TURBOMACHINERY
Introduction

Ho inserito la data in modo che possiate tracciare i cambiamenti che faccio di volta in volta. Per favore segnalate eventuali errori, grazie.

SAPIENZA
UNIVERSITÀ DI ROMA

Monday, September 30, 2019

giovanni.delibra@uniroma1.it

Roma : 06 44 585 901. Lunedì, martedì, mercoledì, giovedì, venerdì?

Nota: il martedì ho lezione anche a Latina, quindi finite la lezione a Roma scappo via

Latina office: 0773 47 65 21 (martedì - venerdì?)

(vale solo per questo semestre, nei semestri successivi dipende dall'orario dei corsi. Per prendere un appuntamento mandatemi una mail. Il venerdì potrei essere a Roma o Latina a seconda di esigenze che variano perchè il corso a Latina è condiviso con un altro docente, se fa lezione lui io sono a Roma)

Materiale didattico

- *Slides*
- *Dixon: Fluid Mechanics and Thermodynamics of Turbomachinery 7th ed*
- *Lewis: Turbomachinery performance analysis*
- *Greitzer: Internal flow*
- *Laptop con Windows+macchina virtuale linux o linux*
- *Tutto il software di cui avrete bisogno è open-source e potete scaricarlo da internet*

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FAQ

Quanto dura il corso?

Fino al 12 dicembre, salvo problemi (max 19 dicembre)

Come funziona l'esame?

Orale: rispondete a max 3 domande

Dovete portare con voi un report relative alla parte di esercitazioni

Almeno una delle domande riguarderà il report in questione

Quando c'è il primo appello?

Prima di Natale

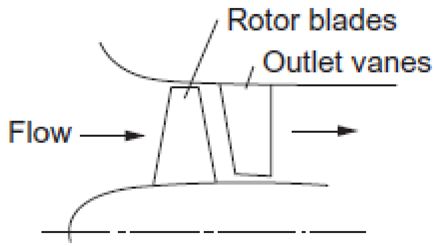
Gli altri?

Non lo so, le date che vedete non le ho inserite io e per il momento non sono in grado di vederle né modificarle

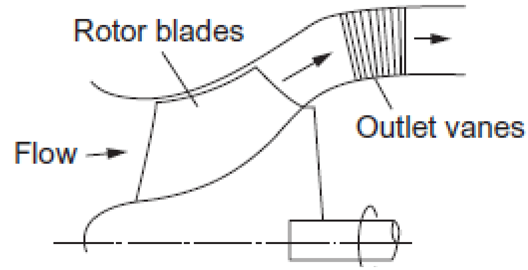
Turbomachinery are devices in which **energy transfer occurs between a flowing fluid and a rotating element** due to dynamic action, and results in a change in pressure and momentum of the fluid. Mechanical energy transfer occurs inside or outside of the turbomachine, usually in a steady-flow process.

The energy transfer can go from the rotor to the fluid (**macchine operatrici**) like in a compressor or a pump or vice-versa (**macchine motrici**) like in a turbine. The classical distinction of machine operatrici/motrici doesn’t have a straightforward translation, they directly refer to compressor and turbines.

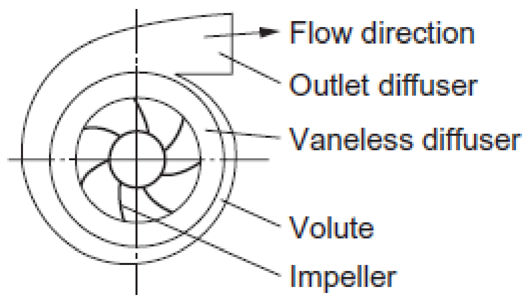
Turbomachinery can be classified in other ways, such as the main direction of the fluid inside the impeller (**axial** or **radial**) and the properties of the fluid (**incompressible** or **compressible**).



