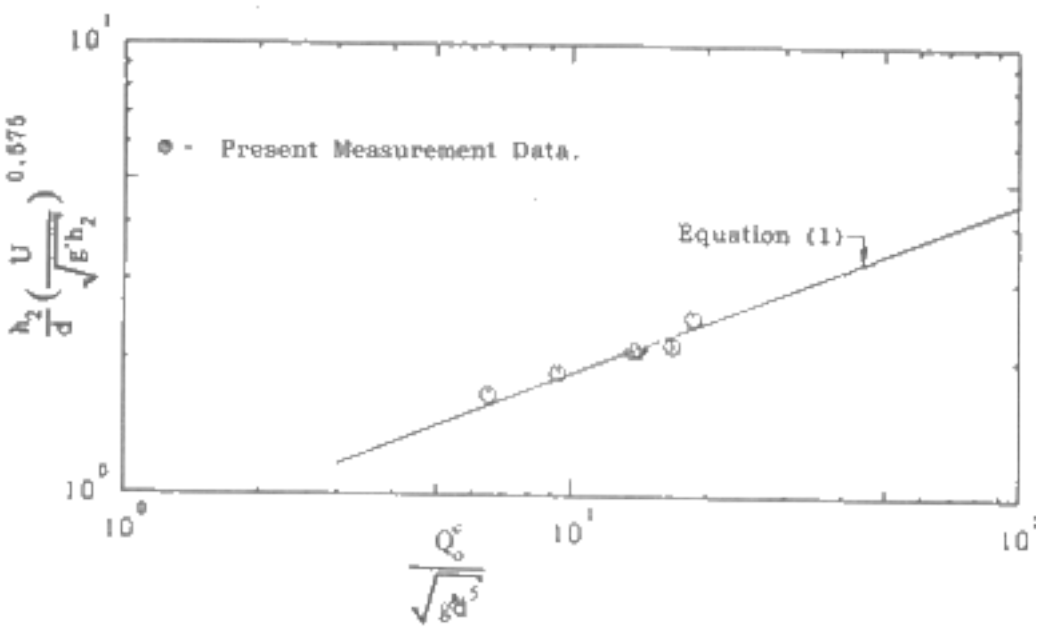


Fig. 5 : Critical drawdown.

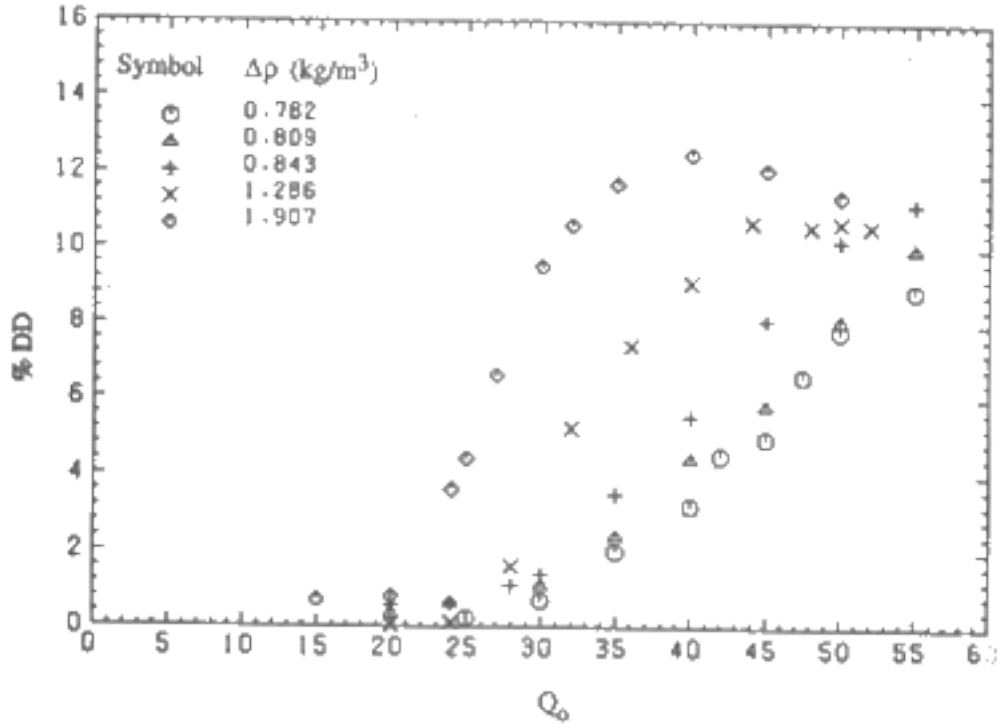


Fig. 6 Effect of buoyancy on drawdown. h_1=20mm, h_2=80mm

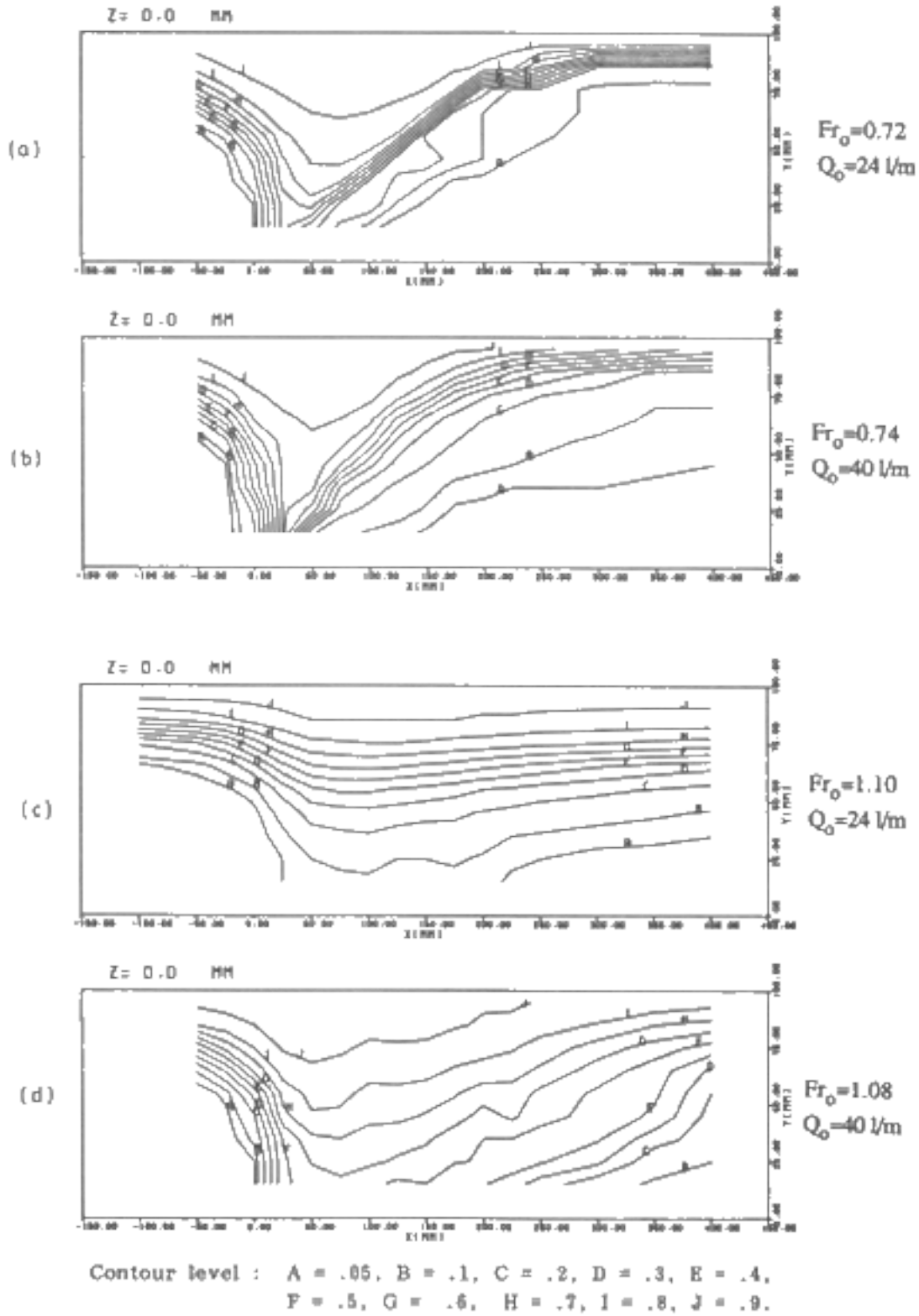


Fig. 7 : Measured contours of normalised temperature, Φ on the symmetry plane.

