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EXPERIMENTAL INVESTIGATION OF THE EFFECT OF THE IMPELLER BLADE EXIT CUT ON THE PERFORMANCE OF A RADIAL COMPRESSOR

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ABSTRACT

This study deals with the experimental investigation of the radial compressor characteristics when the blade trailing edge is cut in the radial direction. Five impeller samples, of type, " HOLSET-4 ", are manufactured. One sample is used as a reference and the other 4-samples are changed, by cutting a radial length from one blade while keeping the next blade fixed. At off design speeds ,the maximum efficiency was found to be affected strongly by these cuts. The relation between the cut ratio and the maximum efficiency was reported and correlated. The corresponding mass flow rate and pressure ratio were also determined. It was found that as the cut ratio icreases the mass flow rate decreased with the increase of max. efficiency. The pressure ratio was found to change in an opposite manner to that of mass flow rate.

NOMENCLATURE

A	Coefficient, or area of flow, m^2.
B	Coefficient.
C	Coefficient.
cp	Specific heat at constant pressure, J/kg K.
g	Gravitational acceleration, m/s^2.
K	Trailimg edge cut ratio (T.E.C.) = $\Delta r/R_2$
m	Mass flow rate, kg/s .
m_o	Mass flow rate at maximum efficiency, kg/s .
N_c	Compressor speed of rotation, rpm .
N_{cd}	Compressor design speed, rpm .

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N_r	Compressor speed ratio (N_c/N_{cd}).
P	Static pressure, N/m^2.
P_o	Total pressure, N/m^2.
PR	Pressure ratio = (P_{o2c}/P_{o1c}).
PR_o	Pressure ratio at maximum efficiency.
R	Gas constant, J/kg K.
T	Static temperature, K.
T_o	Total temperature, K.
X	Axial length, m .

Greek Letters

α	Flow angle, deg
ε	Leading edge cut (L.E.C.) ratio = $\Delta X/L$
γ	Index isentropic process.
η	Efficiency = $\dfrac{\left(PR^{\frac{\gamma-1}{\gamma}} - 1\right)}{\left(\dfrac{T_{o2c}}{T_{o1c}} - 1\right)}$
ρ	Air density, kg/m^3.

Subscript

c	Compressor.
d	Design.
o	Total, or condition at maximum efficiency.

INTRODUCTION

The flow field in centrifugal compressor was studied by several investigators. They showed that most of the energy losess occur in the impeller [1,2,3,4]. The impeller flow phenomenon was found to deviate from the potential flow [5,6]. The impeller-flow may be assumed to be a wake flow which produces large separation zones as well as significant jet and wake regions at the impeller exit. The eddies and wakes occur at the same flow rate [8,9].

M.W.Johnson and J.Moore [5] studied experimentally the influence of the flow rate on the wake in a radial impeller. They found that significant total pressure losses occur in tow regions of the impeller :

1- In the pressure and suction side / shroud corner region in the inducer at both " below design " and " design " flow rates.
2- In the final radial section of the impeller at all mass flow rates.

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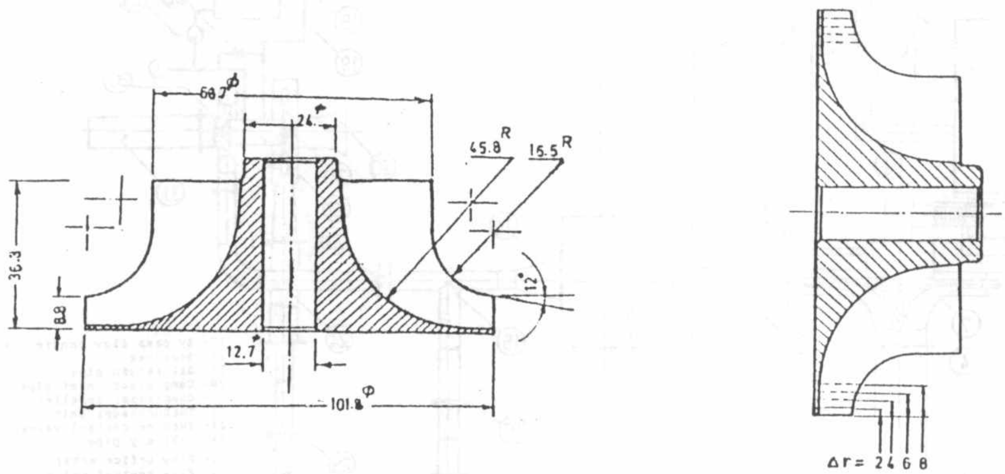
* Number between bracets refer to the reference at end of paper.