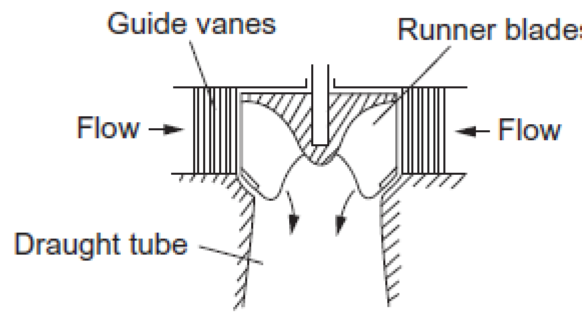
Axial fan



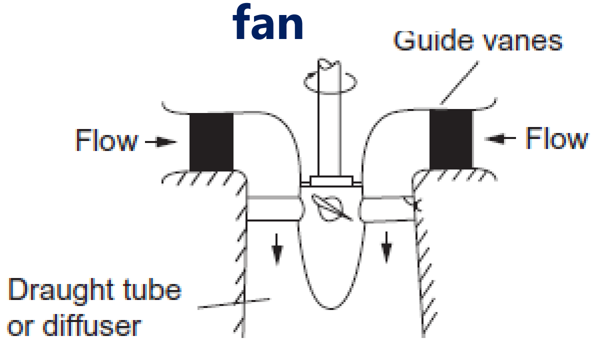
Mixed-flow fan/pump



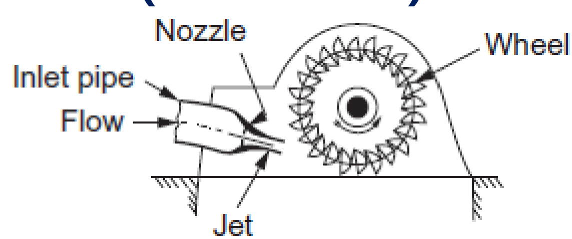
Centrifugal fan



Francis turbine (mixed flow)



Kaplan turbine



Pelton turbine

Dixon, 2014

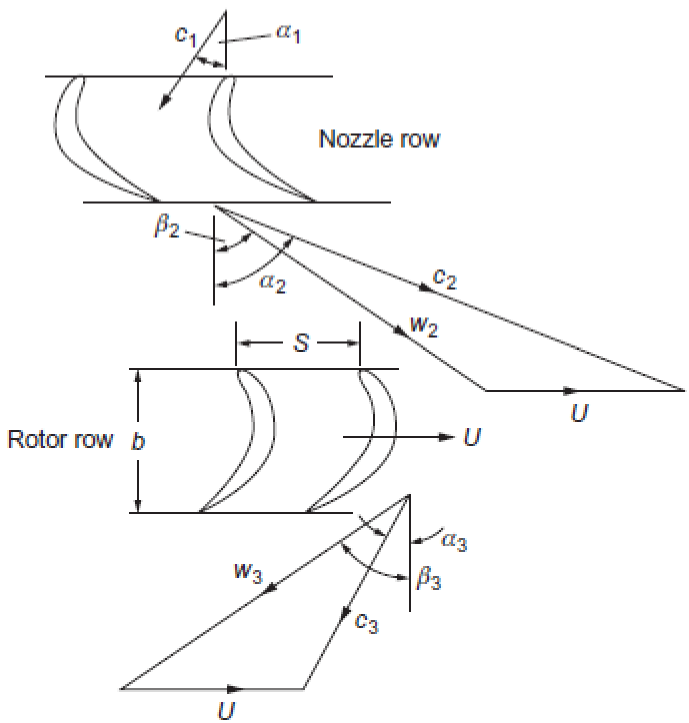
1. **Theory: three-dimensional and viscous effects in turbomachinery flows**
    - Secondary motions
    - Losses in TM

2. **Design: TM design and performance verification**
    - Similarity theory
    - Axial flow fans
    - Radial flow fans
    - Radial pumps

