CENTRIFUGAL COMPRESSORS AND FANS

<u>Outline</u>

Introduction

- Elements of the Centrifugal Compressor
- Inlet Casing
- Impeller
- Diffuser

Inlet Velocity Limitations

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Inlet Velocity Limitations

- During the Second World War, great progress was made in the development of gas turbines using the centrifugal compressor.
- Because they need to be supplied with large amounts of high-pressure air.
- The centrifugal compressor had previously been researched for use in small high-speed internal combustion engines.
- For that the centrifugal compressor became a natural choice.

- Although the centrifugal compressor has been superseded by the axial flow compressor in jet aircraft engines,
- It is useful where a short overall engine length is required and where it is likely that deposits will be formed in the air passages, since, because of the relatively short passage length, loss of performance due to buildup of deposits will not be as great as in the axial compressor.
- The centrifugal compressor is mainly found in turbochargers, where it is placed on the same shaft as an inward flow radial gas turbine, which is driven by engine exhaust gases.

- Pressure ratios of 4:1 are typical in a single stage, and ratios of 7:1 are possible if exotic materials are used for impeller manufacture.
- The best efficiencies are 3- 4 percent below those obtainable from an axial flow compressor designed for the same duty.
- However, at very low mass flow rates the axial flow compressor efficiency drops off rapidly. It is also difficult to hold the tolerances required for small axial flow blading, and manufacture of the axial compressor blades becomes more expensive.

- If the density ratio across the compressor is less than about 1.05, the term 'fan' is used to describe the machine.
- In that case the fluid is treated as being incompressible; otherwise compressible flow equations must be used.
- The term 'blower' is often used in place of 'fan'.

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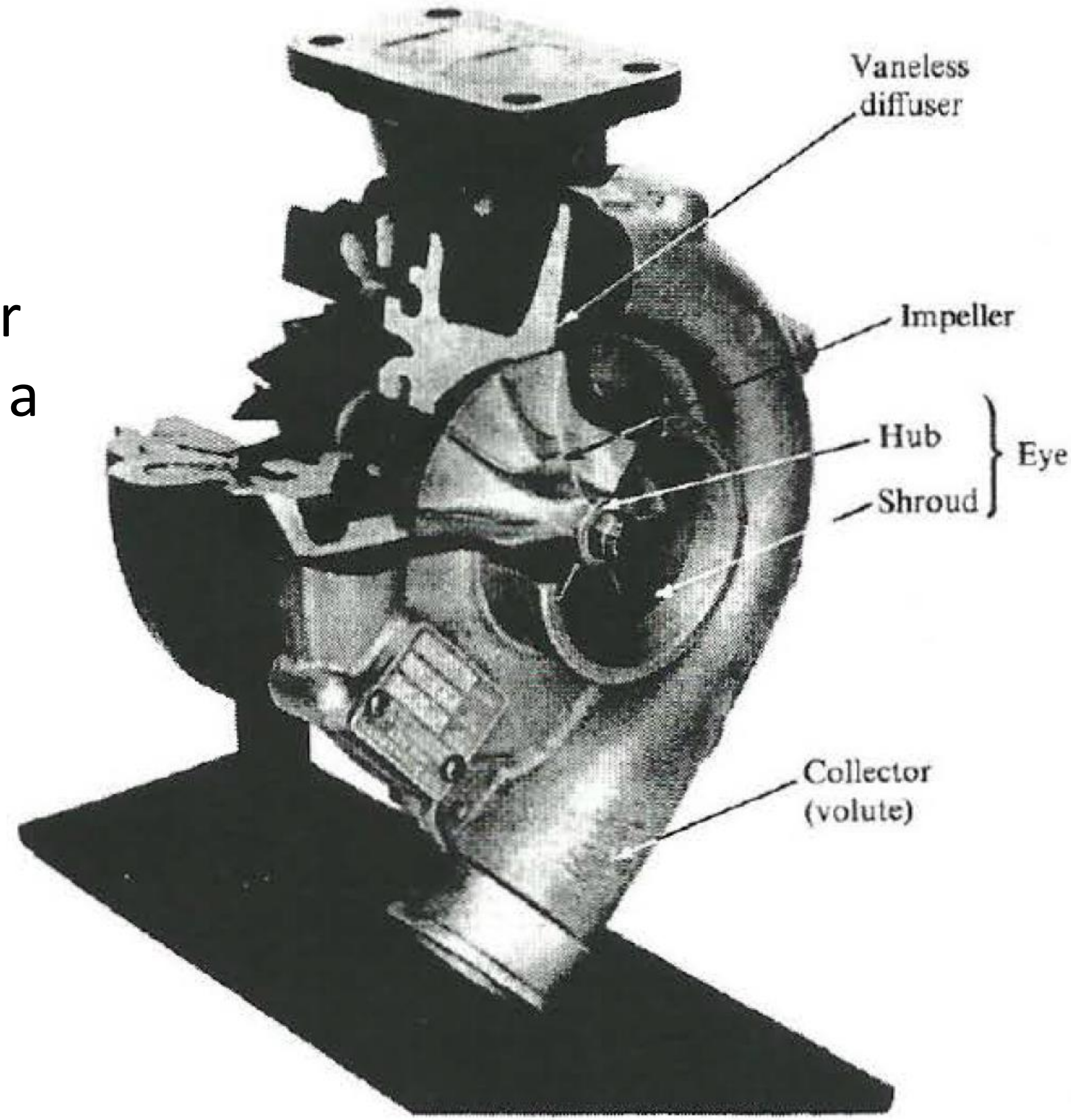
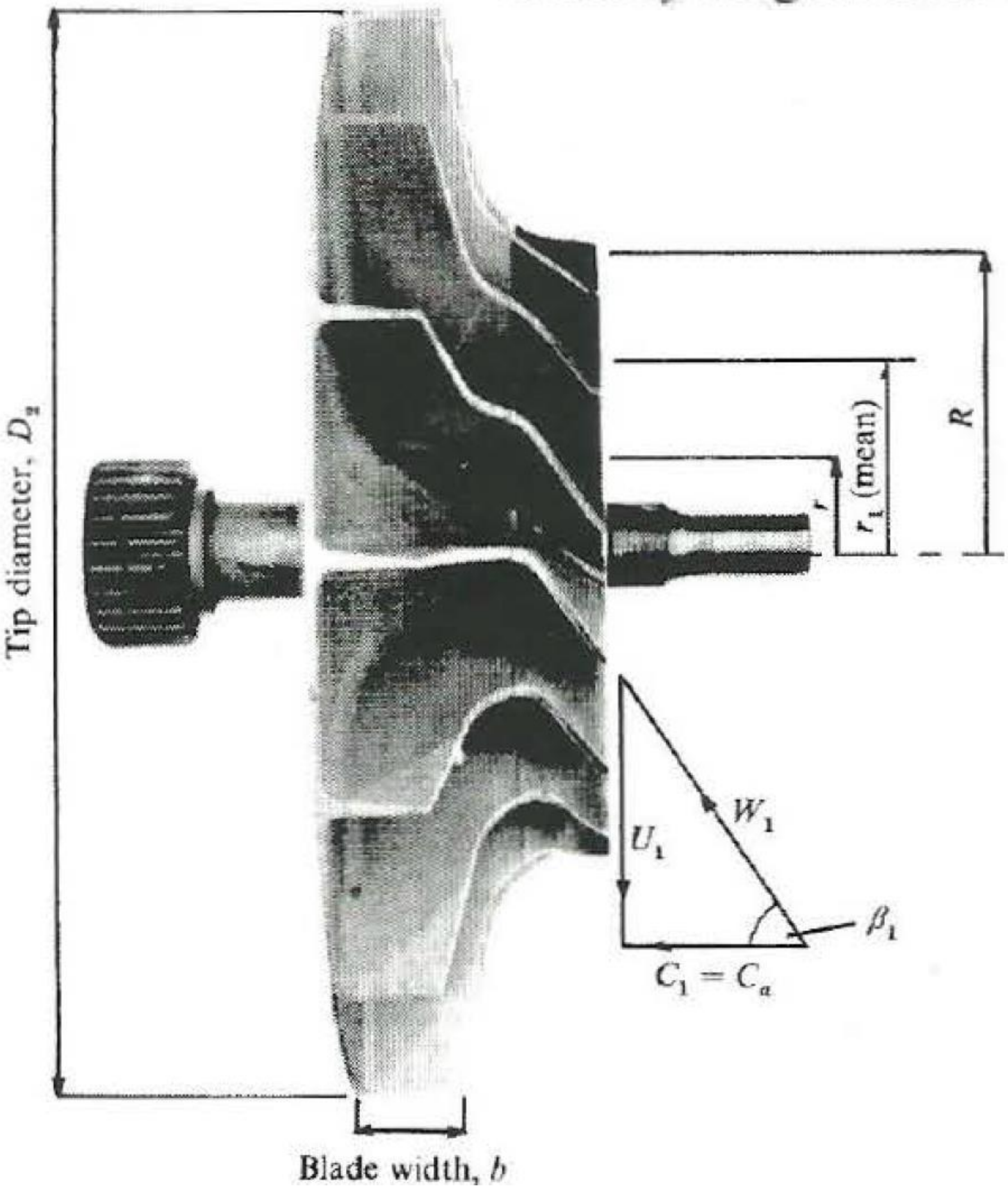
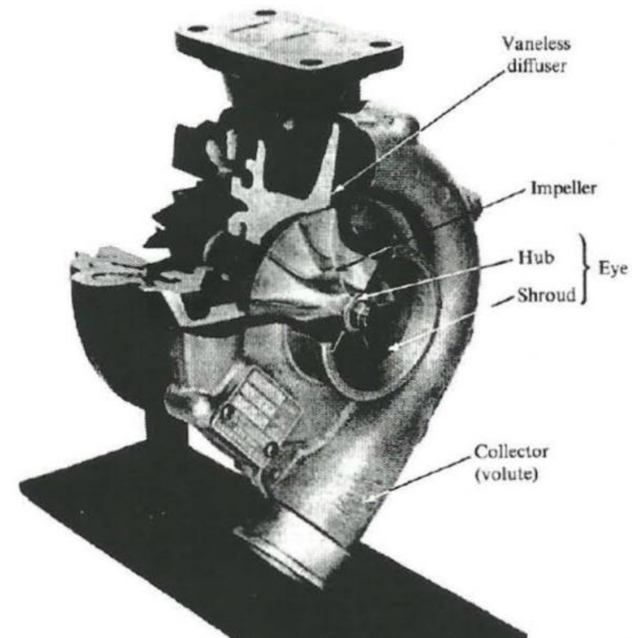
Elements of the Centrifugal Compressor

