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G. R. Price
NOVA HUSKY Research Corporation; Canada

K. K. Botros
NOVA HUSKY Research Corporation; Canada

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G. R. Price and K. K. Botros
NOVA HUSKY Research Corporation
Calgary, Alberta, Canada

ABSTRACT

An experimental and numerical study of compressor channel valves was performed for low pressure air flows at a Reynolds number based on the inlet slot width (Re_S)~$2x10^4$. The objective was to investigate the variation of the effective valve flow and force areas with lift. The results showed that the variation in the effective flow area with valve lift is approximately linear. The numerically determined effective valve flow areas agreed within 15% of those obtained from experimental measurements. The numerical model was then used to determine the valve flow and force areas at operating conditions prevailing in the suction and discharge of a high pressure natural gas reciprocating compressor (Re_S~$4x10^5$). The effect of increasing the Reynolds number (Re_S), over the range considered, on the effective flow area was found to be small. The effective force area decreases rapidly at small values of valve lift and gradually increases at larger lifts.

NOMENCLATURE

A_{ef}	effective force area;
A_{ep}	effective flow area;
A_t	nozzle throat area;
C_D	nozzle discharge coefficient;
C_*	critical flow factor;
h	valve plate lift;
k	turbulence kinetic energy;
L	inlet slot length;
$\dot{m}$	mass flow rate through valve;
P	gas pressure;
R	gas constant;
Re	Reynolds number;
T	gas temperature;
w	inlet slot width;
ε	dissipation of turbulence kinetic energy;
γ	isentropic exponent;
ρ	gas density;

Subscripts

d	recovered downstream of valve;
max	full valve lift;
o	stagnation conditions;
s	based on inlet slot width;
u	upstream of valve;

INTRODUCTION

The use of computer simulations to predict the magnitude and frequency of pressure pulsations generated by reciprocating compressors has become widespread[1,2,3,13]. The suction and discharge valves of the compressor do not only modify the fluctuating flow produced by the piston motion, but also interact dynamically with the associated piping system. Thus, the influence of the valves should be taken into account in the simulations if accurate prediction of the pulsation levels is required, particularly at the higher harmonics of the compressor operating speed.

A one-dimensional quasi-steady state flow equation along with a second-order ordinary differential equation (O.D.E) describing the motion of the valve plate, (i.e. a concentrated mass point with a single degree of freedom spring-mass-damping model), are typically used to characterize the valve dynamics. This quasi-steady state assumption followed from extensive experiments on large scale models of valves by Boswirth[4,5,6] which led to a one-dimensional unsteady flow equation that takes into account the gas inertia and unsteady work exchange between the gas and the valve plate. The calculations based on this equation have shown that the steady flow equation generally gives acceptable results in the absence of valve flutter. Tilting of the valve plates can occur even in completely symmetric flows[7] and hence can not be properly represented by the single degree of freedom spring-mass-damping model. More sophisticated models of the valve dynamics exist[8] which consider the valve plate with more than one degree of freedom, but they require a more complete knowledge of the flow, force and spring characteristics of a valve than was obtained from this work. Consequently, the steady flow equation and the second-order O.D.E (spring-mass-damping) were considered adequate to model the valve dynamics of the low speed, high flow compressors used in gas transmission systems. In these equations, the effective valve flow and force areas are parameters that depend on the lift and Re_s for a given compressor valve.

An experimental study was conducted on a low pressure air test loop. Pressure differentials across the valves at several fixed increments of valve lift were measured to determine the variation in the effective flow area. A numerical analysis of flow through the valves was conducted using PHOENICS[9], a commercial fluid dynamics software, to obtain both the effective flow and effective force areas at conditions similar to the experiments. The performance of the numerical model was verified by comparing the effective flow areas obtained numerically to those determined experimentally. The numerical code was subsequently used to evaluate the variation in both valve parameters with lift for operating conditions prevailing at the suction and discharge of the high pressure (thus higher Re_s) natural gas reciprocating compressors in NOVA's gas transmission system.