As indicated in [1,10] the maximum efficiency at design speed could be increased by about 5 % when reducing about 50 % of the inducer shock losess, due to eddyies formation, that is by using the leading edge cut (splitter) technique .

In this paper the technique of the trailing edge cut is used. This technique is based on creating a rotating mixing zone, [7] in order to reduce the mixing losses. And also, is based on the reduction of the boundry layer growth and total pressure loss when tandem blades are used [8,9]. So, by cutting the trailing edge of one blade and keeping the next one fixed will create rotating mixing zones and about 50 % of the mixing process will be done inside the impeller blades. Measurments have indicated that the compression efficiency is increased by about 7 % at certain off design speed while there is no significant effect at design speed [10].

EXPERIMENTAL MEASUREMENTS

In the proposed procedure the radial compressor of the turbocharger (HOLSET-4), of 100 mm outlet diameter and 25000 rpm design speed, is chosen as an experimental running model. So, five impeller samples are manufactured, similar to the original one. Four samples have blade cut with different trailing edge cuts from 2 to 8 mm. Figure (1) shows the impeller samples drawing.



a- Full bladed b- Exit cut

Fig.(1) Impeller samples

Experimental Setup

The experimental setup which is used herein is discribed in details in reference [10]. Figure (2) illustrates the layout drawing of the experimental setup. The model compressor samples are placed in the test section (21) of figure (2), and are driven by the air turbine (30). The compressor air flow is drafted through the inlet pipe (20) and delivered to the output pipe (24). The air volumetric flow rate is regulated by the manual control gate valve (26). The root compressor (7), of radial type, is used to drive the air turbine (30). It is driven by a 3-phase, variable speed " Schraga " motor (9), through the high speed mechanical transmission (8), to force the air into the air box (13) through the diffuser (12). The bypass gate valve (17) is operated to get the air turbine and the root compressor matched together by adapting the operating point of the root compressor far enough from the surge line.

The stable regimes of the compressor characteristics were studied under changing the following conditions :

1- The blade trailing edge cut ratio from 0 to 0.16 .
2- The speed of rotation by speed ratios from 0.4 to 1.0 .
3- The mass flow rate for each speed.

Measuring Instruments

The measuring instruments used herein are classified according to the following measuring parameters :

1-Total and static temperatures. 2-Static pressures.
3-Speed of rotation. 4-Air mass flow rate.