The elements of a centrifugal compressor are similar to those of a hydraulic
pump, with some important differences.

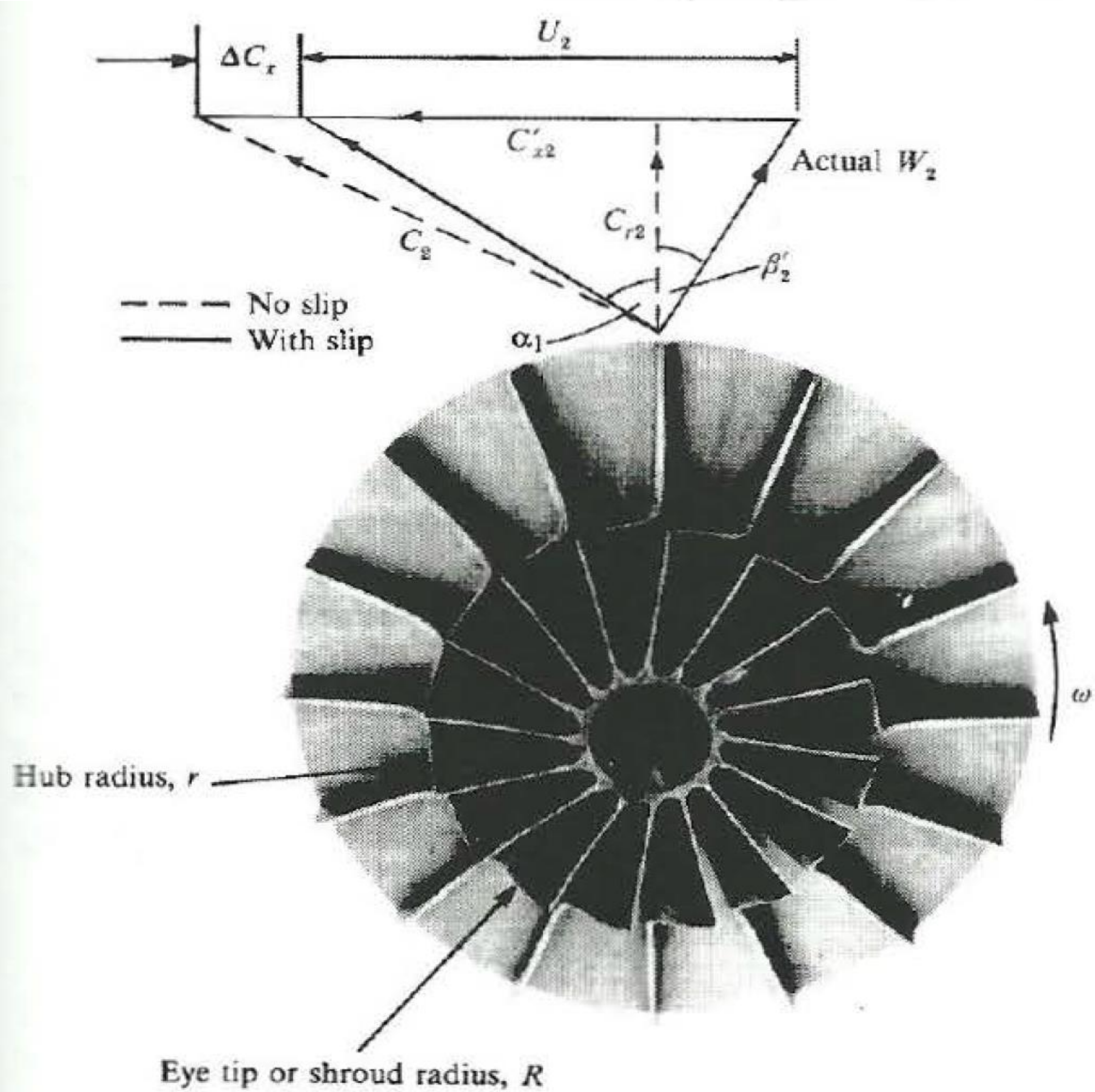


Velocity diagrams for a centrifugal compressor





Velocity diagrams for a centrifugal compressor



slip factor (Eq. (2.9)) is best applied for radial vanes and with $\beta_2 = 0$,

$$\sigma_s = 1 - 0.63\pi/Z$$

By Euler's pump equation (Eq. (1.25)) without slip,