GENERAL EQUATIONS

The effective valve flow area relates the mass flow rate through the valve to the upstream and fully recovered downstream pressures. Using the one-dimensional steady flow equation, the effective valve flow area is calculated from the following expression for subsonic conditions[10] :

$$A_{ep} = \frac{\dot{m}}{P_u\sqrt{\frac{2\gamma}{(\gamma-1)RT_u}}\sqrt{\left(\frac{P_d}{P_u}\right)^{\frac{2}{\gamma}} - \left(\frac{P_d}{P_u}\right)^{\frac{\gamma+1}{\gamma}}}} \tag{1}$$

The gas force exerted on the valve plate is the result of the normal pressure acting on both sides of the plate and the wall shear in the direction of valve lift. This force can be related to the overall pressure loss across the valve by an effective force area defined as:

$$A_{ef} = \frac{F_p}{P_u - P_d} \tag{2}$$

DESCRIPTION OF CHANNEL VALVE

A suction and a discharge channel valve were taken from a reciprocating compressor station in NOVA's gas transmission network. The valves are rated at 5500 kPa for dynamic differentials and 10,350 kPa for static differentials, which make them suitable for the high pressure, natural gas applications. A photograph of the suction valve used in this study is shown in Fig.1. The gas flow enters the valves through seven parallel rectangular slots of equal width, but with varying lengths.

Each slot seats a U-shaped plate that is actuated by a stiff leaf spring. The gas leaves the valve through eight staggered slots that are similar in appearance to the entrance slots. A cross-section of a single slot and the U-shaped plate with spring are illustrated in Fig.2.

EXPERIMENTAL SETUP

Only the effect of the valve lift on the effective flow area was investigated experimentally, while the effects on the effective force area were not considered due to difficulties in measuring the lifting force acting on the plates. Plate lifts were secured for the seven slots by means of precision spacers which gave 20, 40, 60, 80 and 100% of the full lift for each slot. A schematic of the experimental setup is presented in Fig. 3. Ambient air was drawn through the test section and the pressure distributions, both upstream and downstream of the valve, were recorded. A 40D section of 6" pipe upstream of the test section ensured the inlet profile into the valve was fully developed. Pressure measurements were taken at two locations upstream (0.131 and 0.08 m) and four locations downstream (at 0.02, 0.08, 0.181 and 0.284 m). Each set of taps consists of four equally spaced static pressure taps around the circumference of the pipe. Air was drawn through the system by a vacuum pump located downstream of the test section at a constant mass flow rate maintained by the sonic nozzle. The mass flow rate was measured with a 40mm calibrated sonic nozzle, based on the following equation:

$$\dot{m} = C_D C_* A_t \frac{P_o}{\sqrt{RT_o}} \tag{3}$$

A single pressure tap was located upstream of the nozzle in order to record the stagnation pressure used in the calculation of the mass flow rate from the above equation. The nozzles have been calibrated to an accuracy of +/- 0.5%. Pressures were measured with a large U-tube water manometer to an accuracy of 0.25%. The overall uncertainty in the effective flow areas is +/- 1.0%.

COMPUTATIONAL STUDY

Computational simulations were performed using PHOENICS (v1.5) to calculate the flow field through the channel valves at several different valve lifts. The code was run on an IBM 320H RISC workstation. The effective flow area was calculated from eqn.(1) using the numerically determined pressure field; the upstream and fully recovered downstream pressures. The lifting force acting on the valve plates was obtained by integrating the normal pressure acting on the front and back of the valve plate along with the wall shear stress in the direction of valve lift. The effective force area was then calculated using eqn.(2).

In this study the time-averaged forms of the Navier-Stokes and continuity equations were solved along with the standard k-ε model of turbulence[12] on a two-dimensional computational grid. The two-dimensional simplification was made as the length of the valve slots was large compared to the width ($14<L/w<19$ for the seven slots). Additionally, only half of one slot, from the center of the valve plate to the plane of symmetry of the flow produced by the adjacent slot, was modeled. This assumption is justified due to the symmetrical nature of the flow produced by the adjacent slots for the channel valve geometry. Finally, the flow through the valves was assumed to be isothermal and incompressible since the Mach number through the computational domain was below the compressible limit ($M<0.3$), except in the case of very small valve lifts ($M\sim0.4$) where the compressibilty was neglected.

Computational Grid

A body-fitted coordinate system was employed to generate a computational grid that adheres to the walls of the tapered slot in the valve and the rounded edges ($r\sim1$ mm) of the valve plate. The computational domain starts 14 mm upstream of the inlet into the valve slot, while the outlet of the domain was placed 46 mm downstream of the valve exit. The locations of the inlet and outlet boundaries were determined by trial and error in order to minimize the influence on the flow field in the vicinity of the valve plate and allowed the exiting flow to re-develop. A non-uniform grid was generated which is fine in the lift gap and becomes increasingly coarse away from the valve plate.

