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VORTEX SIMULATION IN RADIAL FLOW IMPELLERS OF CENTRIFUGAL PUMPS AT PART LOADS

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ABSTRACT

The phenomena of vortex formation in relative flow field of a radial flow centrifugal pump, running at zero and part loads of the pump discharge, has been investigated. The flow field is described by two dimensional Poisson's equation, which has been solved numerically, in difference form, by relaxation technique. A physical model for the vortex formation and shedding mechanisms, at part loads, has been introduced. The effect of vortex shedding, from the impeller, on the pressure fluctuation at the pump exit has been also discussed. Experimental investigation has shown a satisfactory agreement between the measured and the theoretically predicted pressure fluctuations at part flow rates. At high flow rates, the viscosity effect on the vortex simulation model, has to be considered.

INTRODUCTION

The performance of a centrifugal pump depends to a great extent on the flow pattern throughout the pump elements. It was found by Johnson and Dean, [1], that the majority of the hydraulic losses occur mainly in the pump casing as well as inside the recuperating elements. These losses could be reduced by improving the flow conditions inside the impeller passages and at its exit. The development of stationary and non-stationary flow patterns inside the impeller passages, at different pump discharges, are of particular interest for designers of high head centrifugal pumps. This is because of the strong interaction between the impeller and the diffuser flow conditions, as shown by Lennemann and Howard, [2]. The flow field inside the centrifugal pump impeller has been intensively studied by various means, such as using ; the potential flow theory by Sorensen, [3], Acosta, [4], the conformal mapping by Kimimoto and Matsuoka, [5], the method of series expansion by Susumu, [6], the method of singularity by Reddy and Kar, [7], and the flow visualization method by Senoo Yamaguchi and Nishi, [8].

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In the present work, a simple vortex model is introduced to emphasize the direct correlation between the flow structure, throughout the impeller passages, and the delivered pressure at the pump exit, specially at part loads. Theoretical and experimental investigations have been elaborated to study the impeller flow effect on the volute performance. In the theoretical study, a potential flow field is assumed to exist throughout the vane passages. The flow field is described by the two dimensional Poisson's equation which has been solved numerically by relaxation method. A physical model for the vortex formation and shedding mechanisms, at part load, has been introduced. The amplitude and frequency of the pressure fluctuation in the pump discharge were predicted by the aid of this model. In the experimental study, the amplitude and frequency of the pump pressure fluctuation were measured. The theoretical and the measured values are compared for different pump flow rates.

FLOW FIELD

While the flow field in the impeller of a centrifugal pump is unsteady when viewed from a stationary frame of reference, the flow may commonly be steady when viewed from a relative frame of reference. For two dimensional potential flow, the flow field may be described by Poisson's equation, Massey, [9], as follows :

$$\nabla\psi^2 = -2\omega^2 \qquad (1)$$

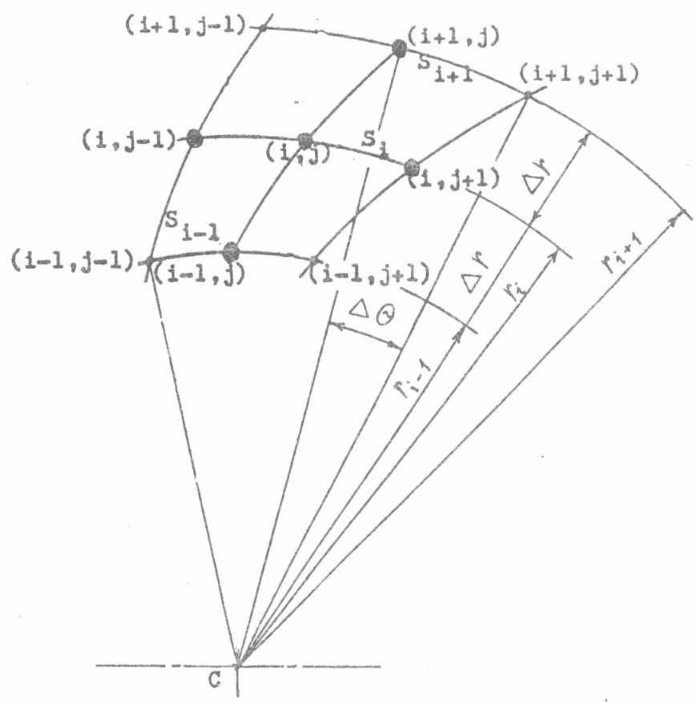

Fig.(1), Elementary Element.