Thepeak value of DD observedalsoincreases with Dr. An explanation was given by Islam (l9g8) for these facts by observing thepredicted flow and temperaiure field forthese cases. The buoyancy forces,

proportional to the density gradient at tie therrnocline inhibit vertical motion of the fluid. For higher density gradienr, this inhibiting process is stronger, so that less fluid is withdrawn vertically as the layer passes over the hole, and a larger fraction is drawn hence mixing the density differences are smaller. This mixed fluid leads then to higher drawdown fracrion. This explanation is also supported by the measured temperature field shown in figure 7. Figure 7 shows the normalised femperature, F field on the sysmmtry plane. The measured perturbation of the temperature field in the near region of the hole is prcsented for approximarely ffio diffcrent inlet Froude numbers and for two different intrke flow ratcs. As Q. increases, the vertical perturbation of a given tempcra_ ture contour increases for the same Fro cases and causes. more drawdown. For the same e cases, the warm water floats up more quickly for the strong buoyant cases (lcss Fr"), but in the vicinity of the hole more warm waters are pulled down and causes more drawdown, which supports the explanation made above.

CONCLUSIONS

The major conclusions from the present experimental observations may be summarised as follows :

(I) Flow visualisation served as a good guirle to uncler_ stand the flow behaviour in the near field region of an intake system.

(ii) The present drawdown data are in goocl agrecnlcnt with the correlarion equarion (t) of Goldring,s (19g4) experimental data.

(iii) An inverse dependency of density clifference on from the downsheam, where due to more turbulence and crjtical inrake flow rate, e"., drawdown behaviour and peak DD valuewas observed, unlike thecase wifiioutcross flow.

ACI(NOWLEDGEMENT

The work reported here was performed at Central Electricity Research lab, Leatherhead, U.K. and at Imperial College, London, U.K., during the renrre by the first. aulhor (A K N,l S I) wirh a Brirish Commonwealth Scholarship.

REFERENCES

1. Brooks N. H. & Koh. R. C. y. (1969) ..Selecrive Withdrawal from Density SradfiedReservoirs,,, J: Hydr. Divn, ASCE Vol.95, Hy4, pp. j36g_1400.

2. Craya A. (1949)..Recherhes Theoriques Sur L, Ecoulement de Couehes Superposees deFluidesde Densitcs Differentes," La Houille Blanche, ptp.44_55.

3. Goldring B. T. (lggl) ,.The effect of a hood on drawdown Criteria inZero Crossflow,,, Report No

RD/L/ 203 I Ai8 1, Central Elec trici ty Research Lab., Leatherhead, U.K.

4. Goldring B. T. (19g4) .. Setecrive Wirhrtrawal by Capped and uncapped inrakes in a Tidal flow,,, Re_ porr' No TPRDn/267glNs+, cenrrar Ercctriciry Research Lab., Leatherhead, U. K.

5. Flarlenran D. R. F., Morgan R. L. & purple R. A

6. Imberger, J. (1980) "selective Withdrawal : A Review", 2nd Int. Symp. on Stratified Flovs, IAHR, Nor way, pp. 381-4

7. Ivey G. N. & Blake S. (1985) "Axisymmetric Withdrawal and Inflow in a Density Stratified Container", . J. Fluid Mech., Vol. 161, pp. 115-1

8. JirkaG. H. & KatavolaD. S. (1979) "Supercritical

9.McGuirk J. J. & Islam A. K. M. Sadrul (1987) "Numerical Modelling of the Influence of a Hood onAxisymmetric Withdrawal from a Density Stratified Environment", Proc.3rd Int. Symp. on Stratified Flows, Caltech, Pasadena, U.S.A.

i0.

Islam A.K.M. Sadrul (1988) "Prediction of Selective Withdrawal Phenomenon in Stratified Cross Flowing Streams", Ph.D. Thesis, Imperial JirkaG. H. & KatavolaD. S. (1979) "Supercritical College, London, U.K.

Table 1

Experimental Parameters for Intake Flow Through a Single Plain Hole.

Run No	h_1	h_2	T_1 °C	T_2 °C	Q_o l/m	Fr_o	Re
1	50	80	12.4	7.6	24	1.28	9850
2	50	80	12.1	7.3	30	1.3	9850
3	50	80	12.3	7.6	40	1.29	9850
4	20	80	22.0	9.7	24	0.72	8330
5	20	80	21.3	9.3	40	0.74	8330
6	20	80	16.2	9.8	24	1.1	8330
7	20	80	16.3	9.7	40	1.08	8330

Appendix

Drawdown fraction is defined as the ratio of the amount of the warm upper layer withdrawn to that of the total mixed fluid withdrawn and is expressed as :-

$$DD = \frac{m_{0,1}}{m_0} = \frac{m_{0,1}}{m_{0,1} + m_{0,2}} \quad \ldots\ldots \quad \ldots\ldots \quad (A.1)$$

where m_o is the mass of the fluid withdrawn and the subscripts 1 and 2 refer to the masses of fluid withdrawn from the warm upper layer and cold lower layer respectively.

Now from mass balance,

$$m_o = m_{o,1} + m_{o,2} \qquad (A.2)$$

and from energy balance,

$$m_o T_o = m_{o,1} T_1 + m_{o,2} T_2 \qquad (A.3)$$

assuming C_p of the fluid is constant in the range of temperature considered here.

Equating equations (A.2) and (A.3) we have,

$$DD = \frac{T_0 - T_2}{T_1 - T_2}$$

For well mixed conditions at the upstream of the intake hole, the intake mass flow rate will be,

$$m_o = k(m_1 + m_2) \qquad (A.4)$$

where k is any constant and m_1 and m_2 are the mass flow rates of the warm upper layer and cold lower layer fluids. Then the peak drawdown fraction is given by,

$$DD_m = \frac{km_1}{m_0} \qquad \ldots\ldots\ldots\ldots\ldots (A.5)$$

Since,

$$\frac{m_1}{m_2} = \frac{h_1}{h_2}$$

for the flume considered here and equating equations (A.4) and (A.5), we have,

$$DD_m = \frac{km_1}{m_0} = \frac{h_1}{h_1 + h_2}$$

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Autoignition of Natural Gas Fuelled Engines -A Review of its Possibilities

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ABSTRACT

This paper outlines a general review of the different possibilities to upgrade Pqwgl output and efficiency bf conventional natural gas fuelled engines to the dieSel fuelled levels. Thii includes a brief introduction on various aspects and feasible ways of how to achieve "diesel-like" combustion in current natural gas engines. This promises to be an interesting area wdrth to be reseaiched for furgher potential development.

Keywords : Atrtoignition, Natural Gas

INTRODUCTION

Conventional natural gas fuelled engines have a considerable economic advantage over their diesel and gasoline engines cqunterparts, and they tend to be technologically innovative and also more environmentally friendly. Natural gas is already chemically ready for use and requires no preliminary treatment' Another advantage of natural gas is its wide flammability limits, especiaily to the fuel-lean side where ignition of natural gas can occur in the presence of more than 200% theeoretical air [1]. Operating in this regime will ensure improved fuel economy and lower emission level. However, this advantage will no longer be enough on its own and may be challenged, and may even decline as a result of the completion for the advanced diesel engines in their next development stage.

As a consequence, the only way natural gas fuelled engines can compete is to improve their shaft efficiencies. This is also because conventional spark-ignition er,gi.es technology does not allow the means to achieve "diesel-like" combustion through the direct injection of high-pressure natural gas into the cylinders.

This paper briefly discusses the issue of direct-injection of high-pressure natural gas and a basic study on the various conventional approaches to ignite natural gas, with respect to their possibilities and changes leading to "diesel-1ike" combustion or autoignition, based on the basis of engineering concerns.

The ramifications of implementing this conceptual development will not only raise the gas engine efficiency and power output, but also the ability to take advantage of future advances in diesel engine technology, such as reducing heat loss through improved combustion chamber insulation.

NATURAL GAS

Basically, natural gas consists of carbon and hydrogen, mainly in the form of methane, CH4. Having a simple and stable molecule it is extremely resistant to self-ignite and thus has a low cetane rating. It is also able to mix homogeneously with air to produce a combustible mixfure, and is excellent for use in "lean-burn" mixtures which are comparatively less polluting. All these characteristics have made natural gas an instrinsically ecological and environmentally friendly fuel. Therefore, it comes close to the concept of an ideal fuel for this generation and the generations to come.

The octane rating of natural gas is approximately 130, this indicates that sparkignition engines running on natural gas can operate at compression ratios up to 16:1 (normal compression ratios for such engines is between 7.5:1, to 9.5:1) without "spark-knock". Such a tremendous rise in compression ratios will certainly bring about power improvement of 25"/o to 40"/' over that of today's low compression gas engines [2].