The grid contained 50 cells parallel to the slot width and 300 cells in the overall flow direction. The influence of grid size was investigated by solving the flow field through a finer grid (100x500 cells) at maximum plate lift for the suction channel valve. The effective flow and force areas were within 3% of that obtained with the coarser mesh. The computational grid used in this study for the suction channel valve at maximum lift is shown in Fig.4.

Boundary Conditions

Along the inlet boundary of the computational domain, a uniform velocity profile was specified based on the average velocity through all seven slots. The turbulent kinetic energy, k, and rate of dissipation of this energy, ε, were set to uniform profiles characteristic of the upstream flow. The sensitivity of the results to the inlet turbulent profiles is small. The no-slip condition at the walls was specified indirectly using the standard wall function approach[12]. At the outlet of the domain the gradient of the velocities and turbulent quantities were set to zero.

COMPARISION OF EXPERIMENTAL AND COMPUTATIONAL RESULTS

Low Pressure Air

The effective valve flow areas, calculated both numerically and from experiment, are presented in Figs.5 and 6.for the suction and discharge channel valves, respectively, at $Re_S \sim 2x10^4$. The effective flow area is normalized by the inlet slot area shown in Fig. 2 for each channel valve. The experimental and numerical data are fitted to third degree polynomials by least-squares regression. Both methods predict a direct linear relationship between the effective flow area and the valve plate lift at values of $h / h_{max} < 0.6$, i.e. the effective flow area increases with increasing lift. However, the numerical simulations predict a leveling of the effective flow area before maximum lift is obtained. The agreement between the two approaches is quite good, within a maximum of 15%. Although no direct comparision with literature is possible for this particular valve geometry and Reynolds numbers, there is general agreement in the tends shown here and results from other papers[14,15].

Computational Flow Field

The flow field in the valve lift gap for low pressure air in the suction channel valve at 100 and 20% of the maximum plate lift are shown in Fig.9. At maximum plate lift, a large recirculation zone, or separation bubble, develops along the inlet valve face in the plate lift gap. The bulk flow is deflected by the valve plate into the outlet slot. At small values of the plate lift (20%), the flow is accelerated and deflected 90 degrees in a relatively short distance. This suppresses the formation of the recirculation zone along the inlet face and a separation wake in the outlet slot as the high momentum jet cannot remain attached along the curvature of the valve plate. In this case, the Mach number of the flow in the gap is approximately equal to ~0.4, thus compressibility effects should be taken into account (which was neglected in the present work). The radius of the valve plate edge (r) was found to have an effect on the results. The radius was measured by a crude tracing technique and found to be ~1 mm which was used in all the cases presented here. Additional simulations showed that changing the curvature of the rounded edge ($0<r<2$ mm) could change the numerically calculated effective flow and force areas by ±5%.

High Pressure Natural Gas

The effective valve flow and force areas for high pressure natural gas, calculated from the numerical simulations, are presented in Fig.7 for the suction channel valve and Fig.8 for the discharge channel valve. The effect of increasing the Reynolds number of the flow through the valve to $Re_S \sim 4x10^5$ slightly increases the effective flow area for a specified plate lift. This is expected as the boundary layers in the lift gap become thinner with increasing Reynolds number and the losses are dominated more by inertia[14]. The effective force area is equal to the valve inlet slot area at zero lift. As the valve plate lift increases, the effective force area decreases because the negative pressure distribution along the outer edge of the valve plate overrides the positive pressures near the center. Eventually, as the lift increases further the negative pressures near the edges of the valve plates, needed to deflect the flow, are reduced and the effective force area increases. The numerical model predicts this trend, however the accuracy of the numerically determined lifting forces on the valve plates needs verification with measurements. The relative magnitude of the wall shear was less than 10% of the normal pressure force for the cases considered.

CONCLUDING REMARKS

Detailed flow field through reciprocating compressor channel valves were determined using a commerical code (PHOENICS). The computations were first performed on low Reynolds number flow through the valves, and compared with actual measurements obtained with a low pressure air test loop. Good agreement between the two results gave confidence in the computation technique, which was then applied to a higher Re at conditions prevailing in high pressure natural gas compression. It was demonstrated that the influence of the Re on the effective flow area in this study was rather small. The computation also revealed values for the effective force area at different valve lifts which is required for any valve dynamic and pulsation simulation.