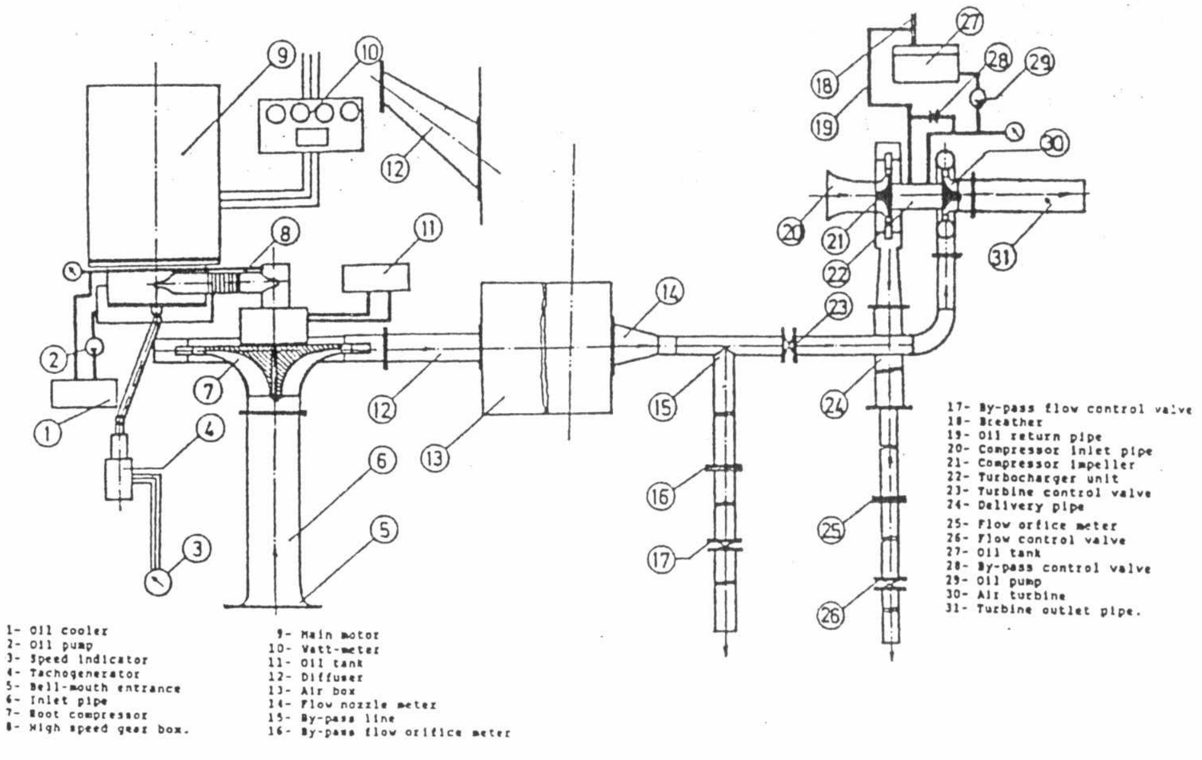


Fig.(2) Experimental setup layout.

Temperature Measurments

Total temperatures of the air flow at inlet and outlet sections of the model compressor as well as the air turbine are measured by fine thermocouple probes of type copper-constantan, encounterd the flow direction. The locations of the temperature probes are selected to confirm the uniformity of the flow stream along the pipe length. The static temperatures are measured upstream the orifice meter.

Pressure Measurements

The static pressures at inlet and outlet of the model compressor and the air turbine are measured in order to determine the total pressures by using the following equation [11] :

$$P_o = P \left[.5 + \sqrt{.25 + \frac{\gamma - 1}{\gamma} \left(\frac{m}{P.A}\right) \frac{R\ To}{\cos\alpha}^{2-}} \right]^{\gamma/\gamma-1} \tag{1}$$

Mass Flow Rate Measurements

The mass flow rate of the model compressor and the air turbine are measured by using a standared and calibrated orifice plates. The orifice meter is placed at the outlet pipe of the compressor. The static pressures are measured also upstream the orifice meter in order to correct the value of the mass flow rate, together with the static temperature. The measured mass flow rates are reduced to the standared conditions [11], ($P_a = 1.0132 \times 10^5\ N/m^2$, $T = 288.2$ K).

Speed of Rotation

Speed of rotation is measured by a hand tachometer of type (SMITHS J9847), its accuracy is cheked periodically by using a stroboscope.

EXPERIMENTAL RESULTS AND EVALUATION

Sample of the experimental results is presented graphically in figure (3). The original sample characteristics, is shown in figure (4), is used as a reference for comparison.

To correlate the measured results, the " Graphical and Numerical Method of Partial Relations ", was used to get some suitable empirical relations. The characteristic maps are refined and the desired parameters are determined. The following parameters are chosen for evluation :

1-The maximum efficiency at each speed (η_{max}).

2-The mass flow rate at maximum efficiency (mo).

3-The pressure ratio at maximum efficiency (PRo).