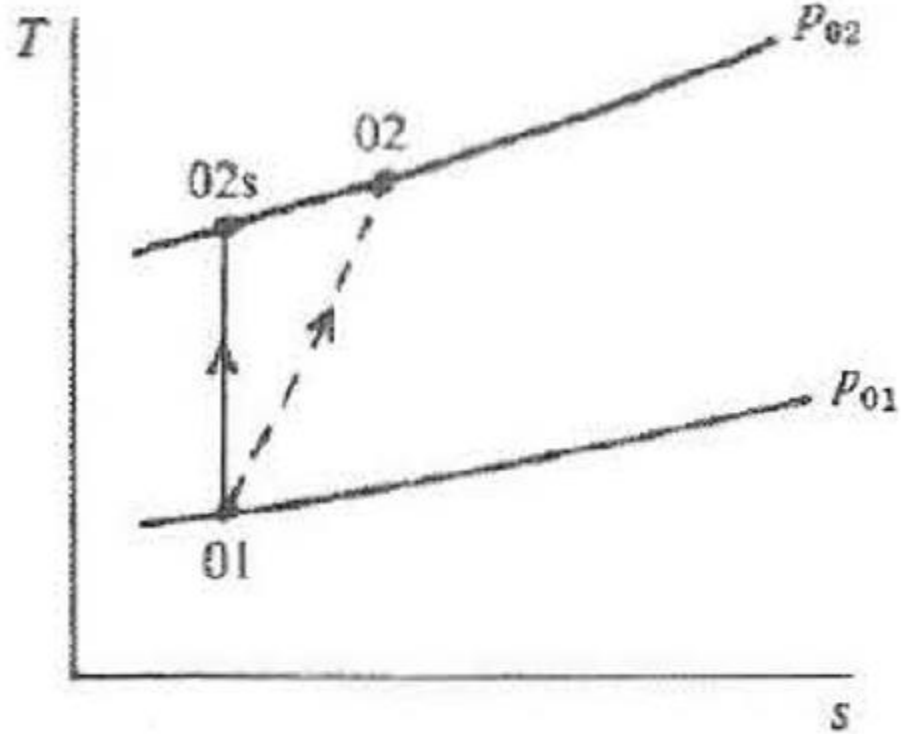
$$E = U_2 C_{x2}/g = U_2^2/g$$

and with slip,

$$E = \sigma_s U_2^2/g$$

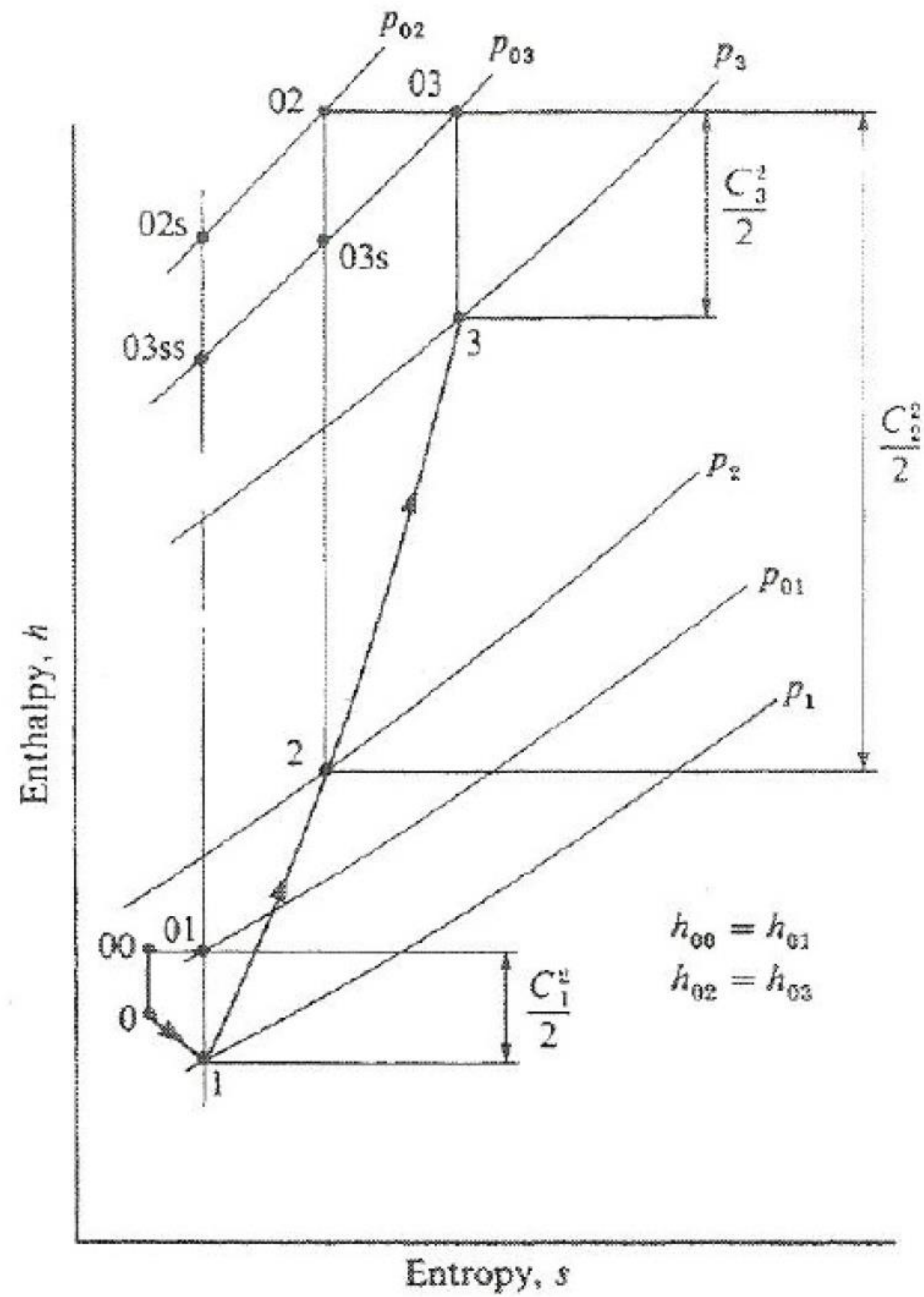
power input factor ψ

$$E = \psi \sigma_s U_2^2/g$$



Compression in compressor

Mollier chart for a centrifugal compressor



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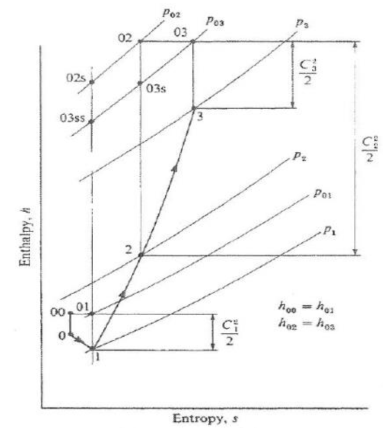