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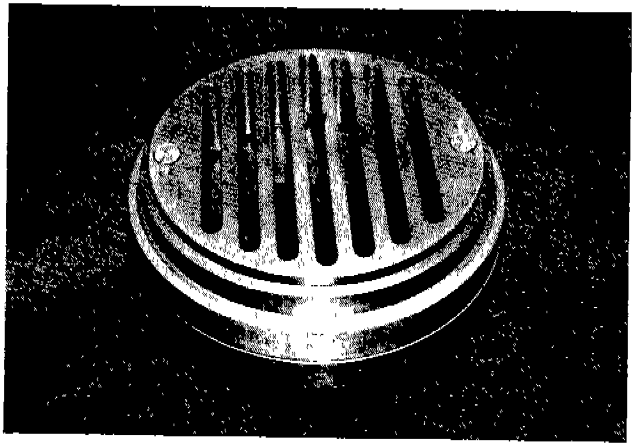

Fig.1. Photograph of Suction Channel Valve

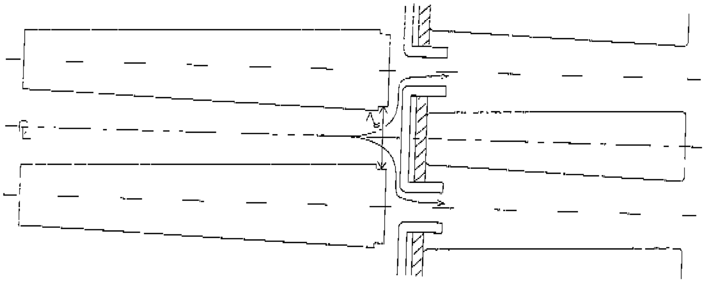

Fig.2. Cross–section of Channel Valve

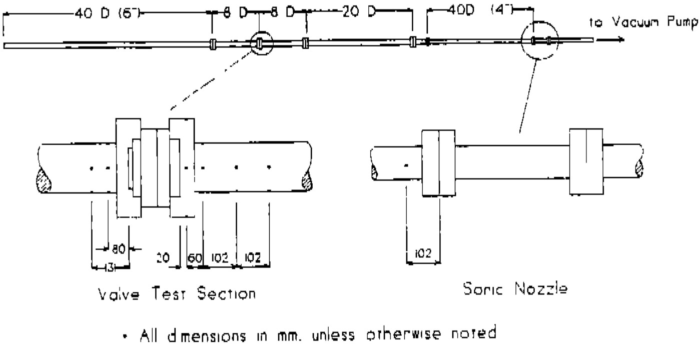

Fig.3. Schematic of Experimental Setup

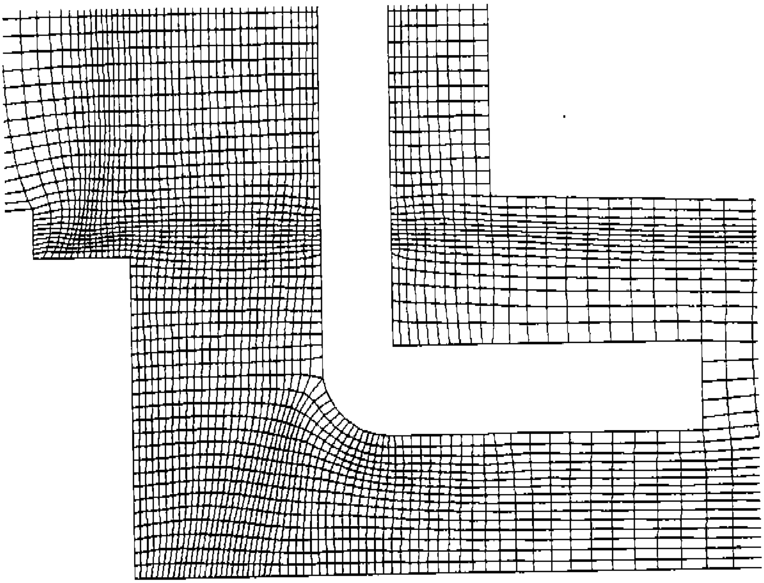

Fig.4. Computational Grid in Valve Plate Region

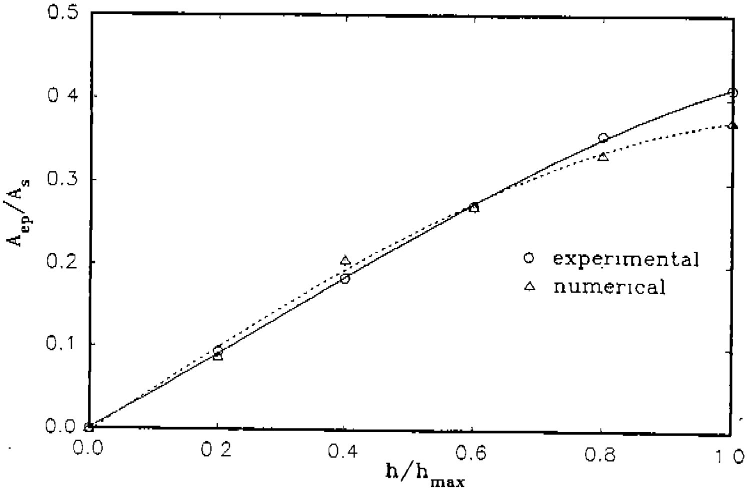

