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AN EXPERIMENTAL INVESTIGATION ON THE CLEARANCE EFFECTS OF SEMI OPEN RADIAL FLOW IMPELLERS

by

T. Kurian and H.C. Radha Krishna

SUMMARY

An experimental investigation has been carried out to study the influence of clearance between the impeller and the casing on the performance of a radial flow semi-open impeller. The results show considerable variations in the performance characteristics until a particular clearance is reached, beyond which the performance characteristics remain unaltered. The possible reasons for the considerable variations in the performance with clearance have been discussed.

Hydroturbomachines Laboratory, Department of Mechanical Engineering, Indian Institute of Technology, Madras:600 036.

1. INTRODUCTION

1.1 It is well known that the performance of a semi impeller depends to a great extent on the clearance space between the impeller and the casing at the front side. Several experimental studies have been done on the clearance effects especially in axial flow pumps (1,2,3). In centrifugal pumps, the studies of Folsom (4) and Wood et. al (5) show the importance in giving due consideration for the clearance in the design of semi-open impeller centrifugal pumps.

1.2 The maximum clearances achieved in the reported cases (4, 5) are limited. To design semi-open impellers for handling liquids with solid particles of larger sizes of irregular shapes more information is required.

1.3 The rate of the variation of the performance coefficients with clearance are observed to be not same for impellers of different designs (5). From the flow analysis near the vane tips (3, 6) it appears that the flow patterns are very complex. Methods are not available at present for the prediction of clearance effects on the semi-open impeller performance. As the clearance effects are found to be the same order for different designs, the experimental data for as many designs as possible would help the formulation of some methods for predicting the clearance effects on the performance characteristics.

2. EXPERIMENTAL SET UP

2.1 The schematic representation of the experimental set up is shown in Fig. 1. The centrifugal pump was designed by the authors and fabricated at the Institute workshops. The design was incorporated with the provisions for changing the axial clearance in steps up to 12 mm without losing the symmetric position of the impeller outlet with the casing inlet. The impeller used in the investigations is shown in Fig. 2. The inlet angle β_1 of the impeller is 18 deg. and the outlet angle β_2 is 28 deg. The number of vanes for the impeller is 8.

2.2 The pump was driven by a 9.5 kW d.c. swinging field dynamometer motor by which the shaft torque was measured. The discharge was measured by a calibrated orifice meter. The suction pressure was measured by an U-tube mercury manometer and the delivery pressure by a calibrated pressure gauge. The speed measurement was done by an inductive pick up and an electronic counter.

3. RESULTS AND DISCUSSION

The following abbreviations are used in the discussions:

max	maximum
opt	optimum

3.1 Performance tests were done for several clearances varying from 0.4 mm to 12 mm and for different speeds. The results are presented in non-dimensional terms. The performance coefficients are defined by

Energy coefficient $$\psi = \frac{2Y}{u_2^2} \qquad (1)$$

where Y - specific work in m^2/s^2

u_2 - peripheral speed of the impeller outlet (m/s)

Flow coefficient $$\varphi = \frac{V}{u_2 A} \qquad (2)$$

where V - discharge in m^3/s

A - $(\pi/4)\ D^2$ in which D is the impeller outer diameter in m.

Power coefficient $$\nu = \frac{N}{\rho n^3 D^5} \qquad (3)$$

Efficiency $$\eta = \frac{\rho V Y}{N} \qquad (3a)$$

where N - coupling power in Watts

ρ - fluid mass density in kg/m^3

n - shaft speed in revolutions per minute.

The clearance is expressed as a non-dimensional factor, clearance ratio, defined as the ratio of the clearance to the impeller diameter i.e. Cl/D. The performance characteristics for the constant shaft speed of 3000 rpm are shown in Fig. 3 where clearance ratio is the changing parameter.