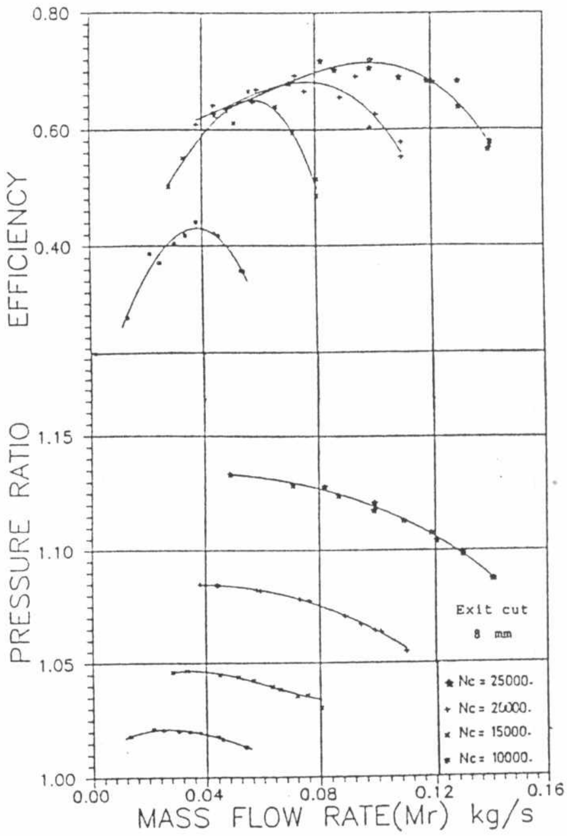


Fig.(3) Exit cut impellers characteristics

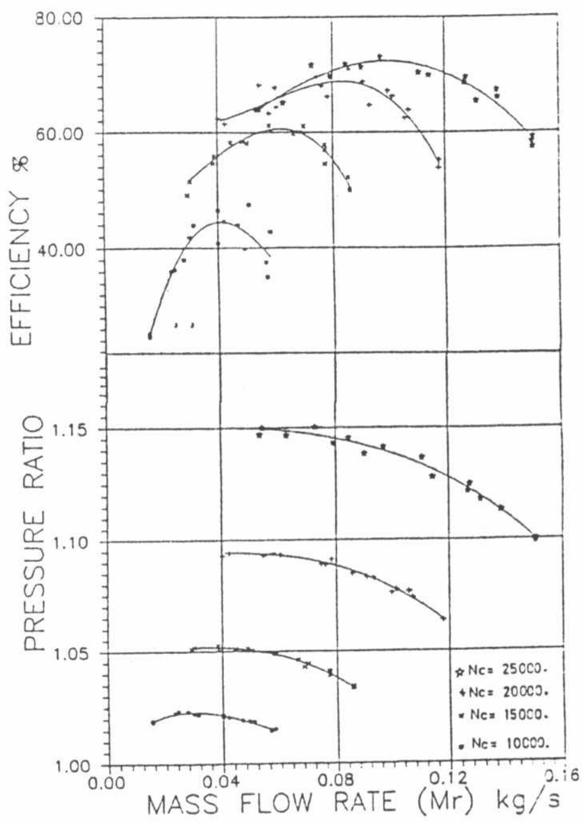


Fig.(4) Full bladed impeller characteristics

Maximum Efficiency Criterion (η_{max})

The maximum efficiency, at all different rating speeds are found to be affected strongly by impeller geometry changes as well as the speed of rotation.

The effect of varying the speed of rotation on the maximum efficiency is presented in figure (5) for the reference and exit cut samples. The figure shows that the maximum efficiency is icreased at speed ratios less than one (off design speeds) while it decreased at design speed. The speed ratio which corresponds to the peak value is shifted to lower speed value.

For the different cut ratios the maximum efficiency-speed relation was found to be almost of parabolic form such that :

$$\eta_{max} = A_1 + B_1 Nr + C_1 Nr^2 \qquad (2)$$

where A_1, B_1, and C_1 are functions of the blade and channel geometry

The variations of the maximum efficiency with the blade trailing edge cut ratios are shown in figure (6). The figure shows fluctuating variation pattern for the different speed ratios. The symmetric variation, around a range of exit cut ratio from 0.06 to 0.09 , suggests that the flow conditions repeat themselves as the cut ratio varies between 0.06 to 0.1579 The locaction of the peak values are almost the same for all speed ranges, but it is reversed for a speed ratio of 0.8 .

It is observed that the maximum efficiency is increased by a relative value of about 7.0 % at exit cut ratio of 0.1179 and speed ratio of 0.6 .