Fig.5. Effective Flow Area vs. Lift for Suction Channel Valve with Low Pressure Air (Re_s~2×10^4)

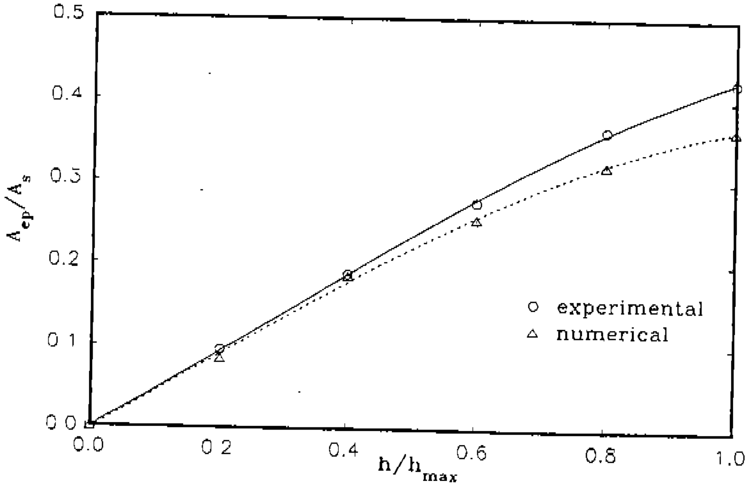

Fig.6. Effective Flow Area vs. Lift for Discharge Channel Valve with Low Pressure Air (Re_s~2×10^4)

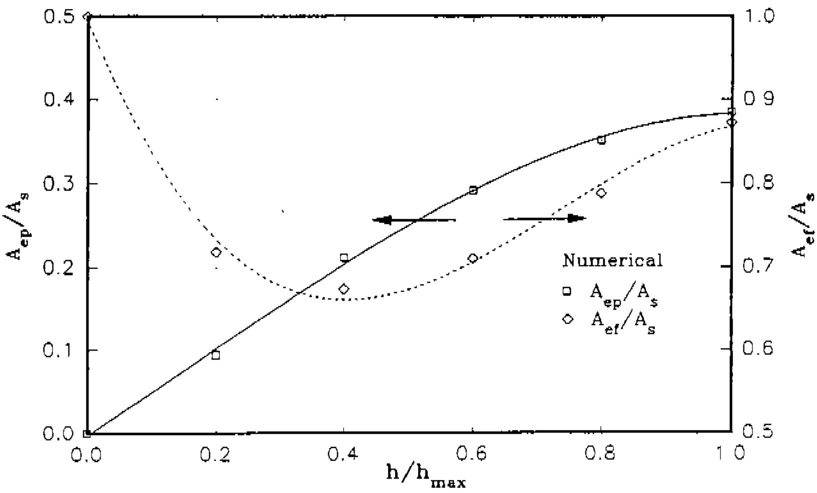

Fig.7. Effective Flow and Force Areas vs. Lift for Suction Channel Valve with High Pressure Natural Gas (Re_s~$4x10^5$)