In polar radial coordinates, this equation may be written in finite difference form by using Taylor expansion, shown in figure (1), as follows :

$$\psi_{i,j} = A_1 [\, A_2\psi_{i+1,j} + A_3\psi_{i-1,j} + A_4(\psi_{i,j+1} + \psi_{i,j-1}) + 2\omega \,] \qquad (2)$$

where

$$A_1 = \frac{0.5}{(1/\Delta r)^2 + (1/r\Delta\theta)^2}$$

$$A_2 = \frac{1}{(1/\Delta r)^2} + \frac{1}{(2r\Delta\theta)^2}$$

$$A_3 = \frac{1}{(\Delta r)^2} - \frac{1}{2\Delta r . r}$$

$$A_4 = \frac{1}{(r\Delta\theta)^2}$$

A polar grid is constructed in the flow space of the impeller passage in r and θ directions. A set of simultaneous linear equations (2) have been formed for all nodal points of the grid. The set of algebraic equations have been solved numerically by the aid of relaxation method [10], with zero stream function as a boundary condition on the vane surfaces. The flow field could be deduced by superimposing two flow fields; a circulatory flow field, (zero flow rate $Q = 0$), and flow field throughout the passages without impeller rotation ($\omega = 0$), [11]. Figures (2a), (2b) and (2c) show sample results for the impeller flow field at part loads. A well defined vortex flow field is formed, of a size that varies as flow rate changes. The model assumes that, as soon as the vortex is completely formed it is shed out under the effect of the flowing fluid. Therefore, another vortex starts to be built up and shed again. Figure (3) shows that the vortex shedding process causes two distinct effects. First, a reduction in the effective flow area corresponds to the vortex size, which results in a change of the outflow velocity of the impeller. Second, alteration in the inflow and outflow angles, which affects the pump performance. It was found by Lennemann and Howard, [2], that these effects may be considered as time dependent, so a periodical change in the amplitude and frequency of pressure at the impeller exit may be estimated.

Hence, the simultaneous building up and shedding process of the vortices affect the flow field and the pressure at the impeller exit. This mechanism, which simulates the actual flow condition in the impeller passages, suggests a strong correlation between the pressure fluctuation at impeller exit and the principal dimensions of the impeller, such as; number of vanes, profile of vane, inlet and outlet impeller diameters,..etc.