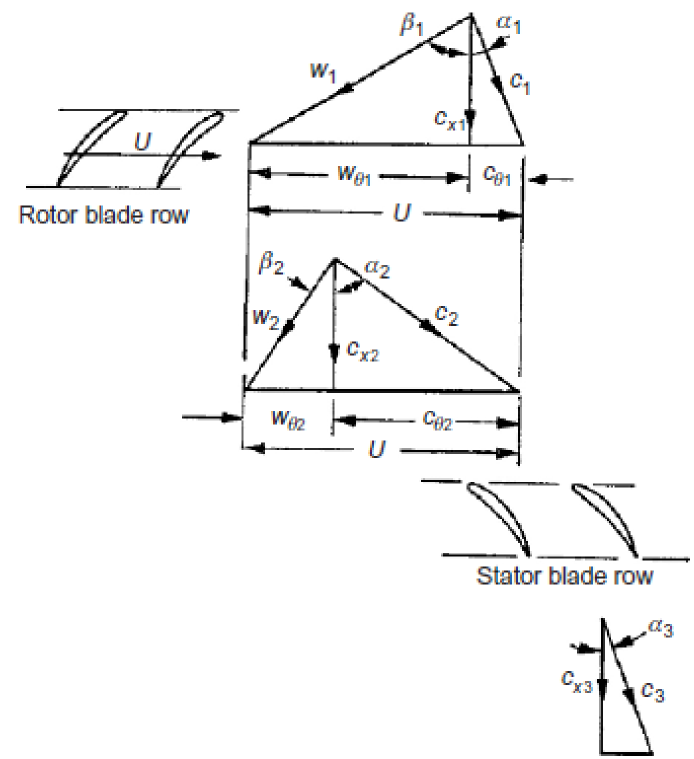
3. **Numerical: Practical exercises**
    - Euler Work
    - Computation of cascade flows
    - Design and optimization of axial flow fan

<u>This is a list and not a chronological order</u>

In TM class you learnt how to design the geometry of the blade of radial and axial TM, how to evaluate their ideal performance (Euler Work) with 1D method