Being able to achieve a relatively high compression ratio in a natural gas engine is important. The higher ratio causes an increase in thermal efficiency, which in turn reduces fuel consumption. While refering to the general trends revealing the relationship between cetane number, temperatures, fuel type and ignition type shown in the chart generated by Haddad and Watson [3], it was reflected that fuels having cetane number lower than 25 will exhibit a significant feature in ignition delay. Thus it is not hard to perceive that the considerable delay in ignition with natural gas is characterised by its low cetane number.

DIRECT-INIECTION IN NATURAL GAS ENGINE TECHNOLOGY

Power output of the current natural gas fuelled engines, utilizing a carburetted, spark-ignited combustion system (Otto-cycle) is limited by knock of the homogeneous fuellair charge to approximately 1250 kPa (180 psi) brake mean effective pressure [4].

The knock problem and its related power limitations could be eliminated by directly injecting high-pressure natural gas up to 20000 kPa (3000 psi) into the cylinder near the top dead centre with a gas injection, incorporating ignitionassist (glow plug is used) to ignite the low cetane natural gas. And this will produce a "diesel-like" autoignition.

By eliminating the knock limit, the direct-injected gas system can utilize higher compression ratios and higher charge air density than the current gas engines, and furthermore, the flow losses associated with the carburetor will also be avoided. Moreover, the future diesel engine technology improvement

including insulation can be applied to direct-injected gas engines, but cannot be applied to the current gas engines due to knock limitations.

In conjunction with the above, future improvement in the direct-injected natural gas engine, even accounting for the additional parasitic losses associated with increasing the natural gas pressure with a gas compressor is expected to give a significant improvement in net thermal efficiency.

For the direct-injected natural gas engines, special hardware included an electronic gas injector (similar to the diesel's electronic unit injector), a cylinder head incorporating a glow-plug-ignition-assist system and a compressor in the line of fuel supply will all need to be developed to attain a good combustion of the low cetane natural gas fuel.

The Gas Research Institute of Chicago of U.S.A. revealed succinctly in their Annual Report, that natural gas engine power and thermal efficiency comparable to a diesel are achieveable by incorporating direct gas injection with glow plug ignition assist (DIG/GPIA) during tests on a Caterpillar 3400 series single rylinder laboratory demonstration engine [a]. The DIGIGPIA system will allow the natural gas engine to have power and thermal effiency improvement of up to 50% and 137o, respectively, compared to current nafural gas engines.

HYPERGOLTC I SPONTANEOUS COMBUSTION

Hypergolic combustion is a type of ignition and combustion process, in whcih fuel and oddant pair ignites spontaneously and is rapidly combusted. As a result both the ignition delay and the combustion duration of fuel are negligible. For the purposes of this discussion, the phenomenon of hypergolic combustion can be seen to resemble autoignition,/spontaneous combustion. Taylor [5] defines autognition in a fuel-air mixture as a rapid chemical reaction not caused by an external ignition source such as a spark, a flame, or a hot surface.

The hypergolic combustion is a phenomenon which has a promising potential to be applied to a direct.injected natural gas engine. It is aimed at reduction in the ignition delay, improving the quality of combustion, reducing emission levels and increasing overall engine performance.

The primary drawback of most of the listed ignition and combustion alternatives to be discussed in the ensuing section (section 5) revolves aroudn the fact that the bulk of the heat release would be flame speed limited as the flow of natural gas is much slower than gasoline fuel.

Conceptually speaking, the aforementioned problems could be overcome if the natural gas is hypergolically combusted. In another word, prior to injection, if the natural gas is chemically activated, once it is injected it will ignite sponteneously and combustion will occur rapidly. Therefore, if this could be done, the flame speed limitation could be circumvented and the combustion event could be also controlled through the rate of injection. Furthermore, the overall air/fuel ratio and compression ratio could be arbitrarily selected so as to obtain the best trade-off for high efficiency and low emissions.

The basic theory of hypergolic combustion and the experimental results obtained in an engine test fuelled by liquid fuel are presented by Hoppie [6] and Hoppie and Schamweber [2.

Theoretical results prediction speculated by Hoppie [6] indicated that the ignition delay can be made arbitrarily small and essentially independent of air temperature if the fuel is sufficiently preheated.

As demonstrated by Hoppie and Schamweber f7l, Hypergolic combustion could be realised in a internal combustion engine by means of preheating the fuel. The reduction in ignition delay arid increase in combustion rate will result, in conjunction with fuel injection rate and'duration control would provide much better control of the combustion event. Therefore, following this, the engine power output, emission levels and engine efficiency can be optirnised'

With reference to the above arguements, one can propose that thermal energy can be used as a means of activating the fuel, i.e., as a means to produce fuel radical such as; CH4 to CH3, CH2, H,etc.

In the midst of optimism, engineering problems in handling high temperature fuel must not be ignored and it needs to be resolved before the idea of hypergolic combustion can become [ractical and technically viable.

IGNITION AND COMBUSTION ALTERNATIVES

In combustion, the "reaction" is not a single oI even a few-step Process; the actual chemical mechanism consists of a large number of simultaneous, interdependent reactions or chain reactions. In such chains there is an "initiating reaction" where highly reactive intermediate species or radicals are produced from stable molecules, i.e., from fuel and air. This is then followed by propagation reactions radicals react with the reactant molecules to form products and other radicals to continue the chain, Heywood [8].

This section presents the possible alternative approaches, having potential to start "intiating reaction", in which more effort is required to evaluate and investigate both theoretically and experimentally each scheme's possibility in leading to "diesel-like" combustiou for natural gas engine.

Thermal Energy Activation

The mathematical model developed by Hoppie [6], which predicts that ignition delay is a function of fuel and air temperature, implies that the concentration of chemically active fuel radicals can be significantly increased via thermal dissociation of the fuel by preheating it. Therefore, if the fuel is so activated, it n'ill ignite and be consumed much more rapidly upon injection into cylinder than through fuel injected at ambient temperature.

A major anticipated shortcoming, associated with thermal activation is in the iact that there is a tendency of coke formation if fuels are held at high temperatures for long duration time due to clehydrogenation. Compression heating could be possible to minimise the occurrence of coke

formation. Using the process of compression heaitng, i.e. through rapid compression, it is conceivable that the desired radicals could be created and injected prior to the coke formation. Another practical way of eliminating coke formation is through the addition of water to the gas prior to heating.

Catalytically Enhanced Thermal Energy

This criterion is based on the fact that upon employment of some suitable catalyst, chemical reactions can be forced to occur at a temperature lower than normally expected. This is also a practical way to generate fuel radicals at a temperature just low enough to avoid the formation of coke. Other advantages of this scheme are the provision of a sustained combustion with lean mixture and also low pollutant product, owing to more complete combustion.

There are two possible means of modifying natural gas in order to achieve autoignition, and hence combustion, without assistance upon injection into an engine. They are (i) catalytic activation of natural gas and (ii) catalytic activation of natural gas and air. The basic idea behind catalytic enhancement of natural gas rests on the fact that a suitable catalyst can create certain chemical species or active radicals, at a lower temperature than normally would be required if no catalyst is present. Some potential catalysis are MgO, metallic nickle, both alpha and gamma phase Al2o3 and sio2. This can be accomplished by the introduction of an in-line catalyst chamber at the inlet of the fuel line, in whcih contains the pelletized catalytic materials.

Partial Catalytic Combustion

This concept has been developed in the Jet Propulsion Laborary and Siemens of u.s.A., Houseman and Cerini [10]. Their idea is to improve liquid-fuelled sparkignited engines by first converting the liquid fuel into a hydrogen-rich gas, in *ni.r, such a fuel would offer improved effeciency and lower emissions. To convert to a hydrogen-rich gas, the liquid fuel was first vaporized, mixed with a small amount of air, and then allowed to partially combust with the aid of and in the presence of a suitable catalyst. It was found that this could be accomplished without coke formation, and that the product gas was rich in fuel radicals and atomic hydrogen, at an approximate temperature of. 66 degree celeius (150 degree F). This product gas would be directly injected into the engines to achieve autoignition, which is t\e concept of hypergolic combustion.

Compression Ignition

In this concept, compression combustion via preheated inlet air can occur without relying on any external source of ignition in a direct-injected natural gas engine' Having u ,r"iy low cetane rating compression ignition of natural gas is quite difficult to achieve in a conventional diesel engine without

modification of the engine. However, conventional compression ignition can be accomplished via preheated inlet air. This is a very promising method leading to hypergolic combustion. In this case, waste heat given out by engine can be harnessed to preheat the inlet air, either through heat-exchange from colling system, lubrication system or exhaust system.

Electrical Energy Ignition

Theoretically, the establishment of an electric discharge in the natural gas will generate radicals. This may be done by utilizing a corona or an arc discharge. One can anticipate that, even through direct injection of natural gas, spontaneous ignition is still far from reality. The actual heat release would most probably result from flame propagation, which is predicted on the establishment of a sufficiently large flame kernel resulting from the interaction of a rather small portion of the injected fuel with the ignition source. Due to the excessively combustion duration, low efficiency and high emission could thus be expected.