The energy equation along a streamline may be written as

$$\text{Total enthalpy } h_0 = h + C^2/2 = \text{Constant}$$

Therefore, for the fluid that is being drawn from the atmosphere into the inducer section, the total enthalpy is

$$h_{00} = h_0 + C_0^2/2$$

Total enthalpy at section 1 is

$$h_{01} = h_1 + C_1^2/2$$

and since no shaft work has been done and assuming adiabatic steady flow,

$$h_{00} = h_{01}$$

Thus

$$h_0 + C_0^2/2 = h_1 + C_1^2/2$$

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From section 1 to 2 the fluid moves through the impeller where work is done on it to increase its static pressure from p_1 to p_2. Writing the work done per unit mass on the fluid in terms of enthalpy we get

$$W/m = h_{02} - h_{01}$$
$$= U_2 C_{x2} - U_1 C_{x1}$$

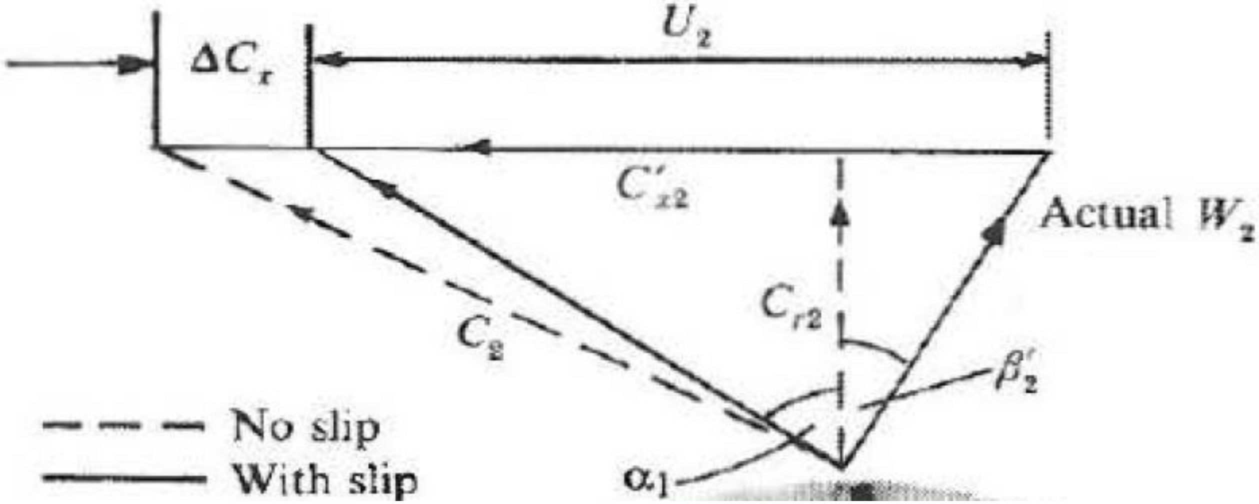
or after substituting for h_0,

$$I = h_1 + C_1^2/2 - U_1 C_{x1} = h_2 + C_2^2/2 - U_2 C_{x2}$$

where I is a constant throughout the impeller. In general

$$\begin{aligned} I &= h + C^2/2 - UC_x \\ &= h + (C_r^2 + C_x^2)/2 - UC_x \\ &= h + (W^2 - W_x^2 + C_x^2)/2 - UC_x \\ &= h + [W^2 - (U - C_x)^2 + C_x^2]/2 - UC_x \\ &= h + W^2/2 - U^2/2 - C_x^2/2 + UC_x - UC_x + C_x^2/2 \\ &= h + W^2/2 - U^2/2 \\ &= h_{0,\text{rel}} - U^2/2 \end{aligned}$$

where $h_{0,\text{rel}}$ is the total enthalpy based on the relative velocity of the fluid.



where I is a constant throughout the impeller. In general

$$
\begin{aligned}
I &= h + C^2/2 - UC_x \\
&= h + (C_r^2 + C_x^2)/2 - UC_x \\
&= h + (W^2 - W_x^2 + C_x^2)/2 - UC_x \\
&= h + [W^2 - (U - C_x)^2 + C_x^2]/2 - UC_x \\
&= h + W^2/2 - U^2/2 - C_x^2/2 + UC_x - UC_x + C_x^2/2 \\
&= h + W^2/2 - U^2/2 \\
&= h_{0,\mathrm{rel}} - U^2/2
\end{aligned}
$$

where $h_{0,\mathrm{rel}}$ is the total enthalpy based on the relative velocity of the fluid.

$$h_2 - h_1 = (U_2^2 - U_1^2)/2 + (W_1^2 - W_2^2)/2$$

since $I_1 = I_2$. In Eq. the main contribution to the static enthalpy rise is from the term $(U_2^2 - U_1^2)/2$.

In preliminary design calculations it is usual to assume $C_{x1} = 0$, although this is not always the case, whence from Eq.

$$E = \psi \sigma_s U_2^2 / g$$

the work done on the fluid per unit mass becomes

$$h_{02} - h_{01} = \psi \sigma_s U_2^2 \text{ (J/kg)}$$

After writing $C_p T_0$ in place of h_0, we get that the work input is given by

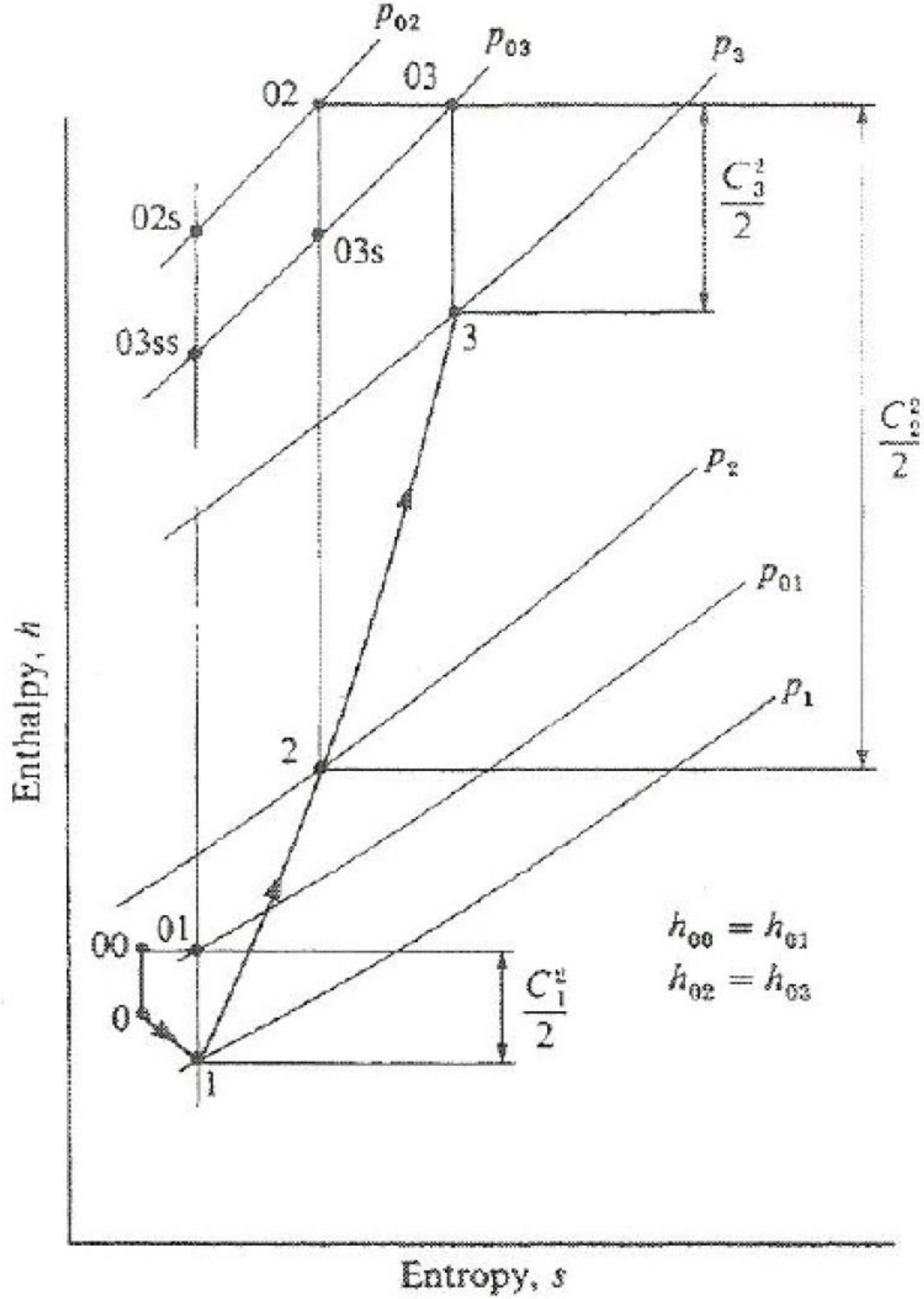
$$T_{02} - T_{01} = \psi \sigma_s U_2^2 / C_p$$

where C_p is the mean specific heat over this temperature range.