da *Dixon*, 2014



da *Dixon*, 2014

Centrifugal impeller, velocity triangles

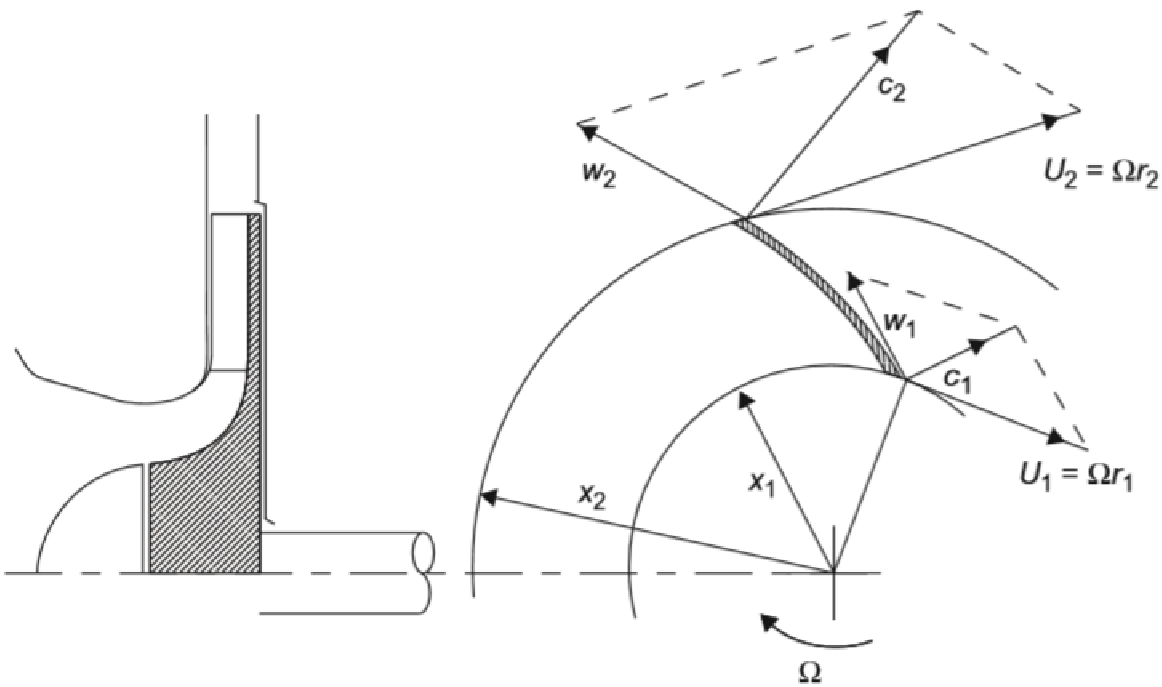


FIGURE 7.5

da *Dixon*, 2014

Centrifugal pump and velocity triangles.

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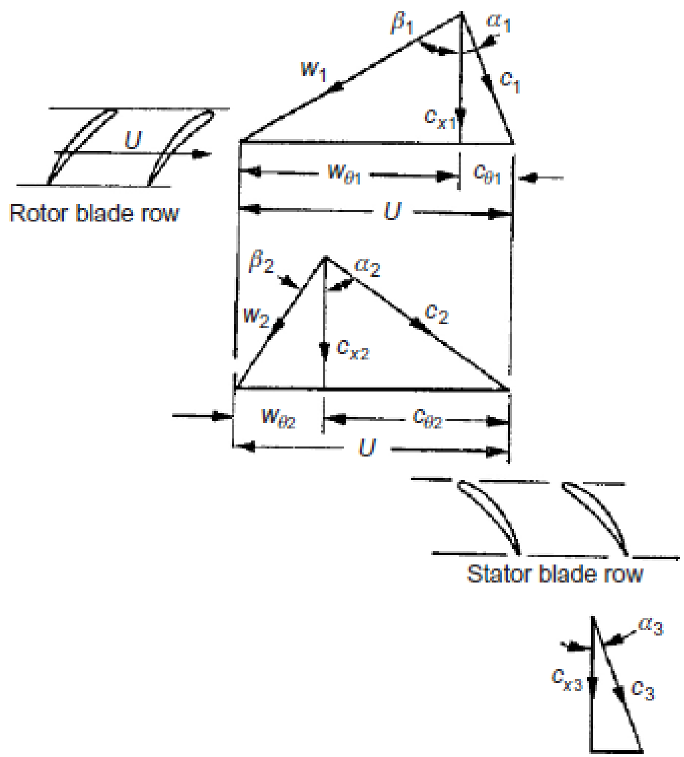
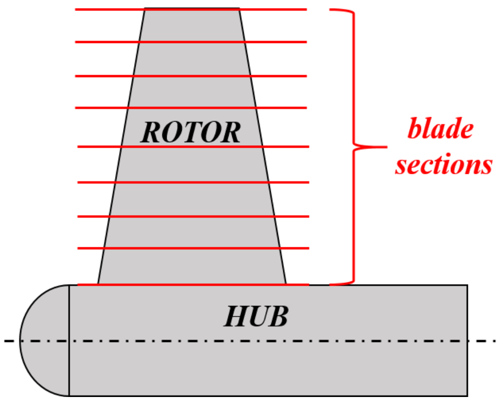
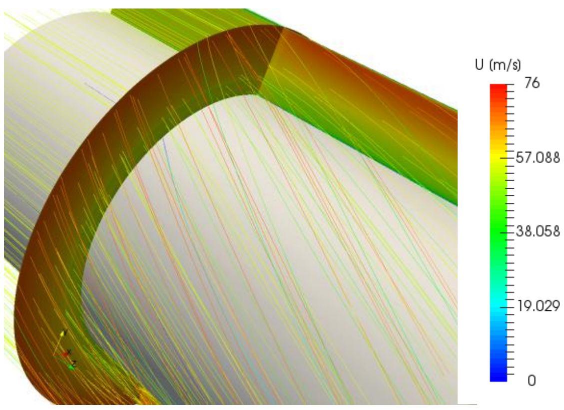
You were also told that coupling cascade theory with radial equilibrium theory you can achieve an axy-symmetryc non viscous solution much more accurate