3.2 The efficiency curve for the smallest clearance falls on the top compared with those for higher clearances indicating that for a semi-open impeller, the best performance occurs when the clearance as reported by Spencer (2) for axial flow pumps and predicted by Hammit (7) for centrifugal pumps is not observed here. It is seen that with the increasing clearance ratio, the performance curves lie one below the other as observed by others (4, 5) till the clearance ratio becomes 2.66%, But for all the clearance ratios above 2.66%, noticeable changes in the performance coefficients are not observed and the performance curves fall almost on the same lines. From the fall in the efficiency with the increased clearance it is seen that the rate of decrease in energy coefficient is higher than the rate of decrease in the power coefficient.

3.3 From the performance characteristics given in Fig. 3 and those for the other speeds, the maximum efficiency η_{max} the power coefficient, energy coefficient and flow coefficient corresponding to η_{max}, ψ_{opt}, ν_{opt} and φ_{opt} respectively with Cl/D are shown in Figs. 4 to 7. From Fig. 4, it is seen that the decrease in maximum efficiency in the initial regions of clearance ratio is very steep till Cl/D becomes 1%. After this η_{max} decreases at a slow rate until Cl/D = 3%. For Cl/D greater than 3%, the η_{max}, remains almost constant. The total change in maximum efficiency with clearance can be seen to be around 20%. Regarding the variations of ν_{opt}, ψ_{opt} and φ_{opt} with Cl/D, the trends are almost similar to η_{max}-Cl/D variation, i.e. they fall from the values corresponding to the lowest value of the clearance considerably till Cl/D = 3% and afterwards remain the same.

3.4 Such variations in the performance as affected by the clearance variation may be explained as follows: The leakage loss increases with the clearance due to the increase in the flow passage area. As a result of this, for a constant flow rate, the energy coefficient falls down. The effect of an increased leakage loss on the power coefficient is apparently a rise; but it is found here that the power coefficient is not at all showing a tendency for a rise. This indicates that the effect of the rise in the leakage loss on the power coefficient is exceeded by the other effects arose from non-active vane surfaces and unloading of the vanes. The cross flow from the pressure surface of the vanes to the suction surface causes the decrease in the vane loading. The high pressure leakage flow find its way to the vane passages and as a result shift the main flow, thus reducing the vane loading and rendering a certain portion of the vanes non active to energy transfer. Also the cross flow is associated with losses by the formation of tip vortices (3). The decrease in the energy coefficient due to the unloading and non-activeness of the vanes is directly followed by the decrease in the power coefficient at the same rate. If this were the only effect of the axial clearance the efficiency must have remained the same. But as mentioned before the efficiency decreases considerably which means the rate of decrease of the energy coefficients is much higher than that of the power coefficient. This indicates that the energy coefficient is decreased by leakage flow, vane unloading and the losses associated with the formation of tip vortices. The expected rise in the power coefficient with the leakage flow is compensated by the effects of vane loading and further reduces it showing only fall in the power coefficient. This power requirement for compensation may be the reason for the large variation of the efficiency in the initial regions of clearance ratio. Also since the power coefficient falls with further increase of axial clearance, it becomes evident that further unloading of the vanes takes place with increasing clearance.

3.5 But for clearances beyond about 3% no more unloading of the vanes takes place and the effects of the leakage loss remains almost constant as seen from the nature of the performance curves.

3.6 Disk friction losses do not exist on the open side of the impeller due to the absence of a solid surface. However it is present for the back shroud of the impeller, but its variation with the varying clearance is of no significance when compared with the effects of the clearance at the front side of the impeller on the performance.

4. CONCLUSION

From the observations, several deductions can be made regarding the operation of semi open impeller centrifugal pumps. The best performance for a semi open impeller is obtained for the lowest value of the axial clearance. For clearance ratios more than about 3%, the clearance effects on the performance remain the same. For pumps handling liquids with solid particles of irregular shape and considerable sizes, large clearances can be provided between the impeller faces and the casing walls if the required clearance is greater than 3% of the impeller diameter to avoid jamming.