It is obsereved also that the maximum efficiency tends to decrease as the speed ratio increases, and there is no increase at speed ratio of 1.0 . So, the trailing edge cut technique may improve the maximum efficiency at off design speeds.

The aforementioned variation pattern may be attributed to the nature of the flow and energy transfer in the radial rotating channels [1 to 6]. So, one can mention here, that the changes of the blade trailing edge cut affect cosiderably the induced eddies and wakes, and accordingly the accompanied level of turbulence.

The incremental changes in the maximum efficiency may be correlated by the following relation :

$$\Delta\eta_{max} = \eta_{max} - \left(\eta_{max}\right)_{ref} \tag{3}$$

$$\Delta\eta_{max} = (A'_1 + B'_1 N_r + C'_1 N_r^2) \tag{4}$$

Curve fitting techniques are utilized to fit the obtained coefficients A'_1, B'_1, and C'_1 with the trailing edge cut ratio (K), in order to evaluate their general correlation , as follows :

$$A'_1 = 0.9195\ K\ (\ 1 - 32.97\ K + 308.54\ K^2 - 880.36\ K^3) \tag{5}$$

$$B'_1 = 10.36\ K\ (\ 1 - 32.07\ K + 324.7\ K^2 - 1030.3\ K^3) \tag{6}$$

$$C'_1 = -16.88\ K\ (\ 1 - 30.94K + 302.3\ K^2 - 983.46\ K^3) \tag{7}$$

Mass Flow Rate variation (m_o)

The effect of speed of rotation on the mass flow rate at maximum efficiency, is shown in figure (7). The relation is found to be linear for the reference sample and other exit cut samples.

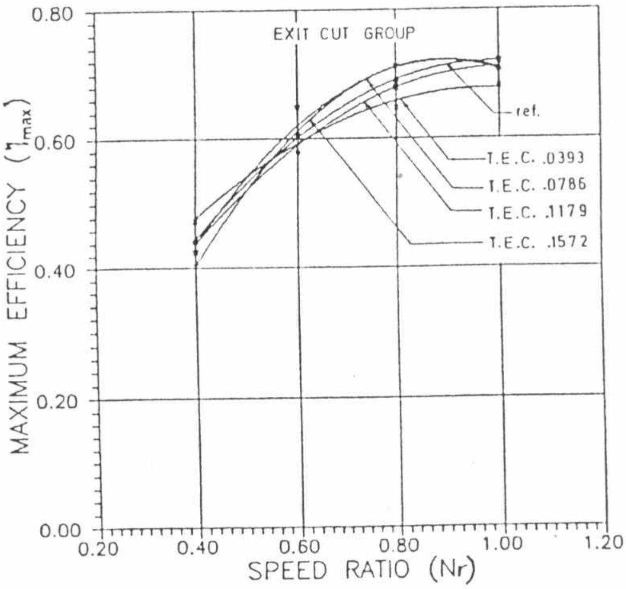


Fig.(5) Effect of varying speed of rotation on η_{max}

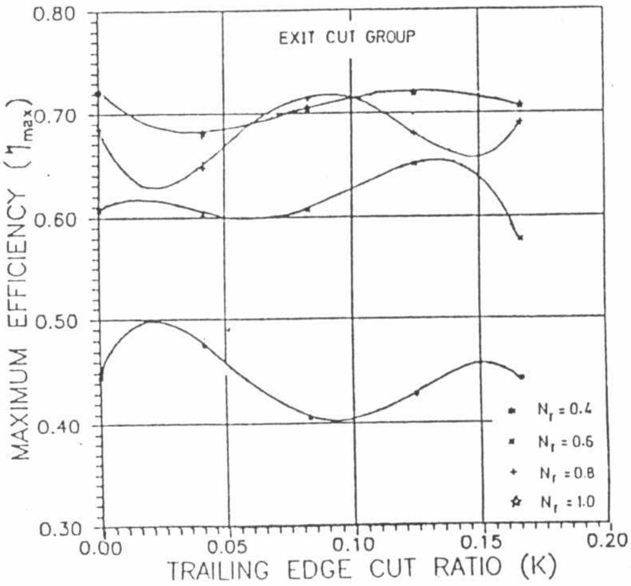


Fig.(6) Effect of trailing edge cut on η_{max}

Figure (8) shows the variation of the mass flow rate with respect to the trailing edge cut. The figure shows that the flow rate parameter is fluctuating in a manner that it opposes the corresponding maximum efficiency variations. The peak values correspond to the minimum value of the maximum efficiency. This appeared almost at all the speed ranges.