Also, since no work is done in the diffuser, $h_{02} = h_{03}$,

$$T_{03} - T_{01} = \psi \sigma_s U_2^2 / C_p$$

Mollier chart for a centrifugal compressor



a compressor overall total-to-total isentropic efficiency η_c may be defined as

$$\eta_c = \frac{\text{Total isentropic enthalpy rise between inlet and outlet}}{\text{Actual enthalpy rise between same total pressure limits}}$$

$$= (h_{03ss} - h_{01})/(h_{03} - h_{01})$$

where the subscript 'ss' represents the end state on the total pressure line p_{03} when the process is isentropic. Thus

$$\begin{aligned} \eta_c &= (T_{03ss} - T_{01})/(T_{03} - T_{01}) \\ &= T_{01}(T_{03ss}/T_{01} - 1)/(T_{03} - T_{01}) \end{aligned}$$

But

$$\begin{aligned} p_{03}/p_{01} &= (T_{03ss}/T_{01})^{\gamma/(\gamma-1)} \\ &= [1 + \eta_c(T_{03} - T_{01})/T_{01}]^{\gamma/(\gamma-1)} \\ &= (1 + \eta_c \psi \sigma_s U_2^2/C_p T_{01})^{\gamma/(\gamma-1)} \end{aligned}$$

The slip factor should be as high as possible since it limits the energy transfer to the fluid even under isentropic conditions and it is seen from the velocity diagrams that C_{x2} approaches U_2 as the slip factor is increased. The slip factor may be increased by increasing the number of vanes but this increases the solidity at the impeller eye, resulting in a decrease in the flow area at inlet. For the same mass flow rate, the flow velocity C_a at inlet must therefore be increased and this increases the loss due to friction. A compromise is usually reached, slip factors of about 0.9 being typical for a compressor with 19–21 vanes.

While it may seem that a high value of power input factor ψ is desirable, it is found that the rate of decrease of isentropic efficiency with increase in ψ negates any apparent advantage, so the ideal should be to have a power input factor of unity.

The pressure ratio increases with the impeller tip speed but material strength considerations preclude this being increased indefinitely. Centrifugal stresses are proportional to the square of the tip speed and, for a light alloy impeller, tip speeds are limited to about 460 m/s. This gives a pressure ratio of about 4:1. Pressure ratios of 7:1 are possible if materials such as titanium are used.

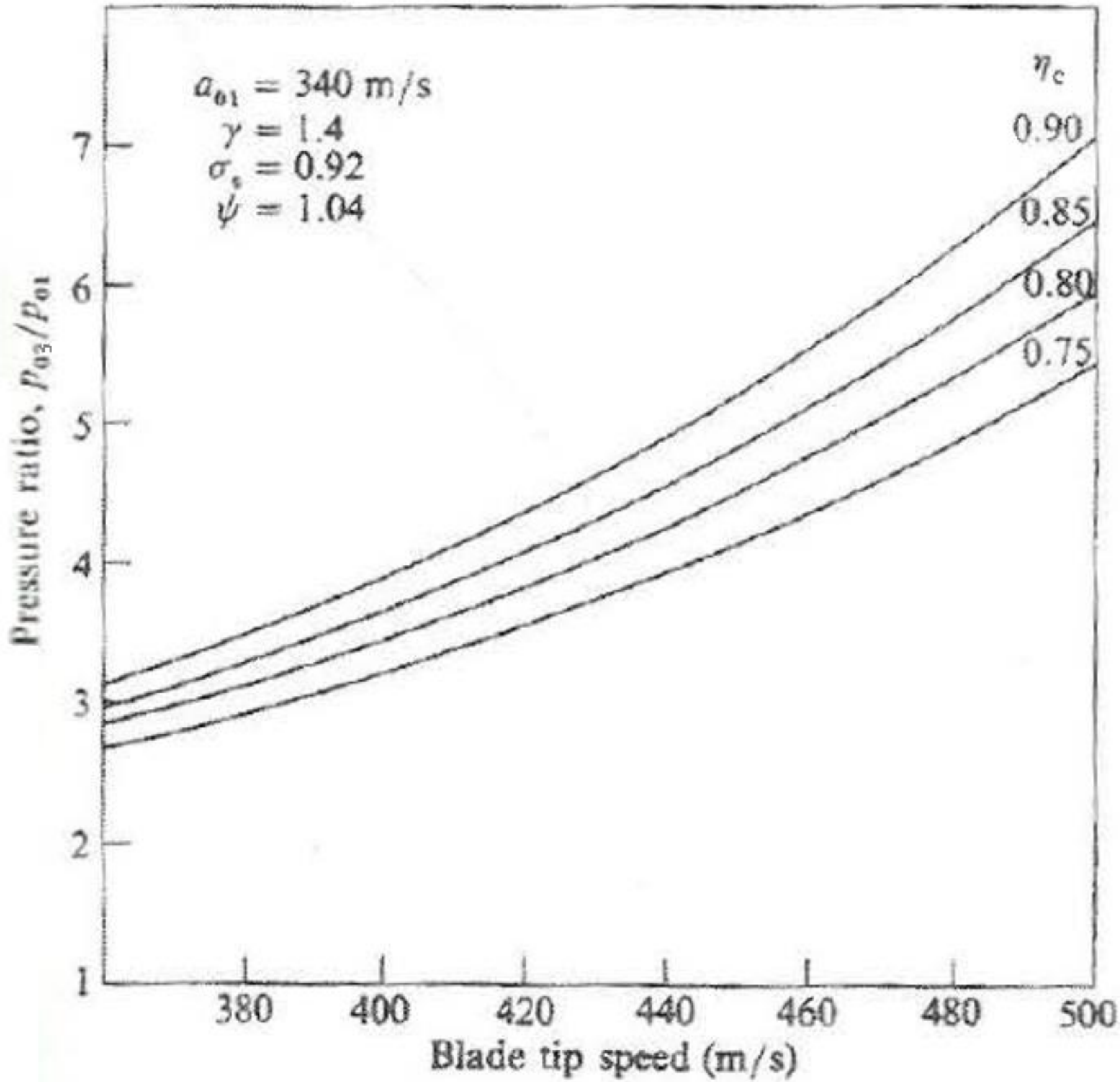
Equation can be written in terms of fluid properties and flow angles as follows: since

$$a_{01}^2 = \gamma R T_{01} \qquad \text{and} \qquad C_p = \gamma R/(\gamma - 1)$$

then

$$p_{03}/p_{01} = [1 + \eta_c \psi \sigma_s (\gamma - 1) U_2^2/a_{01}^2]^{\gamma/(\gamma-1)}$$

The change of pressure ratio with blade tip speed is shown in Fig. for various isentropic efficiencies.