If methane or natural gas is electrically activated with an arc discharge during injection, negligible ignition delay and rapid combustion are possible. Theoretically, this offers a great practical value. With proper amount of electrical activation, a controlled combustion event could be achieved with accompanying low levels of emission.

Photochemical Dissociation

Prior to injection, an ultra violet source could be employed to activate methane by projecting an ultra violet beam into a portion of natural gas to establish chemically active, unstable fuel radicals from stable molecules to enhance the vigour of the combustion.

Pilot Injection

Pilot injection of diesel fuel as a primary means of providing a flame kernel is predicted to be sufficient to ignite directly injected natural gas. However, the profile of flame front for combustion of the gas is the main parameter to bontrol the efficiency and emissions output.

Glow Plug Ignition

A glow plug appears to be quite promising in leading to autoignition in natural gas. It is very similar to a liquid-fuelled, directed, spark-ignited engine or a spark-assist diesel engine.

Plasma Jet Ignition

The concept in this particular type of ignition scheme is borrowed from the technology of the torch ignition engine, in which, charge separation may be achieved by using a pre-chamber for a rich mixture

of fuel and air and the main chamber for a weak mixture. The injection system can be used to supply fuel, Benson and Whitehouse [11]. By the creation of a plasma jet of fuel (diesel fuel or just natural gas), or air or both air and fuel in a prechamber, while injecting natural gas into the combustion chamber, natural gas will be ignited by the impingement and interaction with the second jet, i.e., the plasma jet. This ensuing plasma jet is rich, consisting of chemically active species and would create a vigorous reaction within the primary fuel/ air mixture. This plasma jet can be attained via electrical discharge in air, fuel, or air/fuel. Under this phenomenon, combustion duration can be greatlh reduced, hence, improved combustion process will result.

Laser Ignition

Instead of using electrode spark plug, a laser beam can be utilised to introduce an intense electric field breakdown in the combustion chamber to ignite the natural gas as it is injected. In this case, a laser beam is focused inside the combustion chamber to create a region of such intense electric field that breakdown of the stable species of fuel occurs.

CONCLUSIONS

Based on hypergolic combustion, a direct ignition incoporating glow-plugignition-assist system, the dream of achieving 'diesel like' performance in a conventional natural gas engine is viable through the means of activation of natural gas as mentioned above. This will not only reuslt in improved power output and thermal efficiency, it also permits the natural gas engine to take full advantage of future advanced diesel engine improvements such as the application of ceramic liner for combustion chamber insulation. With the advent of the technology, low heat rejection diesel engine natural gas as an alternative fuel has become possible offering the advantages of burning low cetane fuel.

With hyprgolic combustion, the combustion event will be completely different from that of spark-ignition or compression-ignition, hence, it is logical to suspect that the injected fuel pattern, heat transfer profile and thus the combustion shape itself should be different.

At this stage, sufficient data are not available to further determine and compare as to which concept,/scheme is superior from the engineering point of view.

RECOMMENDATIONS

1.Each of the scheme / approach as well as their amalgam of differnt schemes or 'hybrid' system consisting of the combination of the above mentioned schemes should be experimentally evaluated to determine if they are technically viable to achieve autoignition in the natural gas engine

2. Improved air turbulence is likely to provide the desired improvement in combustion.

3. As natural gas has high resistant to 'spark-knock', therefore, high compression ratios can be attempted, giving high thermal efficiency, similar to the comparable diesel engine, which in turn reduces the fuel consumption.

4. Multiple plugs with different configurations can be tried on the diameter of each cylinder to achieve optimal combustion rates, and leading to higher efficiency.

5. Generally, a three-way catalytic converter can be fitted to a natural gas engine to cut down the amount of carbon dioxide, carbon monoxide,oxides of nitrogen and nonmethane hydrocarbons in the exhaust emission.

6. By the use of a small prechamber with each cylinder that burns a rich mixture of natural gas and air, which is ignited by a spark plug, and on the other hand, the main portion of the combustion chamber burns a lean mixture, which is ignited by the burning gases in the prechamber, combustion will occur at lower temperatures than usual. Thus reduction of the nitorgen oxides could be achieved.

REFERENCES

1.Crane, M.E. and King. s.R., Enission Reductions Through precombustion Chanfuer Design in a Natural Gas, Lesn Burn Engine, Transactions of the ASME, VoL. 714, pp. 446-452,July 1992.

2.Natural Gas Training Manual, British Columbia Institute of rechnology, pp.9,7991.

3.Haddad, Sam and Watson, Neil, Prirciples and performnnce in Diesel Engineering, Ellis Horwood Limited, pp. 59-60,1984.

4.Richards, 8.G., Direct Gns Injection witlt Glozu plttg lgnition, Gas Research Institute, Annual Report, GRI-88 / 0271, 1988.

5.Taylor, C.F., Internal Combustion Engine in Tlrcory ttnd practice, The M.I.T. press/ pp. 43,1966.

6.Hoppie, L.O., The lnfluence of Initial Fuel Tentperatw.e on Ignition Delay, SAE Paper No. 820356, 1982.

7.Scharnweber, D.H. and Hoppie, L.O., Hypergolic Corubustion in an Interntrl Conrbustion Engine, SAE Paper No. 850089, 1985.

8.Heywood, J.8., Internal Combustion Engine Fnndamentals, McGraw-Hill Intrnational Editions, pp. 462-464, 1988.

9.Strehlow, R.A., Cortbustion" Fundanrcntal, pp. 778,248,408,7985.

10. Houseman,]. and Cerini, D., On-Board Hydrogen Generntor for n partial Hydrogen Injection Internal Contbttstiorr Engine, SAE paper No. 740600, 7974

11. Benson, R.s. and whitehouse, N.D., Internnl Combustiort Engines, Pergamon Press, pp. 22, 1979.

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Computation Studies of Aii Flov/ in Two Dimensional Coaxial Pipe

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ABSTRACT

The present study conducted a compurer simulation of a turbulent flow in a two co_axiat iipes with varying pipe diarneter and entrance displacement (AL). The entrained flow characteristics together with the effect of the temperature of the entrancJ flow inlet were studied with variation of AL. The numerical method employs differencing scheme for integrating the continuity equation and energy equation. An equation k-e turbulent rnodel was used to simulate the turbulent transport quantities. A Fluent CFD package was used to produce 2-D predictions of the flow pattern. Results show that increasing the air entrained temperature until 600 K will decrea-se the air flow rate and the diameter ratio of 2.gl has the highest entrained flowrate.

Key words: Coaxial pipes, k-e turbulent modet, CFD modeling, 2D Jlow, entrance displacement, air ejector.

INTRODUCTION

Ejectors are apparatus which entrained the row pressure gas by high pressure gas interchanging the momentum_of high speed driving. gas discharged through a nozzle (or nozzles) with momentum of low piessure entraine-d -gu, urouna'rt" -ariuing jet (or jets). Ejectors usually consists ofa nozzie (or nozzles) and a diffuser. That is, it have no moving part such as-a rotor or piston' one of the merits of ejector is that it is simpre in construction and it could compressed a large flow rate of driving gas for the-small and simple size ejector design as shown on Figuie l.

There are other names for the system which have the same working principles, such as jet pump, injector and eductor. The general name for these system is jet appartus. The names above mentioned are distinguishid according to the kinds states and properities such as compressibility (gas, riquid, mixture of liquid and solid) of the driving and entrained fluids. Based on th, nor*ul ejector design the present study is to propose some data for the design of the subsonic air in a coaxial pipe connguration.

A research on ejector by Johannesen (1951) reported that the features ofejector driven by and entraining compressible fluids. These include the aerodynamics of flow in the actuating nozzle, rnixing chanrber and the supersonic and subsonic diffusers. In general, the emphasis are made on the steam driven air ejector and experiments were u-,idr on such ejlctor; covering a range ofgeometrical proportions and pressure ratios.