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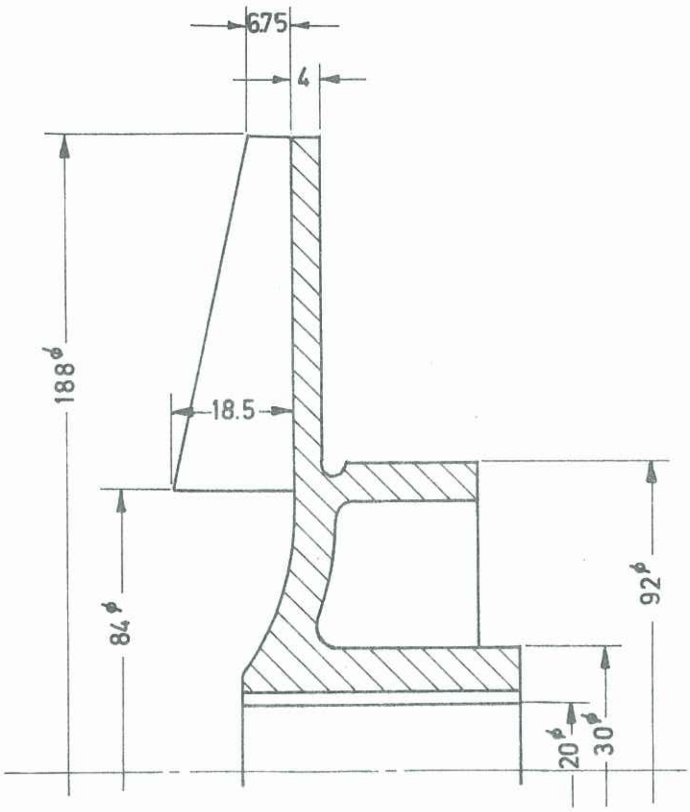


CROSS SECTION OF IMPELLER

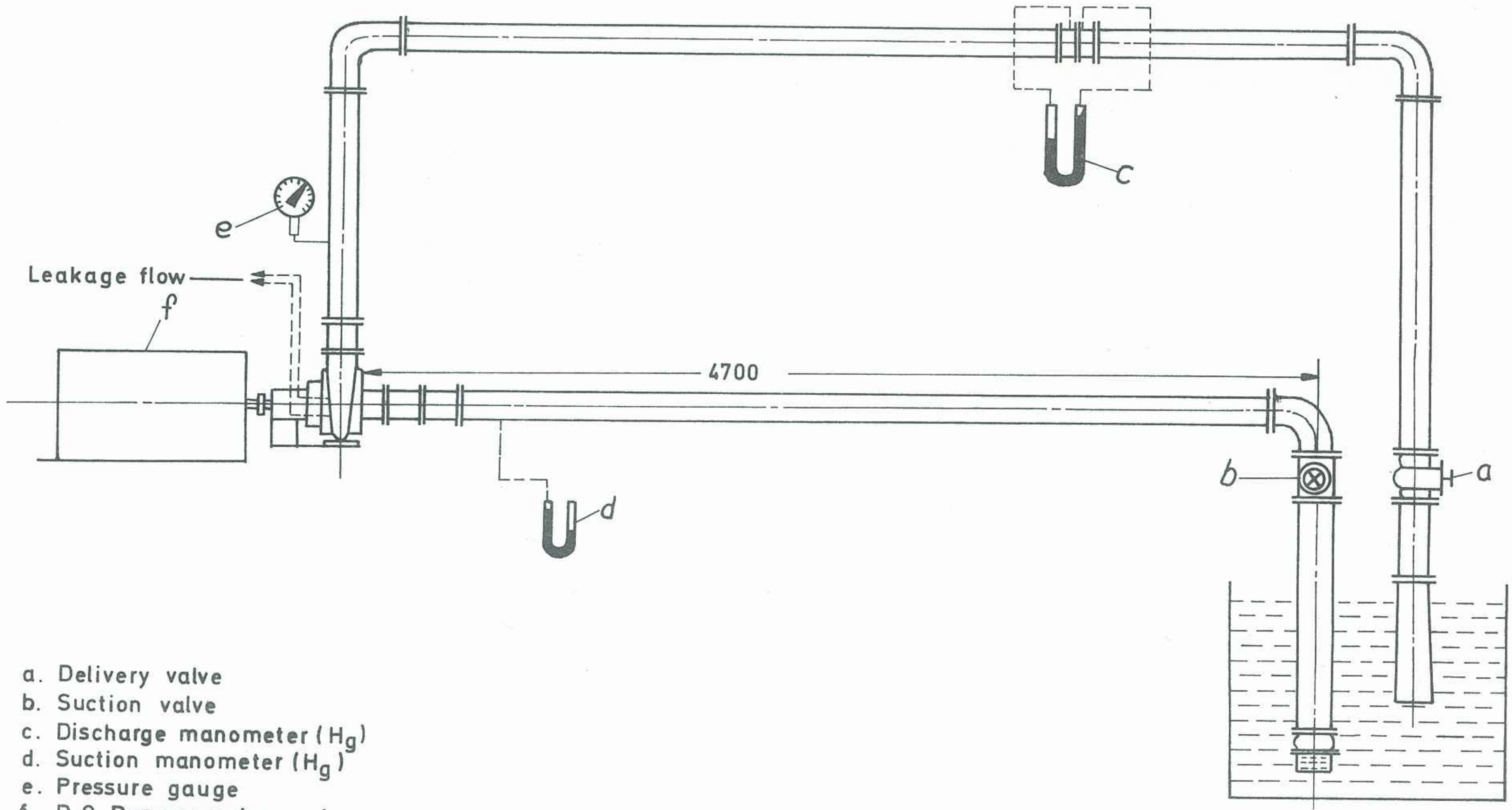
$$n_{sh} = \frac{1000 \times n \times \sqrt{V}}{y^{3/4}}$$