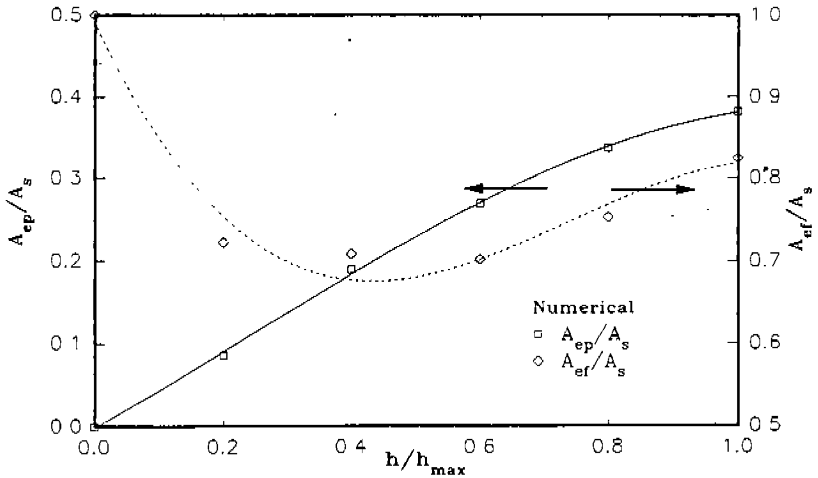

Fig.8. Effective Flow and Force Areas vs. Lift for Discharge Channel Valve with High Pressure Natural Gas (Re_s~$4x10^5$)

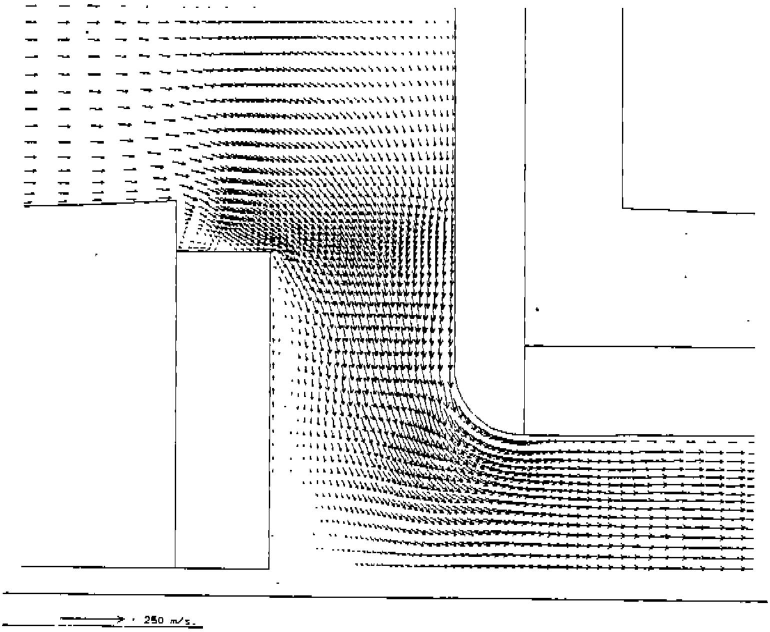

Fig.9 i.) Velocity Vectors in Valve Plate Region @ $h/h_{max} = 1.0$ for Low Pressure Air ($Re \approx 2x10^4$)

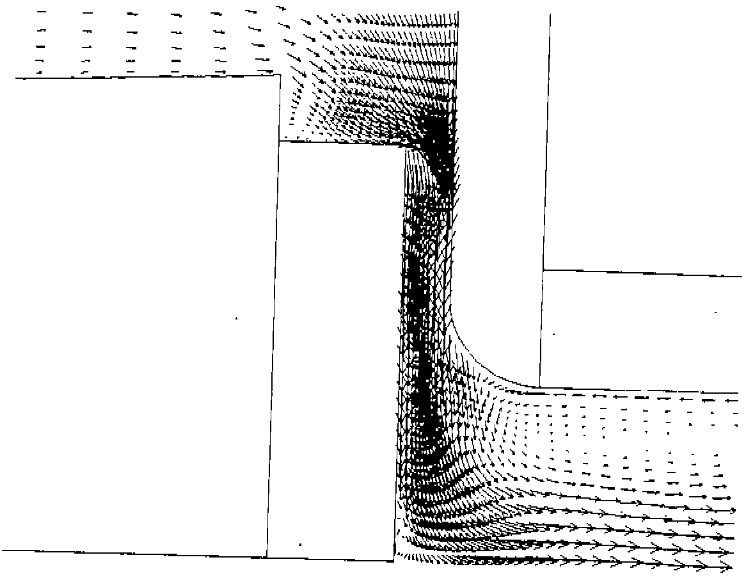

Fig.9 ii.) Velocity Vectors in Valve Plate Region @ $h/h_{max} = 0.2$ for Low Pressure Air ($Re \approx 2x10^4$)