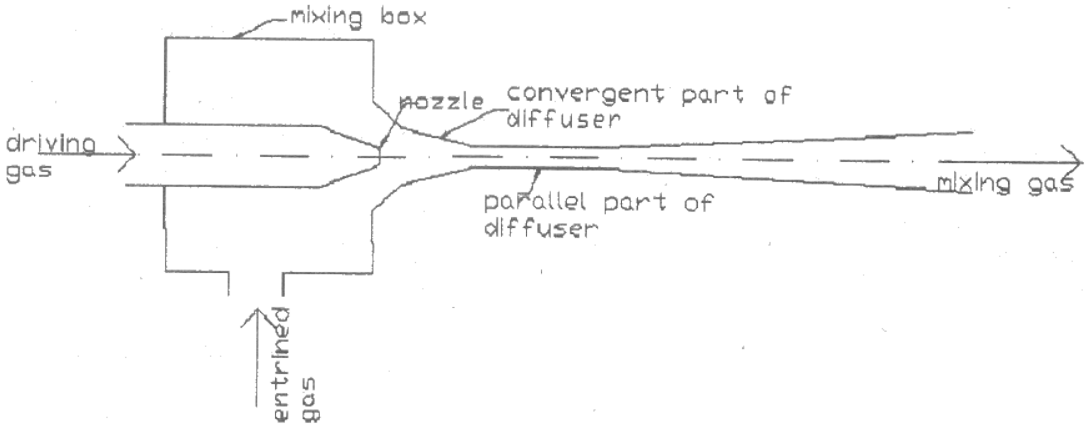


Figure 1. Ejector configuration

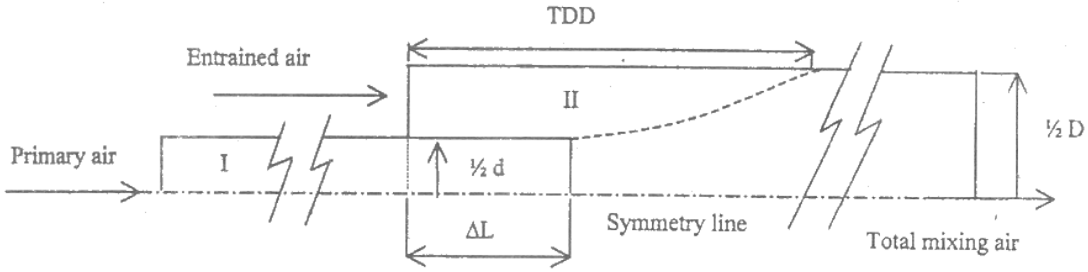
A one dimensional method of analysis of air ejector is presented by Keenan et al, (1950). The analysis considers mixing of the primary and secondary streams at constant pressure and constant area and it is concluded that better performance will result from constantpressure mixing. It should be noted, however, that while experimental verification of the analysis constant area is good, the lack of a suitable design of constant-pressure mixing chamber has prevented from demonstrating the calculated supermacy of this ejector design.

There are many of the previous paper deal with single and multiple nozzle ejector, In the implementation of these design should have a high air compression () 2 bars) to have a sufficient entrained air. In the present work it will be developid by anoiher type of a simple ejector in the form a pair of coaxial pipe which has a- purpbse to work on low air compression (S 2 bars)' The present study is objected to so-called an ejector which deals with compressible gases as the driving gas and entrained gas. One of tlre merits is that it could use a blower with low power and entrained gas could occur which matches a specially fi xed application.

THE AIR EJECTOR MODEL

Figure 2 shows the air flow and configuration of the coaxial pipes. The primary air or driving air is introduced into the inlet pipe I which acts as nozzle function, It is to be expected due to pressure drop at outlet pipe I occur the entrained air flow in pipe II which acts as diffuser function.

In this model the length of driving pipe or pipe I is 665 mm and the length of entrain ejector pipe is 650 mm. The pipe entrance displacement start from from AL=50 mm (0.050 m) and it increased until AL= 420 mm (0.420 rn). Diameter Ratio (DR) means here the diameter ratio of entrain ejector pipe to the drive pipe. The different size of entrain ejector diarneter are varied while the drive pipe diameter is constant. The Diameter Ratio (DR) variation are 1.63,2.34,2.62and 2.81.In this model is assumed that there is no friction on the surface pipe. Therefore the calculation regarding to the wall roughness is negligible.



I : driving pipe; ΔL : pipe entrance displacement; TDD : Touch down distance
II : entrained pipe; D,d : entrain and drive pipe diameters; DR: Diameter Ratio (D/d)

Figure 2. Air flow and ejector - drive pipes

GOVERNING EQUATIONS

The physical model considered in this work are shown schernatically in Figure 2. This configuration is a two dimensional of co-axial pipe with turbulent flow in the two pipes. The governing Reynolds-averaged equations for steady-state turbulent flow of twodimensional (Fluent User's Guide, 1997; Shaw, 1998; Patankar, 1980) are as followed:

Continuity Equations:

a. incompressible fluid:

$$\frac{\partial U}{\partial X}+\frac{\partial V}{\partial Y}=0 \qquad (1)$$

b. compressible fluid:

$$\frac{\partial \rho}{\partial t}+\frac{\partial(\rho U)}{\partial X}+\frac{\partial(\rho V)}{\partial Y}=0 \qquad (2)$$

Momentum Equations:

a.
$$\rho\frac{\partial U}{\partial t}+\rho U\frac{\partial U}{\partial X}+\rho V\frac{\partial U}{\partial Y}=-\frac{\partial P}{\partial X}+\frac{\partial}{\partial X}(\nu\frac{\partial U}{\partial X})+\frac{\partial}{\partial Y}(\nu\frac{\partial U}{\partial Y})+\frac{\partial}{\partial X}(-\overline{u'u'})+\frac{\partial}{\partial Y}(-\overline{u'v'}) \qquad (3)$$

b.
$$\rho\frac{\partial V}{\partial t}+\rho V\frac{\partial V}{\partial X}+\rho V\frac{\partial V}{\partial Y}=-\frac{\partial P}{\partial Y}+\frac{\partial}{\partial X}(\nu\frac{\partial V}{\partial X})+\frac{\partial}{\partial Y}(\nu\frac{\partial V}{\partial Y})+\frac{\partial}{\partial X}(-\overline{u'v'})+\frac{\partial}{\partial Y}(-\overline{v'v'}) \qquad (4)$$

The two equations, Eq. (3) and (4) derived from Newton's Second Law, describe the conservation of momentum in the flow also known as the Navier-Stokes equatibns. The terms on the left-hand side of each of these equations describe acceleration term, the second and third terms being the convection terms, then the right hand side terms come from the pressure gradient in the flow and the effects of viscosity and the last two terms that so called the Reynold stresses represent the model to account for the effects of turbulence. These momentum equation governs the time-averaged properties of the flow.

Theturbulence Models

To solve Equations (l) to (4), a turbulence model for the turbulent transport quantities has to be specified. In the present work, the standard k-e model (Fluent user's Guide,1997; Shaw, I 998; Launder and Spalding, 1972) based on Boussineq hypothesis is adopted. The local mean state ofturbulence can be characterized by the turbulent kinetik rnergy k and its dissipation rate e according to:

$$\rho \overline{u_i' u_j'} = \rho \frac{2}{3} k \delta_{ij} - \nu \left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) + \frac{2}{3} \nu \frac{\partial u_i}{\partial X_i} \delta_{ij} \quad (5)$$

where

$$k = \frac{1}{2} \sum_i \overline{u_i'^2} \quad (6)$$

$$\nu = \rho C_{\upsilon l} \frac{k^2}{\varepsilon} \quad (7)$$

$$\varepsilon = k^{2/3} / l_\varepsilon \quad (8)$$

Transport Equation For K And e

The values ofk and e Equation (7\ are obtained by solution ofconservation energy:

$$\frac{\partial}{\partial t}(\rho k) + \frac{\partial}{\partial X_i}(\rho u_i k) = \frac{\partial}{\partial X_i}\left(\frac{\upsilon}{\sigma_k} \frac{\partial k}{\partial X_i}\right) + G_k + G_b - \rho \varepsilon \quad (9)$$

$$\frac{\partial}{\partial t}(\rho \varepsilon) + \frac{\partial}{\partial X_i}(\rho u_i \varepsilon) = \frac{\partial}{\partial X_i}\left(\frac{\upsilon}{\sigma_\varepsilon} \frac{\partial \varepsilon}{\partial X_i}\right) + C_{1\varepsilon} \frac{\varepsilon}{k}(G_k + (1 - C_{3\varepsilon}) G_b) - C_{2\varepsilon} \rho \frac{\varepsilon^2}{k} \quad (10)$$

where

$$G_k = \upsilon \left(\frac{\partial u_j}{\partial X_i} + \frac{\partial u_i}{\partial X_j} \right) \frac{\partial u_j}{\partial x_j} \quad (11)$$

$$G_b = -g_i \frac{\upsilon}{\rho \sigma_h} \frac{\partial \rho}{\partial X_i} \quad (12)$$

The Effects of Turbulence on Heat Transfer

When the heat is added to a fluid and the fluid density with temperature flow can be induced due to the force of gravity acting on the density variations. Such flows are termed natural-convection (mixed-convection) flows. The importance of buoyancy forces in a mixed convection flow can be measured by the ratio of Grashof and Reynolds number:

$$\frac{G_r}{R_e} = \frac{\Delta \rho g h_c}{\rho \upsilon^2} \quad (13)$$

When this ratio approaches or exceeds unity, the flow should have a strong buoyancy to the flow (Fluent User's Guide, 1997).