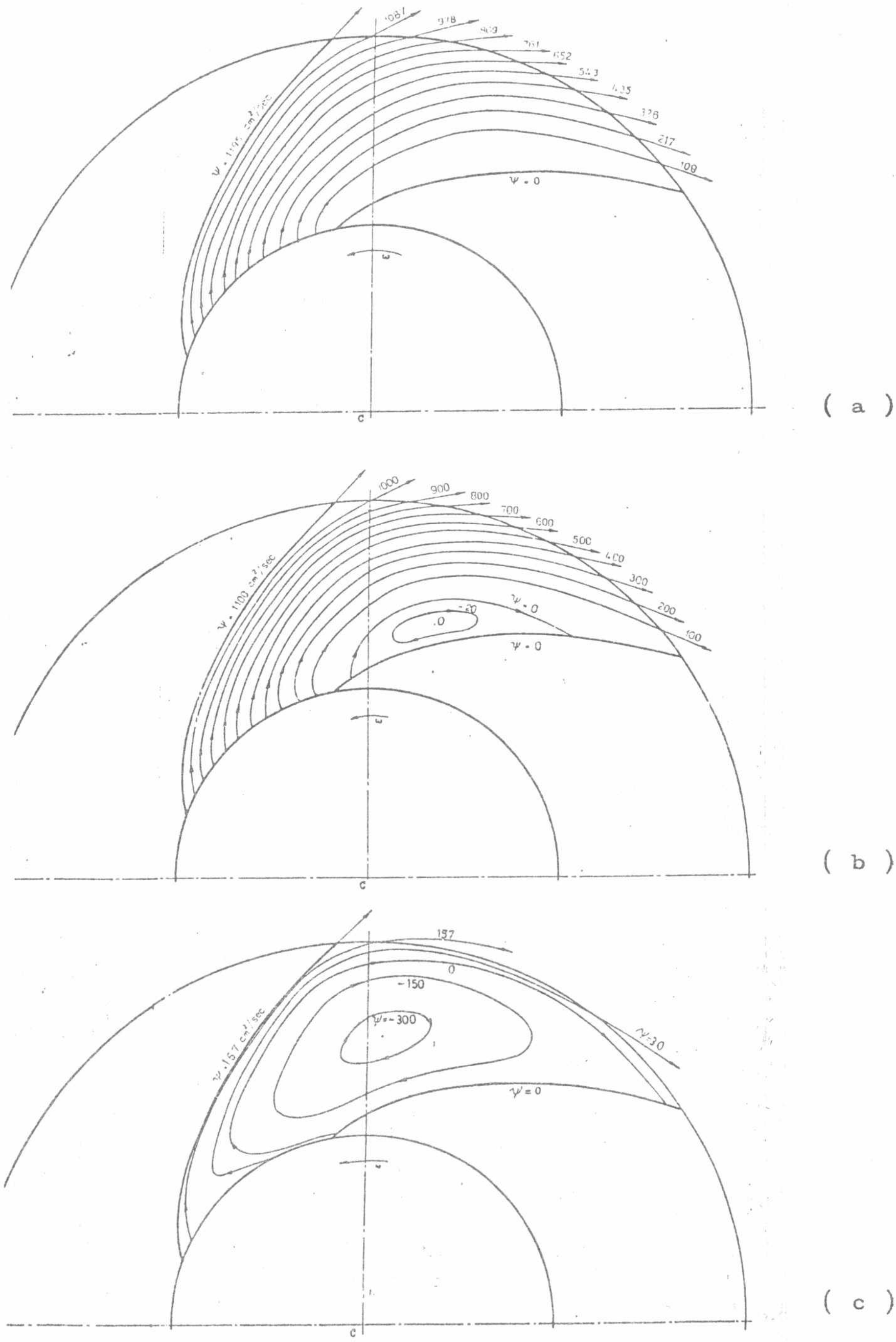

Fig.(2). Flow Field For Different Flow rates:
a- $Q/Q_o = 1$, b- $Q/Q_o = 0.92$, c- $Q/Q_o = 0.14$.

generation to exit sections.

Then;

$$t_d = \frac{R_2 - r_v}{C_{rm}} \qquad (5)$$

In this case, the pressure fluctuation frequency, from the relative, (rotating), reference is f_{sh} :

$$f_{sh} = \frac{C_{rm}}{2(R_2-r_v)} \qquad (6)$$

and from the absolute (stationary), reference is f'_{sh} :

$$f'_{sh} = f_{sh}/Z_v . \qquad (7)$$

where Z_v is the number of vanes for the impeller. The amplitude of the pressure fluctuation may be predicted from the changes of fluid velocity triangle, as the vortex is shed out. The instantaneous pumping head is considered as the mean value during the vortex shedding time.

EXPERIMENTAL STUDY

To verify the assumed modelling, an experimental investigation has been carried out for a radial type centrifugal pump, having the following parameters :

D_1 = 90 mm, D_2 = 180 mm, β_1 = 28°,
β_2 = 44°, z_v = 6 vanes, H = 5.7 m
Q_o = 500 l/min, n = 1450 rpm.