Overall pressure ratio versus impeller tip speed

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Diffuser

The stagnation temperature of the gas at outlet from the diffuser should have as small a kinetic energy term as possible, as this eases the problem of combustion chamber design. Typical compressor outlet velocities are of the order of 90 m/s. The diffusion process is carried out in a diffuser as described in Secs 2.5.2 and 2.5.3, some diffusion also taking place in the vaneless space between the impeller tip and diffuser vanes. The flow theory described in those sections is applicable here. The maximum included angle of the vaned diffuser passage is about 11°, any increase in this angle leading to a loss of efficiency through boundary-layer separation on the passage walls. It should also be remembered that any change from the design mass flow rate and pressure ratio will change the smooth flow direction into the diffuser passage and will therefore also result in a loss of efficiency. This may be rectified by utilizing variable-angle diffuser vanes.

For adiabatic deceleration of the fluid from absolute velocity C_2 to C_3 with a corresponding increase of static pressure from p_2 to p_3,

$$h_{02} = h_{03}$$

or

$$h_2 + C_2^2/2 = h_3 + C_3^2/2$$

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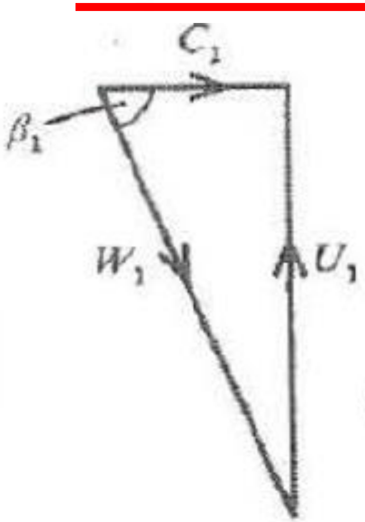
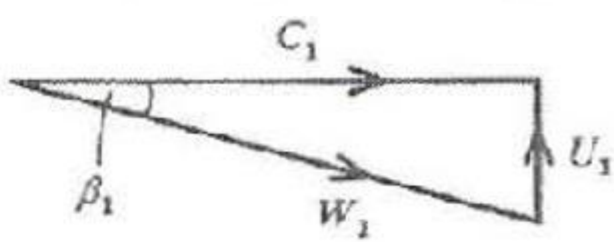
Mach-number considerations at the eye of a centrifugal compressor make the relative velocity W_1 a very sensitive value as far as compressor performance is concerned. Should the Mach number at entry to the impeller be greater than unity, then shock waves will form, with all their attendant losses. Assume that we have a uniform absolute velocity C_1 with zero whirl ($C_{x1} = 0$) at entry to a centrifugal compressor. Two cases may be examined for the same mass flow rate, both cases being extremes.

1. If the eye tip diameter is large, then from continuity considerations the axial velocity C_1 is low and the blade speed is high, resulting in the velocity diagram of Fig. a.

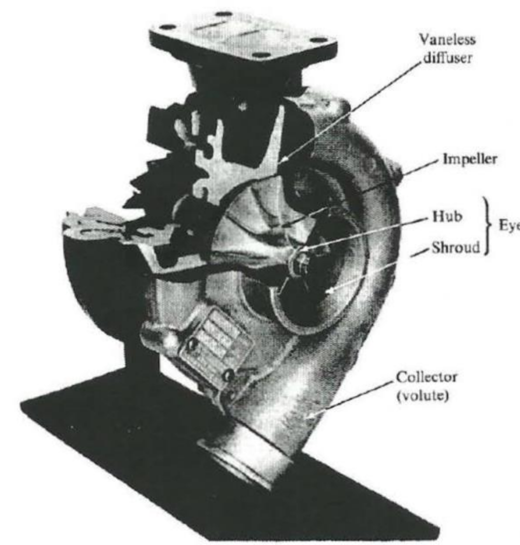
2. If the eye tip diameter is small, the blade speed is small but the axial velocity C_1 is large, and the velocity diagram of Fig. b may be drawn.





Velocity triangles for

(a) large areas (b) small inlet areas



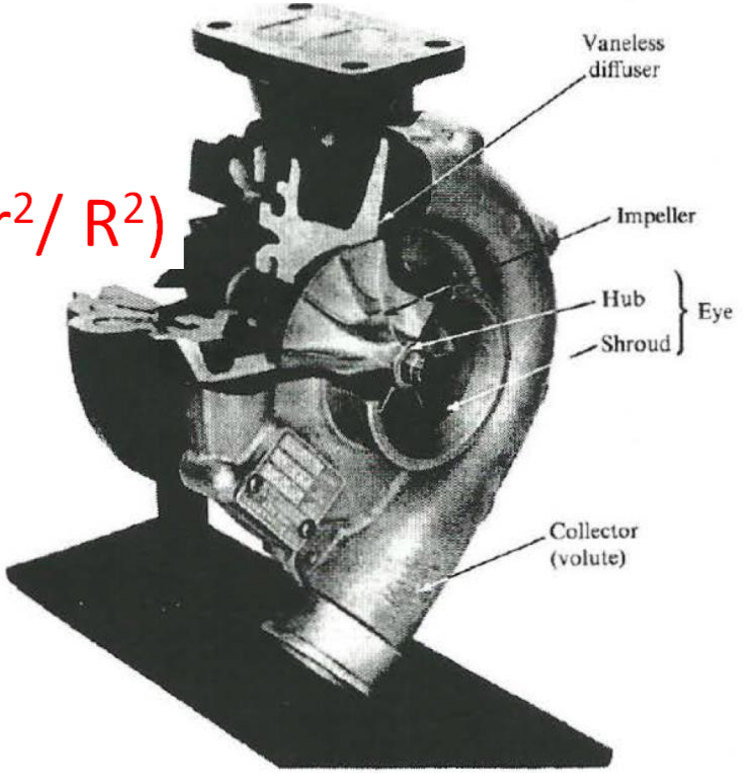
Flow into the eye takes place through the annulus formed by the shroud radius R and the hub radius r. For uniform axial flow into the eye

$$m = \rho_1 A_1 C_1$$

The flow area is
$$A_1 = \pi R^2(1 - r^2/R^2)$$
$$= \pi R^2 k$$

where $k = (1 - r^2/R^2)$.

$$A = \pi R^2 - \pi r^2$$
$$= \pi R^2(1 - r^2/R^2)$$



Substitution for A_1 into

$$U = \omega R$$

$$\frac{U}{\omega} = R$$

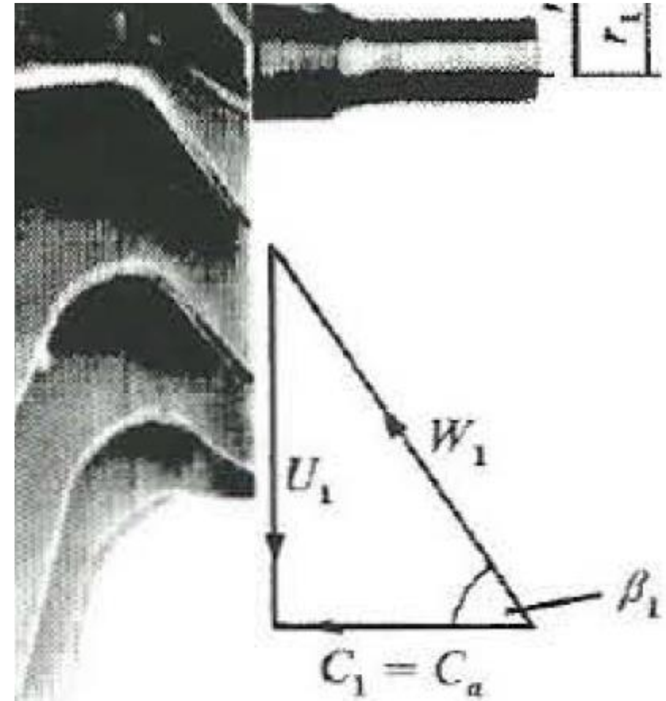
$$m = \rho_1 A_1 C_1$$
$$m = \rho_1 \pi R^2 k C_1$$
$$= \rho_1 \pi U_1^2 k C_1 / \omega^2$$

from the inlet velocity triangle

$$C_1 = W_1 \cos \beta_1$$

and

$$U_1 = W_1 \sin \beta_1$$



$$m = \rho_1 \pi U_1^2 k C_1 / \omega^2$$

$$m\omega^2 / \rho_1 \pi k = U_1^2 C_1$$
$$= W_1^3 (\sin^2 \beta_1)(\cos \beta_1)$$