In this simulation, there are some correlation between the turbulence flow and the fluid flow temperature. This correlation is based on the momentum equation and again the Boussineq model is adopted. For simpli$ing the momentum equation, it will perform the flow in y-direction (Shaw, 1989):

$$\frac{\partial V}{\partial t} + U\frac{\partial V}{\partial X} + V\frac{\partial V}{\partial Y} = -\frac{1}{\rho}\frac{\partial p}{\partial Y} + \frac{\mu}{\rho}\left(\frac{\partial^2 V}{\partial X^2} + \frac{\partial^2 V}{\partial Y^2}\right) + g\beta(T - T_f) \quad (14)$$

Another theoretical correlation is based on the conservation of energy which can predicted the heat transfer process within the fluid and/or within solid in the model (Fluent User's Guide, 1997). The simulation model solve the energy equation in the form of a transport equation for the static enthalpy, h:

$$\frac{\partial}{\partial t}(\rho h) + \frac{\partial}{\partial X_i}(\rho u_i h) = \frac{\partial}{\partial X_i}\left(k_c + k_t \frac{\partial T}{\partial X_i}\right) - \frac{\partial}{\partial X_i}\sum_j h_j J_j + \frac{\partial p}{\partial t} + \tau_{ik}\frac{\partial U_i}{\partial X_k} + S_h \quad .(15)$$

Enthalpy h defined as:

$$h = \sum_j m\, h_j \quad (16)$$

where: $h_j = \int_{T_{ref}}^{T} c_{p,j}\, dT$

NUMERICAL METHOD

The commercial software package, Fluent version 4.4, produced by Fluent Inc. was used as the primary source code for the model. This package employs a control volume-based, finite difference solution technique to allow full characterization of the flow field. The Reynolds-averaged Navier-Stokes equations coupled with the Reynolds-averaged governing differential equations of continuity, energy, and species are solved in a discritized form. The standhrd k-e turbulence modelis employed. To obtain values at control volume interfaces needed for flux calculations, the power law interpolation scheme is utilised. The pressure-iinked continuity and momentum equations are solved using the Serni-lmplicit Method for Pressure-Linked Equations Consistent (SIMPLEC) solution algorithm. Specific details regarding convergence parameters such as multi-grid and under relaxation factors are available in

Fluent. The convergence criterion is specified as the relative difference of every dependent variable between iteration steps being smaller than 10-6. The pipe form in the iresent work is assumed as circular form.

CONDITIONS OF SIMULATION

Sirnulations of subsonic air ejector was done according to obtain an optimum condition of air flowrate in a pair of coaxial pipe. The simulations are being carried out using a CFD software package. The conditions of simulation are as follows:

- subsonic and compressible air flow
- turbulence method: k- ε method
- boundary conditions:
  i) inlet pressure as primary air at inside-pipe: 2 bars,
  ii) inlet pressure where the secondary air or entrance air is induced: 1 bar (atmospheric pressure),
  iii) inlet pressure at the outlet pipe where the exit of total air flow takes place: 1 bar (atmospheric pressure),
- 2 D flow analysing.

RESULTS AND DISCUSSIONS

Effect of Variation of Diameter Ratio on Air Velocity and Air Flowrate

Figure 3 to Figure 4 show the graphs of air velocity (AV) and air flowrate (AF) versus diameter ratio. Each of the Figures has two types of curve, namely, primary (driving air) and entrained air flow. For primary air flow is kept cpnstant. The displacement of entrance pipe has not much influence to the primary air flowrate curve due to the pressure intake is high enough compared to the pipe resistance, but for the entrained air flow there is a reduce both for air velocity and flowrate. The increasing of pipe displacement means increasing of flow path. Therefore the area of entrained air flow is more restricted. It means the flow path for the entrained air has more resistance. From the graph that the highest entrained air flow is achieved at 0.05 m where this distance is on the first drive pipe displacement.

Figure 5 and 6 show the effect Diameter Ratio variation on entrained air flowrate. Figure 5 plotted curves in which the increasing of DR has caused the entrained mass flowrate also increased.

To describing both of these conditions it is necessary the continuity equation is to be implemented:

$$\frac{\partial}{\partial t}\int \rho dv + \int \rho v.dA = 0 \qquad (17)$$

Assumed the flow is steady, therefore the first term is zero ($\frac{\partial}{\partial t}\int \rho dv = 0$), hence:

$$\int \rho v.dA = 0 \quad (18)$$

Since there is no flow through the wall and the flow is compressible, the equation (18) changed into:

$$\rho_1 V_1 A_1 = \rho_2 V_2 A_2 = \dot{m} \quad (19)$$

According to the Equation (19), when DR increase means the area of flow is also increase, so the volumetric flow rate m is also increase as in Figure 5 and air flow velocity is decreased as shown in Figure 6,

Effect of Diameter Ratio and Drive Pipe Displacement on Touch Down Distance (TDD)

Figure 7 and 8 show the graphs of the effect of both diameter ratio and drive pipe displacement (AL) on Touch Down Distance (TDD). In these graphs three curves were plotted, Diameter Ratio (DR) 1.63,2.34 and 2,62 have created the touch down distance, except for the highest displacement LL 0.420 m, In this displacement there is no touch down point on the entrain pipe and it has created turbulence effect on the outlet section of the entrain pipe since TDD is beyond the length of the pipe. Consequently, this condition could cause a flow resistance. The TDD have a tend to increase for higher DR. Figure 7 show the smallest TDD is achieved by the smallest displacement. Also increasing the entrance displacement the TDD increased also as in Figure 8, It should be that there is a correlation between the entrained mass air flowrate and TDD. It means the entrained flow path, since it is affecting reduction of the mass air flowrate, influence the TDD significantly.

Graphs show that higher DR could influence higher entrained mass and increasing of this mass flowrate 'has affected TDD to increase flowrate as in Figure 7. Due to the DR variation has a correlation with flow area which has influenced mass air flowrate, thus it could be deducted the flow area is a factor which has also influenced TDD. Graphs on Figure 8 show that higher AL has increased TDD. It means that TDD is affected by air flow path between the drive pipe and entrain pipe.

Effect of Variation of Gas Temperature on Entrained Air Flowrate and Air Velocity Figure 9 to Figure 12 show graphs of entrained air flowrate and air velocity at different ambient temperature of the inlet secondary pipe (entrance pipe). These figures show that the increasing of air temperatures of 300, 500 and 600 oC at different Diameter Ratio DR of 1.63,2.34,2.62 and 2.81 which have influenced the

variation of air flowrate and velocity. The increasing of gas temperature affected the increasing the entrained air velocity, but there is a decreasing of entrained air flowrate.

The properties of air at atmospheric pressure (Holman, 1993) are listed in tables i,e, value of air density and specific heat which are correlated with different air temperature. Increasing ofair temperature affect the decreasing ofair density and increasing of specific heat, There is a correlation at the gas law (Streeter, 1983; Fluent User's Guide, 1997; Holman, 1993) between air density and the temperature as follow:

$$\rho = p_{op}/RT_a \qquad (20)$$

If the temperature increase, the air density should be decrease. Based on the continuity equation, namely, Equation l9 the decreasing of density affected the air flowrate, it means the air flowrate is decreased. Another explanation is based on energy equation as in Equation 16. Iricreasing the air temperature is effecting the increment of specific heat capasity value co (Cengel, 1999). To balancing the total heat capacity of the system the mass flowrate should decrease.

Increasing of the temperature effecting also the air velocity. When the system changes from some initial velocity vt to a velocity v2, the corresponding change in kinetic energy is (Holman, 1993):

$$\Delta E = KE_2 - KE_1 = \int_{v_1}^{v_2} \frac{m}{g_c}\, v\, dv = \frac{1}{2g_c} m\,(v_2^2 - v_1^2) \qquad . (21)$$

The conservation energy shows that when the temperature in a system is increbsing then the heat Q in the system increase also. It means there is a change in the internal energy AE. The work W is altered according to the conservation energy by the amount of heat energy added Q, thus:

$$Q + W = \Delta E \qquad . (22)$$

It indicates that
Energy added to system = accumulation of energy in the system.
At the present work there is no work W or W = 0, thus Equation 22 becomes

$$Q = \Delta E \qquad (23)$$

where $Q = h_i = \int_{T_{ref}}^{T} c_{p,i}\, dT$.