The fact that the maxima of the maximum efficiency coincides almost with the minima of the mass flow rate parameter, show the interdependence of the fluid flow mechanism and the energy transfer characteristics [4].

The incremental changes in the mass flow rate (Δmo) was correlated as :

$$\Delta mo = mo - (mo)_{ref} \tag{8}$$

$$\Delta mo = A'_2 + B'_2 \, Nr + C'_2 \, Nr^2 \tag{9}$$

where: A'_2, B'_2, and C'_2 are functions of blade and channel geometry.

$$A'_2 = - 0.1776 \, K \, (1 - 34.539 \, K + 383.775 \, K^2 - 1272.1K^3) \tag{10}$$

$$B'_2 = - 0.0249 \, K \, (1 + 83.48K - 1805.59K^2 + 8241.64K^3) \tag{11}$$

$$C'_2 = 0.0 \tag{12}$$

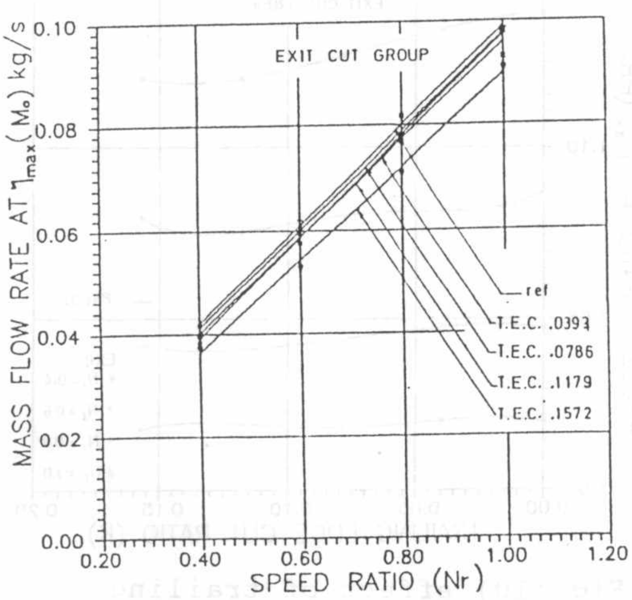


Fig.(7) Effect of speed of rotation on (mo)

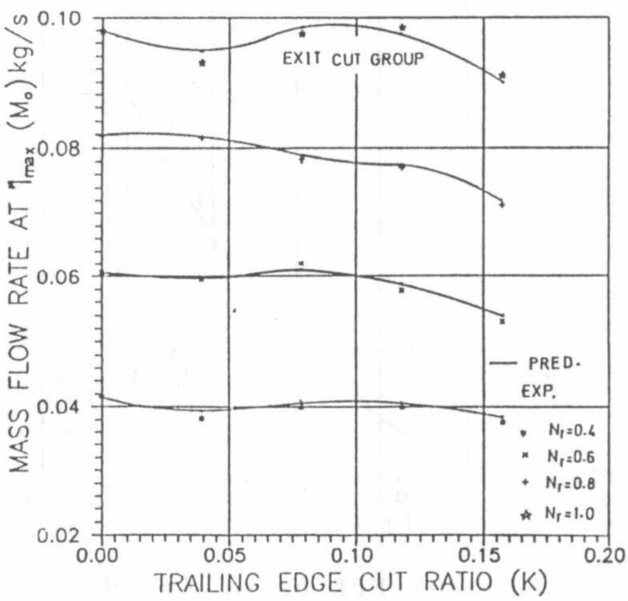


Fig.(8) Effect of trailing edge cut on (mo)

Pressure Ratio Variation (PRo)

The effect of speed of rotation is shown in figure (9), the relationship is parabolic for all exit cuts and the pressure ratio parameter decreases with increasing the exit cut.