n_{sh} (design) = 60

(b.e.p for the minimum $\frac{Cl}{D}$)

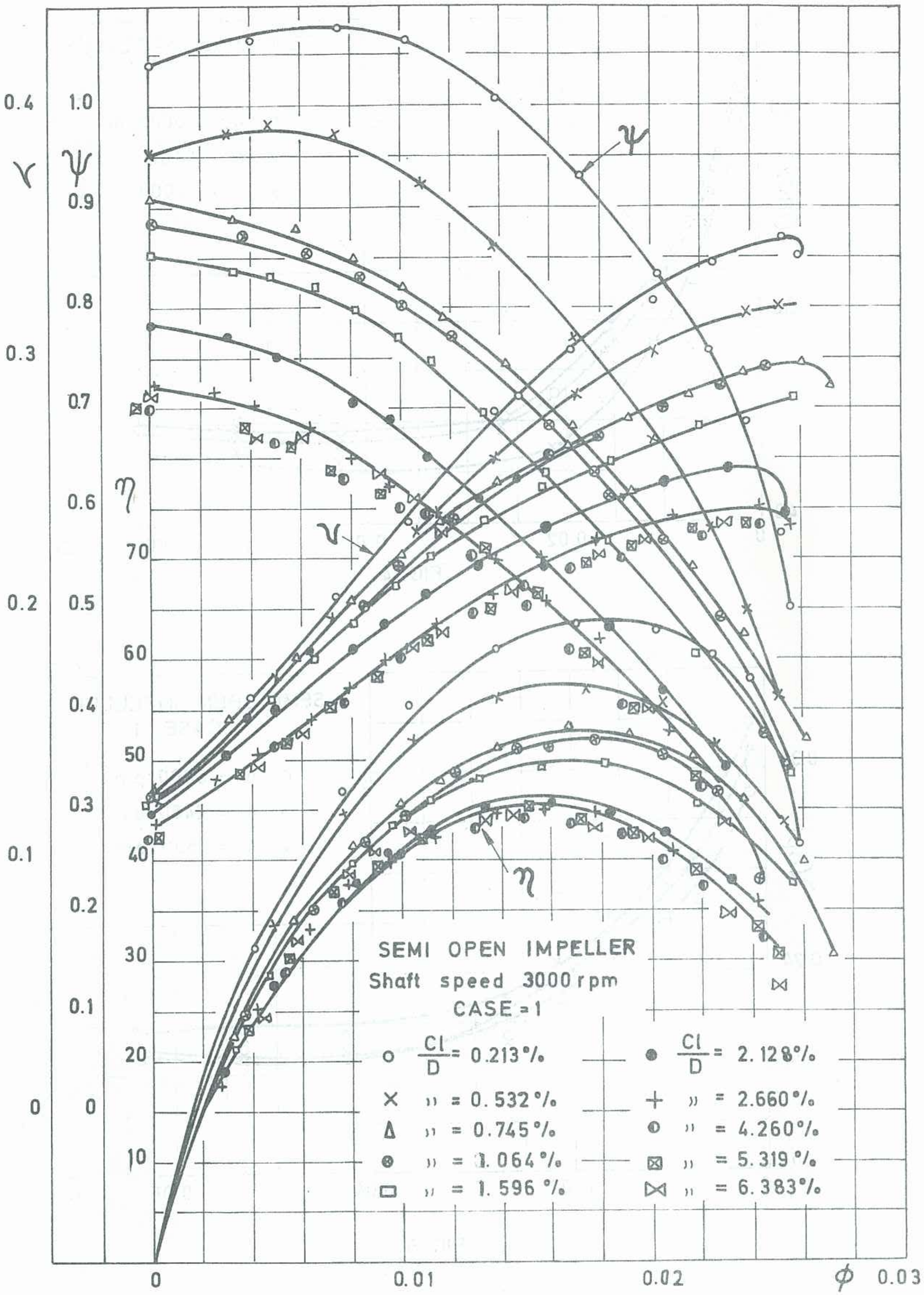
= 74

FIG. 2.



a. Delivery valve
b. Suction valve
c. Discharge manometer (H_g)
d. Suction manometer (H_g)
e. Pressure gauge
f. D.C. Dynamometer motor

FIG. 1. LAY_OUT OF THE EXPERIMENTAL SET_UP



0.4
0.3
0.2
0.1
0
1.0
0.9
0.8
0.7
0.6
0.5
0.4
0.3
0.2
0.1
0
70
60
50
40
30
20
10
ν
ψ
η
ψ
ν
η
SEMI OPEN IMPELLER
Shaft speed 3000 rpm
CASE = 1
○ $\frac{Cl}{D}$ = 0.213 %
× " = 0.532 %
Δ " = 0.745 %
⊗ " = 1.064 %
□ " = 1.596 %
● $\frac{Cl}{D}$ = 2.128 %
+ " = 2.660 %
◐ " = 4.260 %
⊠ " = 5.319 %
⋈ " = 6.383 %
0
0.01
0.02
φ 0.03