The pressure fluctuation, at different flow rates, was measured by using a pressure transducer of a range 1 bar. The fluctuation was amplified and recorded by means of a Hottenger recorder. The pump flow rate was measured by a rotameter. The pressure was measured as near as possible to the impeller exit, through measuring holes on the volute casing, each 30°. The description of the test rig, and the complete experimental results are given in [12]. Figures (5) and (6), show the values for the pressure fluctuation frequency and amplitude , measured at an angle of 120° from the volute tongue.

The figures show satisfactory agreement for the frequency of the pressure fluctuation, while a partial agreement is noticed for the amplitude. This may be due to the separation of flow at overloads, and due to viscous damping effects of real fluids.

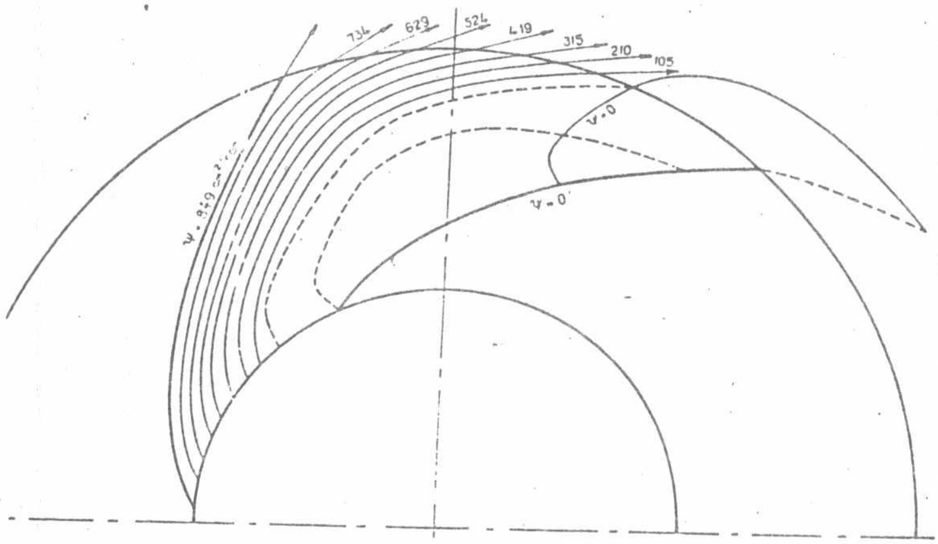

Fig.(3), Vortex Shedding Mechanism, at $Q/Q_o = 0.71$.

MECHANISM OF VORTEX MODEL

Although the impeller rotation for constant flow rate, the formed vortices are found to be of non stationary character as shown above. Successive vortex building-up and shedding process results usually in pressure fluctuation process at the impeller exit. To simulate the vortex formation and shedding mechanisms as well as the pressure fluctuation, it has been assumed that :

a. The size and shape of the formed vortex do not change during the shedding process,
b. There are neither overlap nor time shift between two successive vortices,
c. The vortex moves radially a distance of ($R_2 - r_v$), with a constant radial mean velocity,

$$C_{rm} = (C_{R2} + C_{rv}) / 2. \qquad (3)$$

where r_v is the centroid pitch radius of the generated vortex at the moment of completion of its formation, see Figure (3). C_{rv} is the radial component of the flow velocity at the vortex center. C_{rm} is the mean radial velocity component at impeller exit. From the numerical results, it was observed that the vortex centroid is almost at mid-distance in the channel, hence the time of vortex shedding may be predicted as follows :