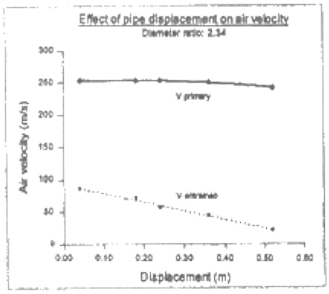


Figure 3

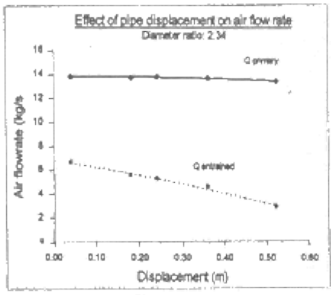


Figure 4

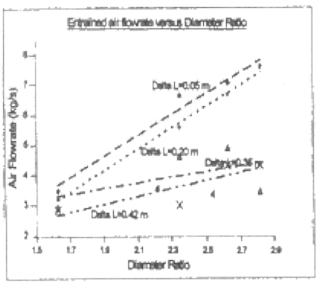


Figure 5

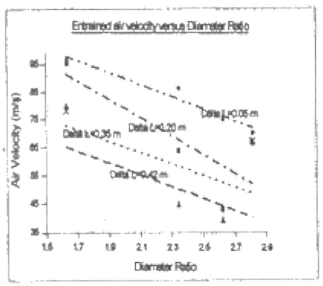


Figure 6

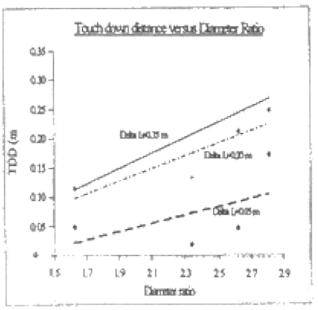


Figure 7

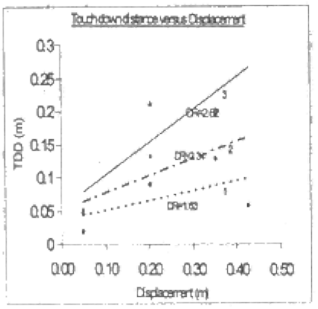


Figure 8

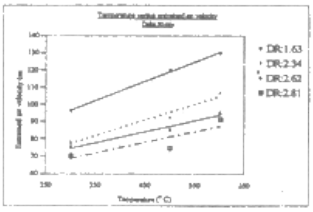


Figure 9

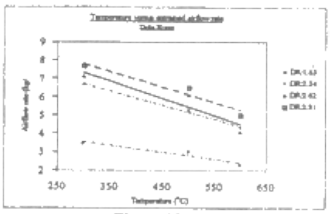


Figure 10

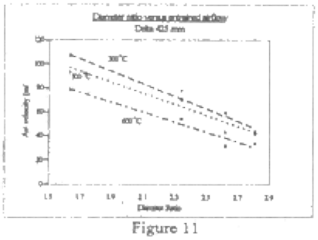


Figure 11

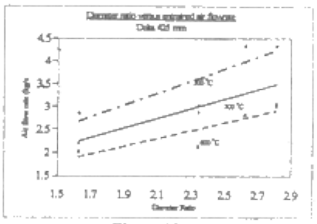


Figure 12

CONCLUSIONS

2-dimensional cold-flow simulations have been performed for the model geometry of coaxial pipes, It was demonstrated that the variation of drive pipe diameter and entrance pipe displacement did play a major role of entrained mass flow rate and Touch Down Distance. Flow simulations result show that the Diameter Ratio of 1.63 has the smallest flow area, as the consequence, there is the lowest entrained air flowrate, For each diameter ratio increasing of Diameter Ratio will increase the air flowrate, but the

entrance pipe displacement will decrease the entrained rate. It has indicated that there is a correlation between entrained mass flow rate and Touch Down Distance. Thus, if the entrained air flow is low then Touch Down Distance is short, The result also indicate that Touch Down Distance is influenced by two conditions i.e., displacement due to pipe resistant, and diameter ratio as well as flow area which has relationship with mass air flowrate,

The gas law and energy conservation law are important equations to explain the correlation of air velocity and air flowrate with the variation of gas temperature. Increasing the temperature until 600 K will decrease the air flow rate and the diameter ratio of 2.81 has the highest entrained fl owrate.

NOMENCLATURE

A	inflow area, m^2
C_{v1}	constant of proportionality (= 0.09)
$C_{1\varepsilon}$	empirical constants (=1.44)
$C_{2\varepsilon}$	empirical constants (=1.92)
dv	element of volume, m^3
h_c	average heat transfer coefficient, W/m °C
i, j	direction of flow, -
g	standard gravity acceleration (= 9.807 m/s^2)
$g\beta(T-T_f)$	additional term for buoyancy
J_j'	diffusion flux of the species j'
G_k	the rate of production of turbulent kinetic energy, kg/m s^3
G_b	generation of turbulence due to buoyancy, kg/m s^2
G_r	Grashof number
k	turbulent kinetic energy, m^2/s^2
k_c	molecular conductivity, W/m °C
k_t	effective conductivity due to turbulence transport ($k_t=c_p\mu_t/Pr_t$)
l_ε	mixing length scale, m
$\dot{m}$	volumetric flow rate, kg/s
P	mean pressure, Pa
p_{op}	operating pressure, Pa
R	gas constant, 8.31434 kJ/kmol K
R_e	Reynold number, -
S_h	term including heat of chemical reaction, any interphase exchange of heat, any other volumetric heat sources, -
T_a	ambient temperature, K
T_f	Temperature reference, K
t	velocity flow time, s
U,V	mean velocity components in the direction of (X,Y), m/s

v	inflow velocity, m/s
u',v'	fluctuating components velocity in the direction of (X,Y), m/s
β	coefficient of volume expansion, 1/K $= \frac{1}{v}(\frac{\partial v}{\partial T})_p = -\frac{1}{\rho}(\frac{\partial \rho}{\partial T})_p$
$\overline{u'u'}$	Reynold stresses, m^2/s^2
$\overline{u'v'}$	Reynold stresses, m^2/s^2
$\overline{v'v'}$	Reynold stresses, m^2/s^2
ρ	density, kg/m^3
δ_{ij}	wall shear layer thickness, m
σ_k	"Prandtl" numbers governing the turbulent diffusion of k and ε, (=1.0)
σ_ε	Prandt numbers governing the turbulent diffusion of k and ε, (=1.3)
μ	turbulent viscosity is proportional to the product of a turbulent velocity scale and length scale, $N.s/m^2$
υ	kinematik viscosity, m^2/s
ε	distribution of dissipation rate of k, m^2/s^3

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LITERATURES

1.Yamamoto, F., Performance of Subsonic Air Eiector with multiple nozzles, Memoires of The Faculty of Engineering, Fukui University, Vol. 30 No.1 1982.

2.Streeter, Y,L., Fluid Mechanics, McGraw-Hill International Book Company, 1983.

3.Fluent User's Guide, Fluent 4.4, Yo1. l, 3 and 4,May 1997 .

4.Holman, J.P ., Thermodynamics, McGraw-Hill Intemational Book Company,1993

5.Johannesen, N.H, Ejector theory and experimenrs, Danish Acad. Tech. sci. No. l.176p(1951)

6. lftry", w.C., Jannaf rhermochemicql Table - An Introduction to combustion, I 993.

7.Keenan, L. H., Neumann, E.P. and Lustwerk,F.p., An investigation of ejector design by analysis and experimenr J. Appl. Mech., 17, 3 (195-0).

8.Shaw, C. T, Computational Fluid Dynamic, Longman, 19g9.

9.Panton, R. L., Incompressible Flow, JoIn Willey & Sons, 19g4.

10.Patankar, s.Y., Numerical Heqt Transfer and Fruid Flow, Hemisphere publishing Corporation, 1980.

11. Launder, B. E., and Spalding, D.8., Lectures in Mathematical Models of Turbulence,Academic Press, London, England, 1972.

12.Hwang, R. R. and Chow, Y . C., Computation of turbulent flow over two dimensional surface-mounted rib, proceedings of t he 6'h Flow Modelling and Turbulence Measurements, Florida, 1996.

l3. cengel, Yunus, Heat Transfer: A practical Approach, McGraw Hill, 1998.

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A Classical Generalized Variational Principle for Pseudo-State Thermoelasticity of Piezoelectric Materials

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ABSTRACT

It is very difficult to establish a classical variatinal Principle (not Gurtin-typeneot involving convolutions) for thermopiezoelasticity. however, the semi-inverse method proposed by He appears to be one of the best and most convient ways to establish variational principles for the physical problems. By such method, a classical generalized variations has been established for the Pseudo-static thermoelasticity of piezoelectric matirials.

Keywords: Variational Theory, Semi-inverse Method, thermopiezoelectricity

INTRODUCTION

Recent interest in piezoelectric materials stems from their potential applications in intelligent structural systems. A comprehensive list of works in this area may be found in[1~4] and the references cited thereby. The rapid development of computer science and the finite element applications reveals the importance of searching for a classical variational principle for the thermopiezoelectricity, which is the theoretical basis of the finite element methods[5] and meshfree methods[6].