FIG. 3

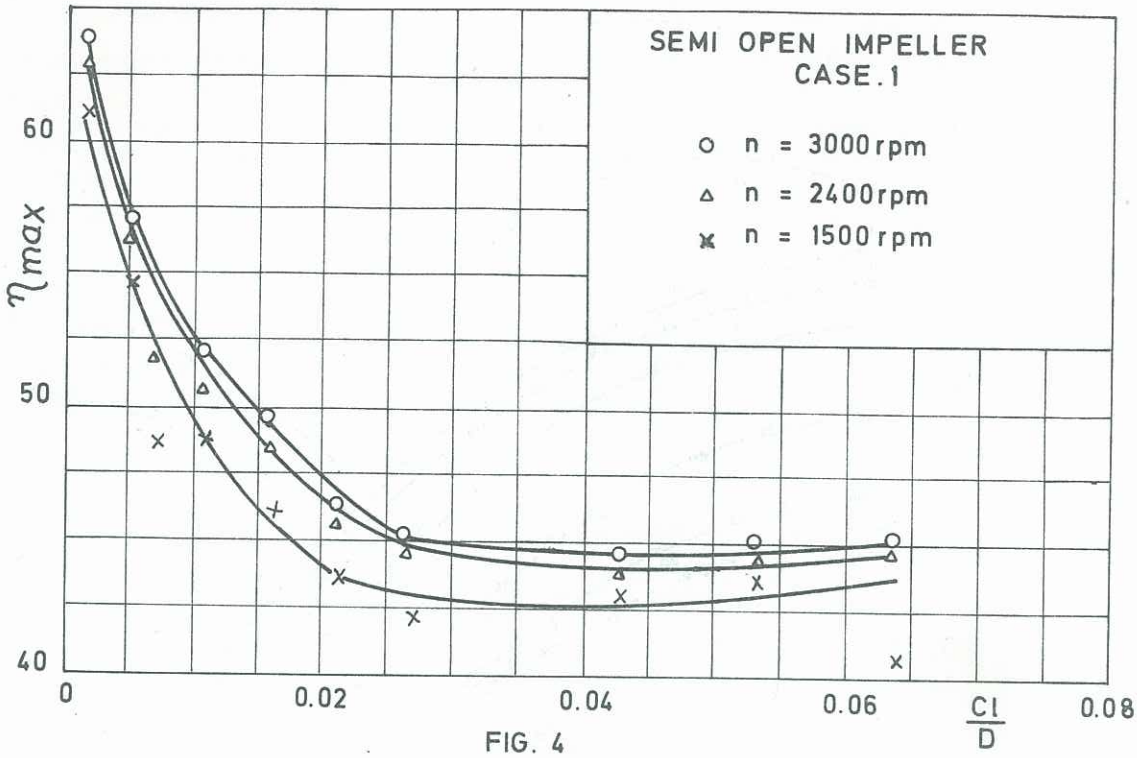

SEMI OPEN IMPELLER
CASE. 1
o n = 3000 rpm
△ n = 2400 rpm
x n = 1500 rpm
60
50
40
η_{max}
0
0.02
0.04
0.06
0.08
$\frac{Cl}{D}$