$$t_{sh} = 2\, t_d \qquad (4)$$

where t_d is the translation time of vortex-center from the

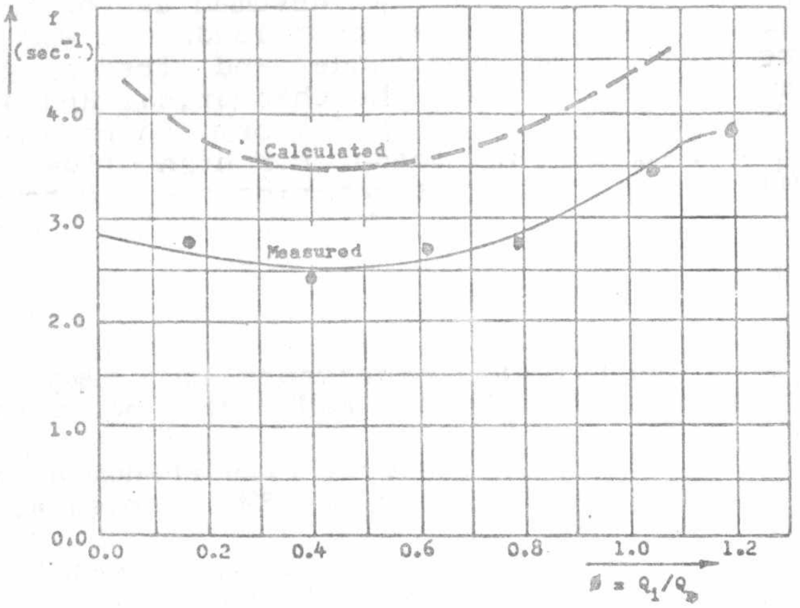

Fig.(5), Pressure Fluctuation Frequency.

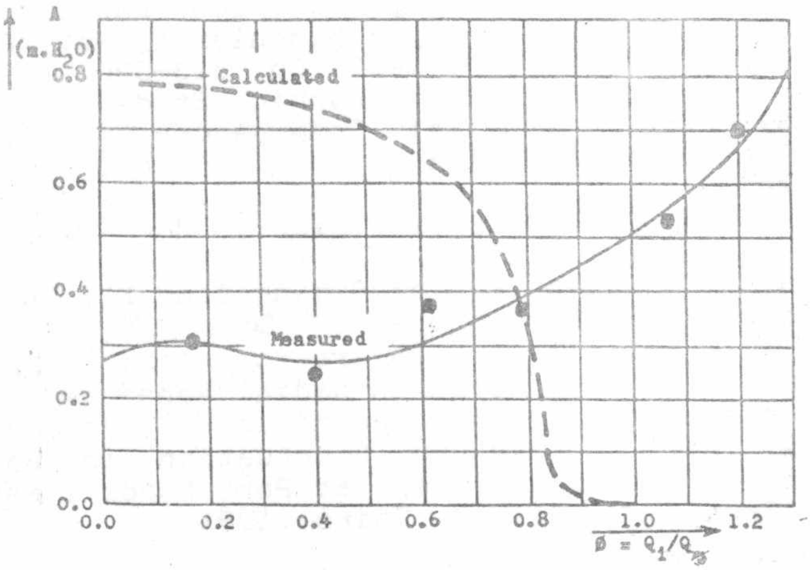

Fig.(6), Pressure Fluctuation Amplitude

CONCLUSION

Theoretical two dimensional model, based on the Poisson's equation of ideal flow, has been proposed to simulate the vortex effect in radial impellers, at part loads. The vortex generation and shedding have been assumed as the main source of the pressure fluctuation at part loads. It was found that the pressure fluctuation, amplitude and frequency, increase for both part and over loads. The theoretical and experimental results, are found to be satisfactory correlated for the frequency behaviour at part loads. For high flow rates, the viscous damping and the flow separation have to be considered.

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NOMENCLATURE

A	variable defined by Eq.(2),
C	velocity, m/s,
D	diameter, m,
f	frequency, 1/s,

H total head, m water,
n speed, rpm,
Q flow rate, m^3/s,
R radius, m,
r radius, m,
t time, s,
Z number of vanes,

β vane angle, rad,
ψ stream function m^2/s,
ϑ angle, rad,
ω angular velocity, 1/s,

Subscripts

d displacement,
i,j point on polar coordinates,
m mean
r radial,
sh shedding,
v vane or vortex,
1 vane inlet,
2 vane outlet.