The variation of the exit cut group,as shown in figure (10), is nearly similar to that of the corresponding maximum efficiency variations.

The incremental changes of the pressure ratio (PRo) cuold be related as follows :

$$PR_o = 1.0 + B_3 N + C_3 Nr^2 \tag{13}$$

$$\Delta PR_o = PR_o - \left(PR_o\right)_{ref} \tag{14}$$

$$\Delta PR_o = A'_3 + B'_3 Nr + C'_3 Nr^2 \tag{15}$$

where A' ,B' and C' are correlated experimentaly as follows :

$$A'_3 = 0.0 \tag{16}$$

$$B'_3 = 0.28203\ K(1-22.766K + 217.3K^2 - 624.66K^3) \tag{17}$$

$$C'_3 = -\ 0.5355\ K(1-20.065K + 2188K^2 - 1742.0K^3) \tag{18}$$

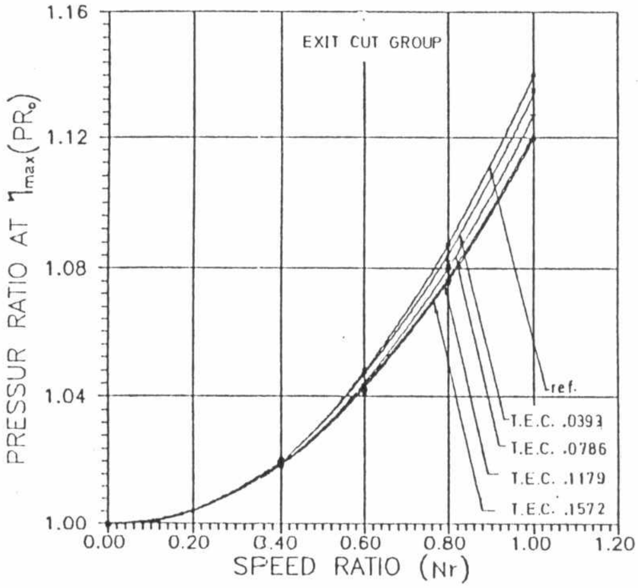


Fig.(9) Effect of speed of rotation on (PR_o)

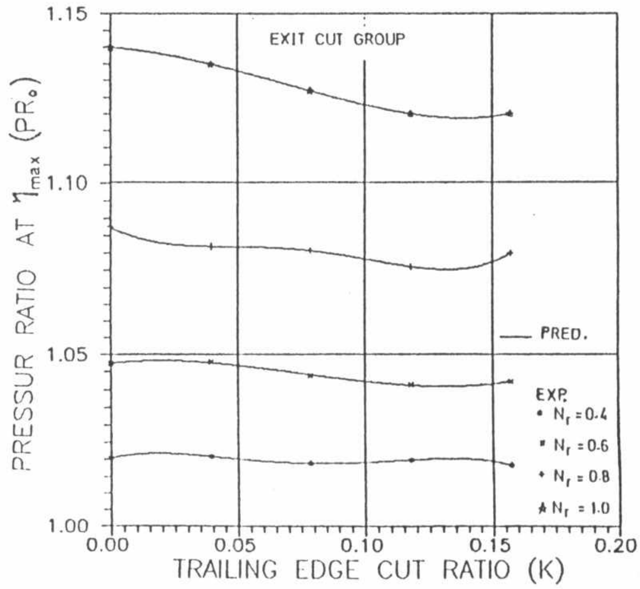


Fig.(10) effect of trailing edge cut on (PR_o)

Conclusion

The maximum efficiency is found to be affected by the trailing edge cut in a fluctuated behaviour. Tt is increased by a maximum value of 7 % at a speed ratio of 0.6 and exit cut ratio of 0.1179 . At design speed, it was found that it is not affected by the trailing edge cut. So, the trailing edge cut could improve the compression efficiency at off design speeds.

The mass flow rate, is found to be affected inversely to that found for the maximum efficiency. At large exit cuts, the mass flow rate parameter is decreased by about 9 % for a cut ratio of 0.125 .

The pressure ratio parameter is decreased considerably by increasing the exit cut ratio as the energy transfer to the flow is decreased.

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