FIG. 4

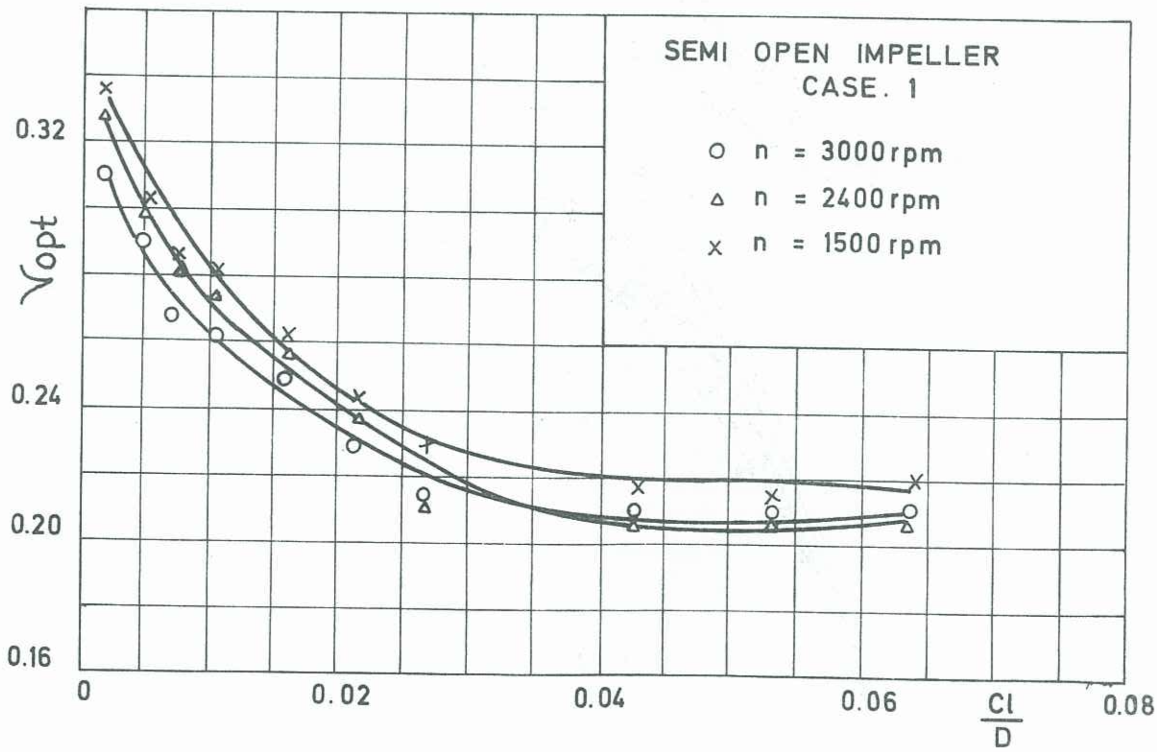

SEMI OPEN IMPELLER
CASE. 1
o n = 3000 rpm
△ n = 2400 rpm
x n = 1500 rpm
0.32
0.24
0.20
0.16
ν_{opt}
0
0.02
0.04
0.06
0.08
$\frac{Cl}{D}$

FIG. 5.