From isentropic flow relationships the ratio of stagnation to static pressure at inlet may be written as

$$p_{01}/p_1 = [1 + (\gamma - 1) M_1^2 / 2]^{\gamma/(\gamma - 1)}$$

From isentropic flow relationships the ratio of stagnation to static pressure at inlet may be written as

$$p_{01}/p_1 = [1 + (\gamma - 1)M_1^2/2]^{\gamma/(\gamma-1)}$$

and

$$T_{01}/T_1 = 1 + (\gamma - 1)M_1^2/2$$

Divide

Then

$$p_1/T_1 = (p_{01}/T_{01})[1 + (\gamma - 1)M_1^2/2]^{-\gamma/(\gamma-1)}[1 + (\gamma - 1)M_1^2/2]$$

$$= (p_{01}/T_{01})[1 + (\gamma - 1)M_1^2/2]^{-1/(\gamma-1)}$$

$$1 - \frac{\gamma}{\gamma - 1}$$

$$= \frac{\gamma - 1}{\gamma - 1} - \frac{\gamma}{\gamma - 1}$$

$$= \frac{-1}{\gamma - 1}$$

PV=mRT

P=ρRT

Now

$$\rho_1 = p_1/RT_1$$

$$= (p_{01}/RT_{01})[1 + (\gamma - 1)M_1^2/2]^{-1/(\gamma-1)}$$

Therefore substituting for ρ_1

$$m\omega^2/\rho_1 \pi k = W_1^3(\sin^2\beta_1)(\cos\beta_1)$$

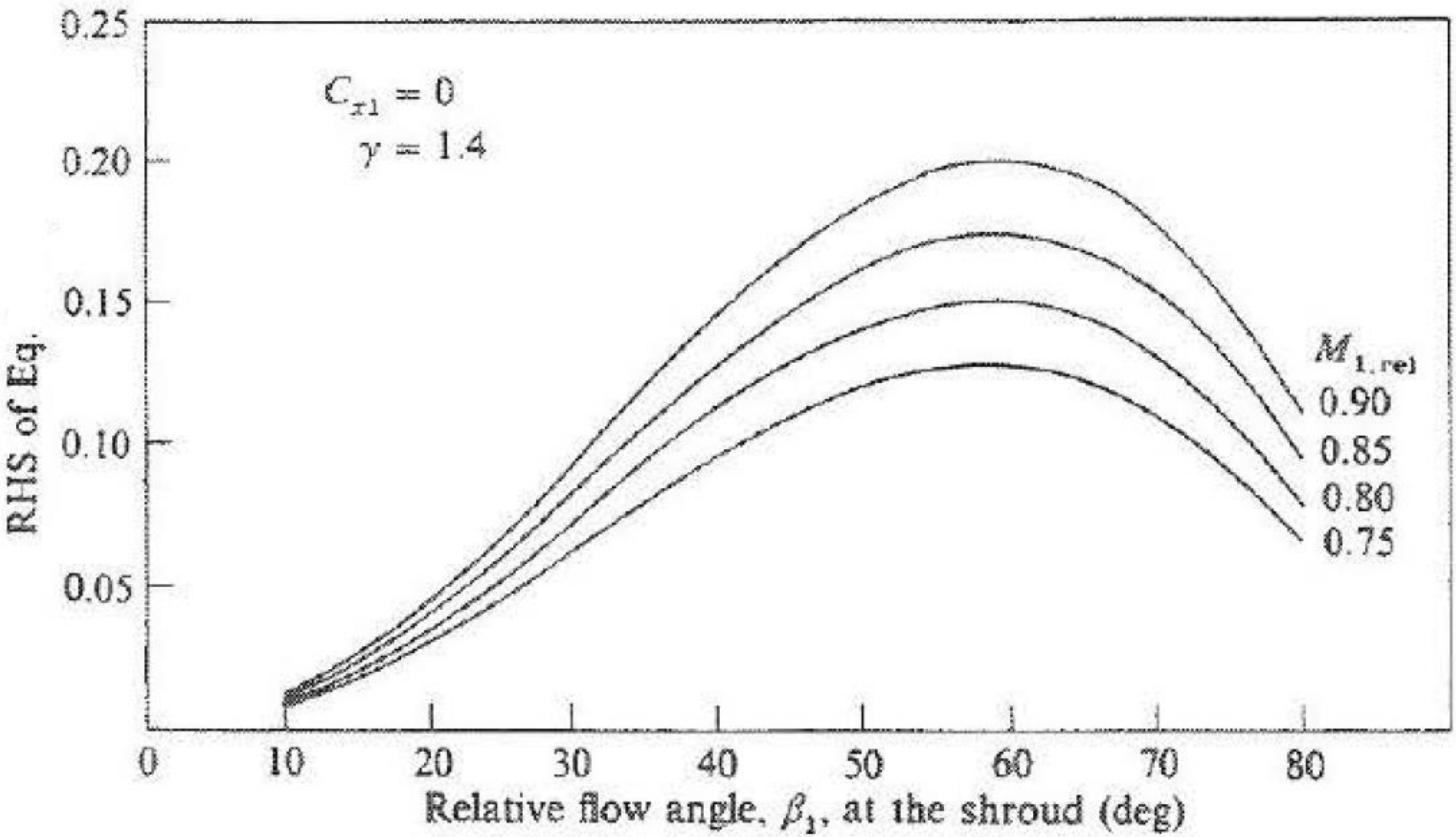
$$m\omega^2 R T_{01}/\pi k p_{01} = W_1^3(\sin^2\beta_1)(\cos\beta_1)/[1+(\gamma-1)M_1^2/2]^{1/(\gamma-1)}$$