A GENERALIZED VARIATIONAL PRINCIPLE

Though it is easy to establish a Gurtin-type functional(involving convolutions), it is very difficult to construct a classical variational model due to the strongly coupled constitutive relations and the terms of the first-order time-derivatives involving in the heat conduction equation. As the author knows, there exist no such classical variational models for the thermopiezoelectricity, the semi-inverse method[6~9] that we are proposing appears to be one of the best and most convenient ways to establish variational principles for the physical problems. By such method we obtained following generalized variational principle with 9 kinds of independent variations (stress σ_{ij}, strain γ_{ij}, displacement u_i, temperature θ, heat flux q_i, electric displacement D_i, electric field E_i, electric potential Φ and entropy S)

$$J(\sigma_{ij},\gamma_{ij},u_i,\theta,q_i,D_i,E_i,\Phi,S)=\int_{t^{(n-1)}}^{t^{(n)}}\int L\,\mathrm{d}V\,\mathrm{d}t+IB$$

where

$$\begin{aligned}L=&\sigma_{ij}\gamma_{ij}-\tfrac{1}{2}\sigma_{ij}(u_{i,j}+u_{j,i})\\&+\gamma_{ij}(-\tfrac{1}{2}a_{ijkl}\gamma_{kl}+e_{mij}E_m+b_{ij}\theta)+f_iu_i\\&+\theta(\tfrac{1}{2}c\theta_0\theta+c_iE_i-t'q_{i,i}-\alpha)\\&+\tfrac{1}{2}(K_{ij}\tau+t')q_iq_j-\beta q_i-E_iD_i+\tfrac{1}{2}E_i\varepsilon_{ij}E_j+D_i\Phi_{,i}\\&+\lambda(\rho S-c\theta-b_{ij}\gamma_{ij}-c_iE_i)^2,\end{aligned}$$

$$\begin{aligned}IB=&\int_{A_1}\sigma_{ij}n_j(u_i-\bar{u}_i)\,\mathrm{d}A+\int_{A_2}\bar{p}_iu_i\,\mathrm{d}A\\&-\int_{A_3}(\Phi-\bar{\Phi})D_in_i\,\mathrm{d}A-\int_{A_4}\bar{D}_n\Phi\,\mathrm{d}A\\&+\int_{A_5}t'q_in_i\bar{\theta}\,\mathrm{d}A+\int_{A_6}t'\theta(q_in_i-\bar{q}_n)\,\mathrm{d}A,\end{aligned}$$

where $t'=t-t^{(n-1)}$, $t\in[t^{(n-1)},t^{(n)}]$, λ is a nonzero constant, α and β are written in the forms

$$\alpha=c\theta_0\theta^{(n-1)}+b_{ij}\gamma_{ij}^{(n-1)}+c_iE_i^{(n-1)}+t'\rho Q \quad\text{and}\quad \beta=K_{ij}\tau q_i^{(n-1)}.$$

$A_1+A_2=A_3+A_4=A_5+A_6+A_7=A$ covers the total boundary surface.

Making the above functional stationary, we obtain following Euler equations

$$\delta u_i:\quad \sigma_{ij,j}+f_i=0 \tag{1}$$

$$\delta S:\ \rho S=c\theta+b_{ij}\gamma_{ij}+c_iE_i \tag{2}$$

$$\delta\gamma_{ij}:\ \sigma_{ij}-a_{ijkl}\gamma_{kl}+e_{mij}E_m+b_{ij}\theta-2\lambda b_{ij}(\rho S-c\theta-b_{mn}\gamma_{mn}-c_mE_m)=0 \tag{3}$$

$$\delta E_m:\ e_{mij}\gamma_{ij}+c_m\theta-D_m+\varepsilon_{mj}E_j-2\lambda c_m(\rho S-c\theta-b_{ij}\gamma_{ij}-c_iE_i)=0 \tag{4}$$

$$\delta\sigma_{ij}:\ \gamma_{ij}=\frac{1}{2}(u_{i,j}+u_{j,i}) \tag{5}$$

$$\delta\Phi: \quad D_{i,i} = 0 \tag{6}$$

$$\delta D_i: \quad E_i = \Phi_{,i} \tag{7}$$

$$\delta\theta: c\theta_0\theta + b_{ij}\gamma_{ij} + c_i E_i - t' q_{i,i} - c\theta_0\theta^{(n-1)} - b_{ij}\gamma_{ij}^{(n-1)} - c_i E_i^{(n-1)} - t'\rho Q$$
$$-2\lambda c(\rho S - c\theta - b_{ij}\gamma_{ij} - c_i E_i) \tag{8}$$

$$\delta q_i: t'\theta_{,i} + (K_{ij}\tau + t')q_i - K_{ij}q_i^{(n-1)} = 0 \tag{9}$$

and following boundary conditions

$$u_i = \bar{u}_i \qquad (\text{on} \quad A_1) \tag{10A}$$
$$\sigma_{ij}n_j = \bar{p}_i \qquad (\text{on} \quad A_2) \tag{10B}$$
$$\Phi = \bar{\Phi} \qquad (\text{on} \quad A_3) \tag{10C}$$
$$D_i n_i = \bar{D}_n \qquad (\text{on} \quad A_4) \tag{10D}$$
$$\theta = \bar{\theta} \qquad (\text{on} \quad A_5) \tag{10E}$$
$$q_i n_i = \bar{q}_n \qquad (\text{on} \quad A_6) \tag{10F}$$

The equations (3) ,(4), (8) and (9), in view of the equation (2), can be re-written down as follows

$$\sigma_{ij} = a_{ijkl}\gamma_{kl} - e_{mij}E_m - b_{ij}\theta \tag{2'}$$

$$D_m = e_{mij}\gamma_{ij} + c_m\theta + \varepsilon_{mj}E_j \tag{3'}$$

$$c\theta_0\frac{\theta - \theta^{(n-1)}}{t - t^{(n-1)}} + b_{ij}\frac{\gamma_{ij} - \gamma_{ij}^{(n-1)}}{t - t^{(n-1)}} + c_i\frac{E_i - E_i^{(n-1)}}{t - t^{(n-1)}} = q_{i,i} + \rho Q \tag{8'}$$

$$\theta_{,i} = -K_{ij}\left(\tau\frac{q_i - q_i^{(n-1)}}{t - t^{(n-1)}} + q_i\right) \tag{9'}$$

When $t \to t^{(n-1)}$, we have

$$c\theta_0\frac{\partial\theta}{\partial t} + b_{ij}\frac{\partial\gamma_{ij}}{\partial t} + c_i\frac{\partial E_i}{\partial t} = q_{i,i} + \rho Q \tag{8''}$$

$$\theta_{,i} = -K_{ij}\left(\tau\frac{\partial q_i}{\partial t} + q_i\right) \tag{9''}$$

$$\tau \frac{\partial q_i}{\partial t} + q_i = -k_{ij}\theta_{,j} \qquad (9''')$$

where $\theta = T - \theta_0$, T is the temperature and θ_0 is the initial temperature , Q is the strength of the internal heat source, K_{ij} is the inverse of k_{ij}.

The obtained Euler equations satisfy all the field equations and boundary conditions of the thermoelasticity of piezoelectric materials.

CONCLUSION

Hereby we obtain a variational principle for the discussed problem, which might find some potential applications.

References

[1] Chandrasekharaiah D S.,A generalized linear thermo-elasticity theory for piezoelectric media, *ACTA Mechanica*, 71(1998), 39~49

[2] Maugin,G.A. (ed.) The mechanical behavior of electromagnetic solid continua, North-Holland, 1984

[3] Maugin,G.A Continuum mechanics of electromagnetic solids, Elsevier, 1991

[4] He,J.H., Liu,G.L & Weng,W.B. A generalized variational principle for coupled thermoelasticity with finite displacement, *Communications in Nonlinear Science and Numerical Simulation*, 3(4), 1998

[5] Liu,G.L., A new finite element with self-adapting built-in discontinuity for shock-capturing in transonic flow, *International Journal of Nonlinear Sciences and Numerical Simulation*, 1(1), 2000, 25-29

[6] He, J.H. Treatment Shocks in transonic aerodynamics in Meshless Method Part I Lagrange multiplier approach, *Int. J. Turbo & Jet-Engines*, 16(1), 1999, 19-26

[7] He,J.H. Generalized Hellinger-Reissner principle, *ASME J. Appl. Mech.* , 67(2), 2000, 326-331

[8] He, J.H. Inverse problems of determining the unknown shape of oscillating airfoils in compressible 2D unsteady flow via variational technique, *Aircraft Engineering & Aerospace Technology*, 72(1), 2000: 18-24

[9] He, J.H., A classical variational model for micropolar elastodynamics, *International Journal of Nonlinear Sciences and Numerical Simulation*, 1(2), 2000, 133-138

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