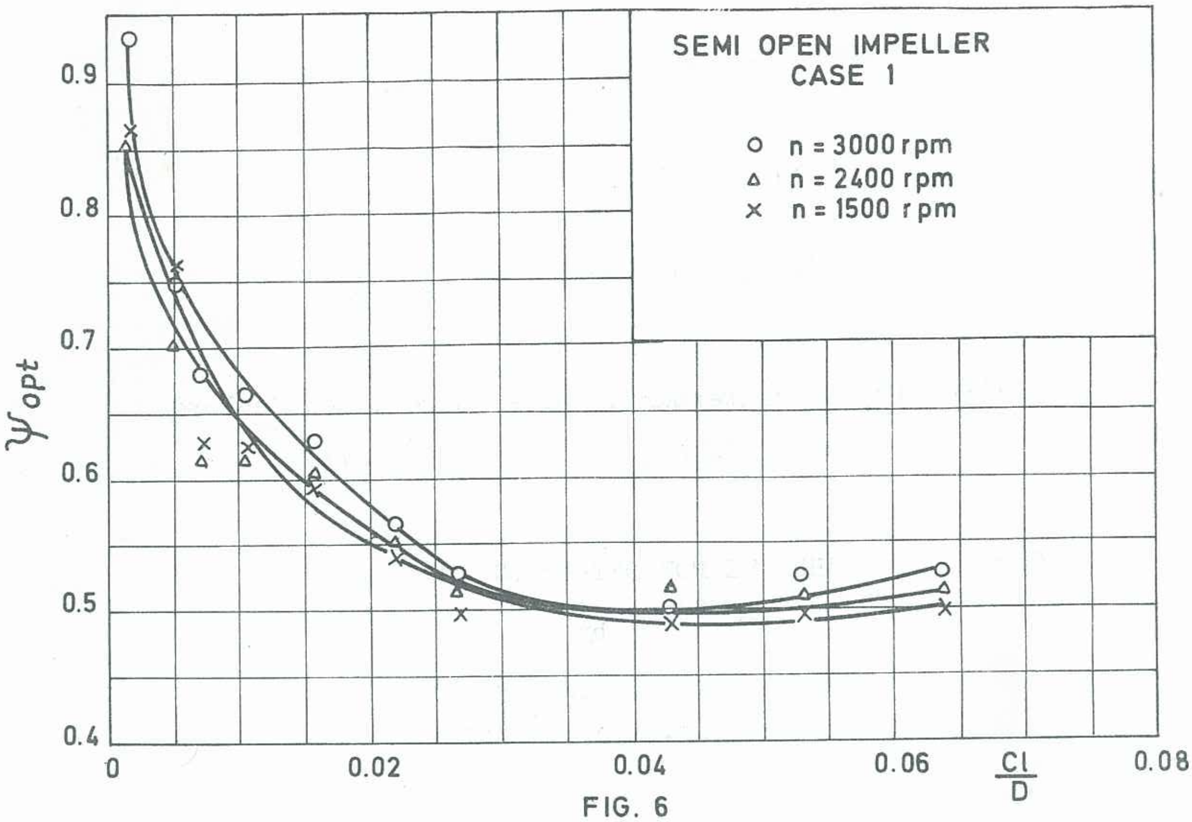

SEMI OPEN IMPELLER
CASE 1
n = 3000 rpm
n = 2400 rpm
n = 1500 rpm
ψ_{opt}
0.9
0.8
0.7
0.6
0.5
0.4
0
0.02
0.04
0.06
0.08
$\frac{Cl}{D}$

FIG. 6