Writing the relative Mach number based on the relative velocity W_1, then

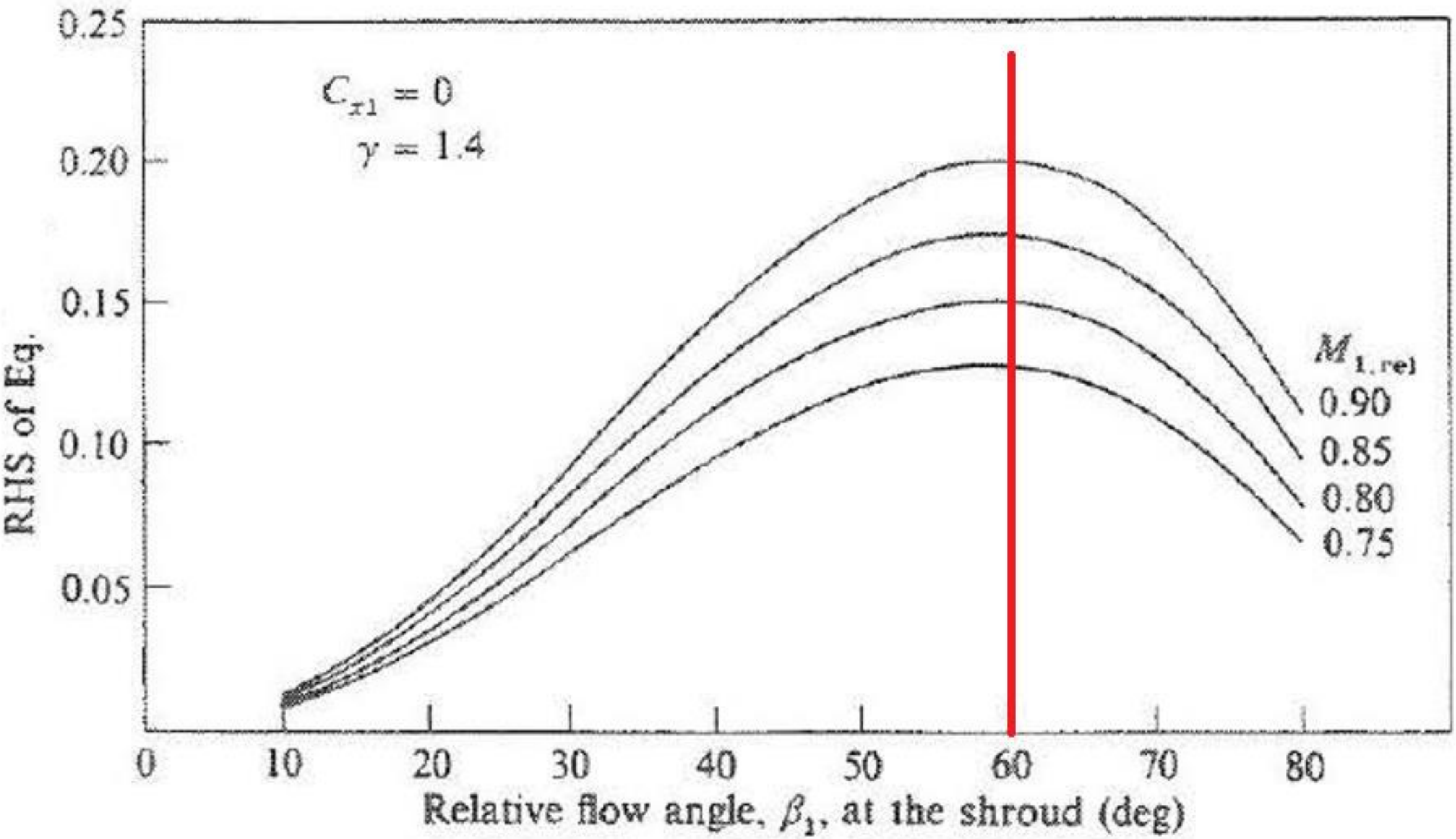
$$m\omega^2 R T_{01}/\pi k p_{01} = M_{1,\mathrm{rel}}^3 a_1^3(\sin^2\beta_1)(\cos\beta_1)/[1+(\gamma-1)M_1^2/2]^{1/(\gamma-1)}$$

From Eq. (4.18), $a_{01}/a_1 = [1+(\gamma-1)M_1^2/2]^{1/2}$ since $a = (\gamma R T)^{1/2}$ and after substituting for a_1 in Eq.   and putting $M_1 = M_{1,\mathrm{rel}}\cos\beta_1$,

$$(m\omega^2)/[\pi k \gamma p_{01}(\gamma R T_{01})^{1/2}]$$
$$= M_{1,\mathrm{rel}}^3(\sin^2\beta_1)(\cos\beta_1)/[1+(\gamma-1)M_{1,\mathrm{rel}}^2(\cos^2\beta_1)/2]^{1/(\gamma-1)+3/2}$$



Optimization of the mass flow function



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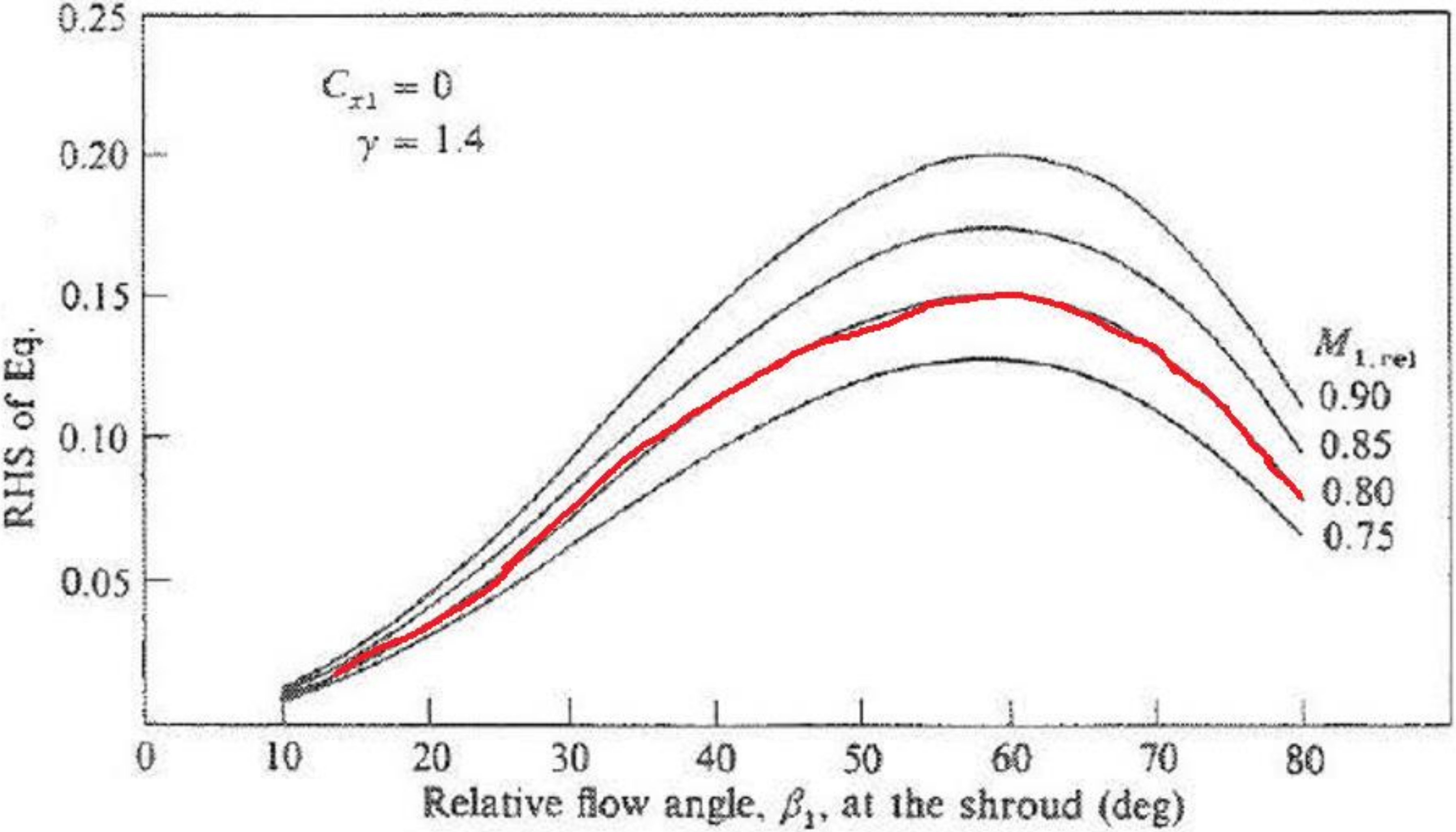

Optimization of the mass flow function