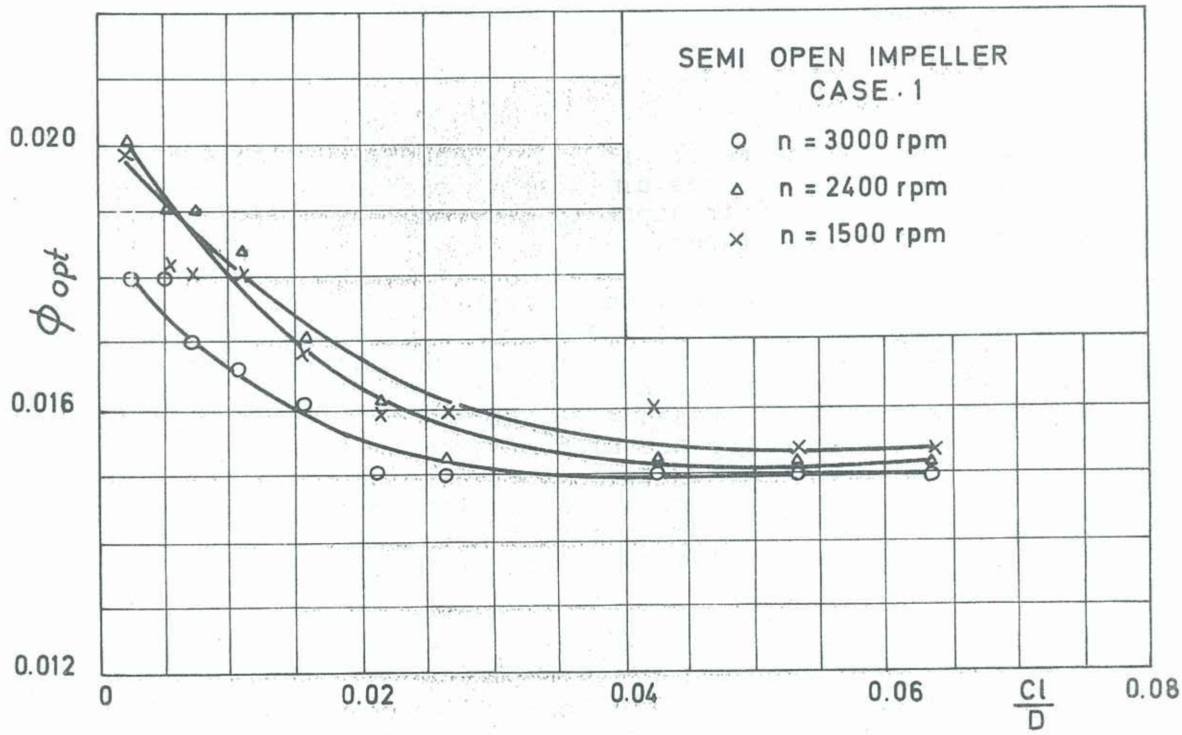

SEMI OPEN IMPELLER
CASE · 1
n = 3000 rpm
n = 2400 rpm
n = 1500 rpm
ϕ_{opt}
0.020
0.016
0.012
0
0.02
0.04
0.06
0.08
$\frac{Cl}{D}$

FIG. 7