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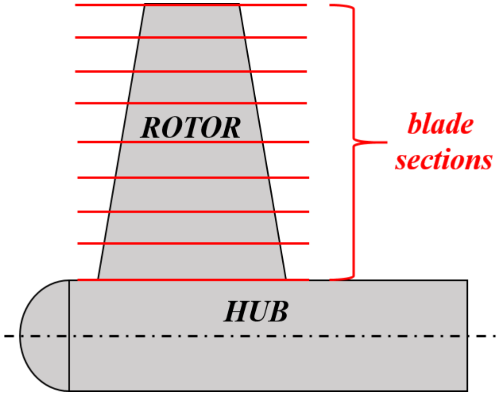
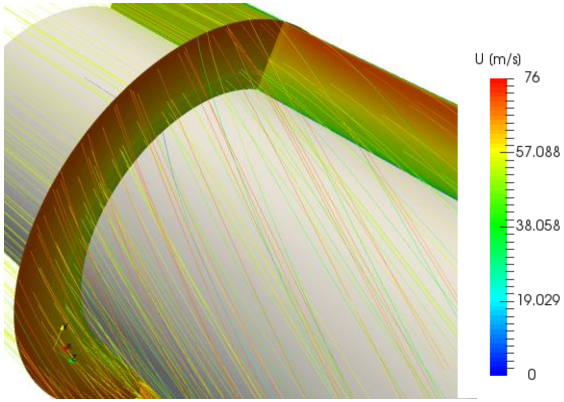
For this we need to couple Euler eqns with radial equilibrium theory:



Vorticity in annulus: $\omega = \frac{1}{r}\frac{d(rC_\theta)}{dr}$

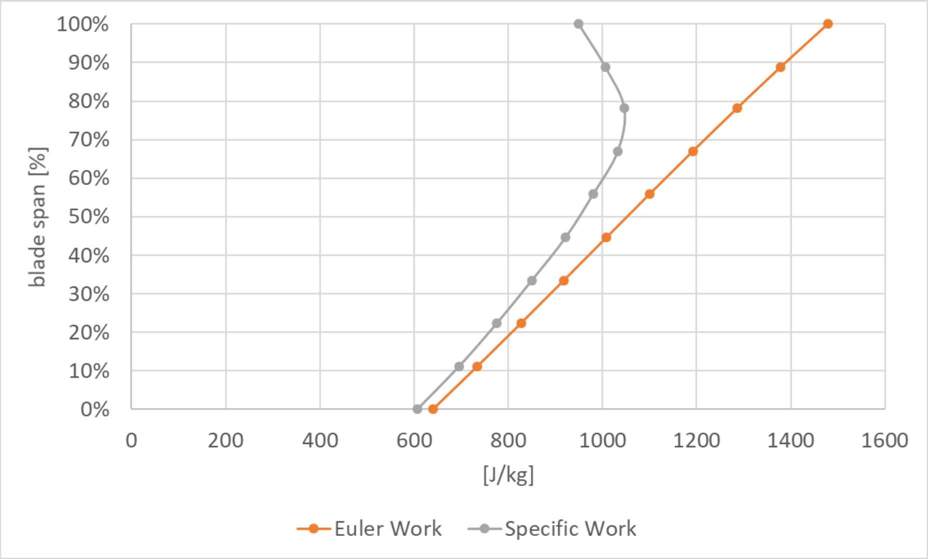
Free vortex: $\Delta W = U\Delta C_\theta = cost$

Forced vortex: different definitions (linear, exponential, mixed…)



You were also told that coupling cascade theory with radial equilibrium theory you can achieve an axy-symmetryc non viscous solution much more accurate. The increase in accuracy comes from two major factors:

1. Axysimmetric solvers are able to compute a proper radial distribution of axial velocity
2. Have a more accurate prediction of exit flow angle that entails viscous cascade effects



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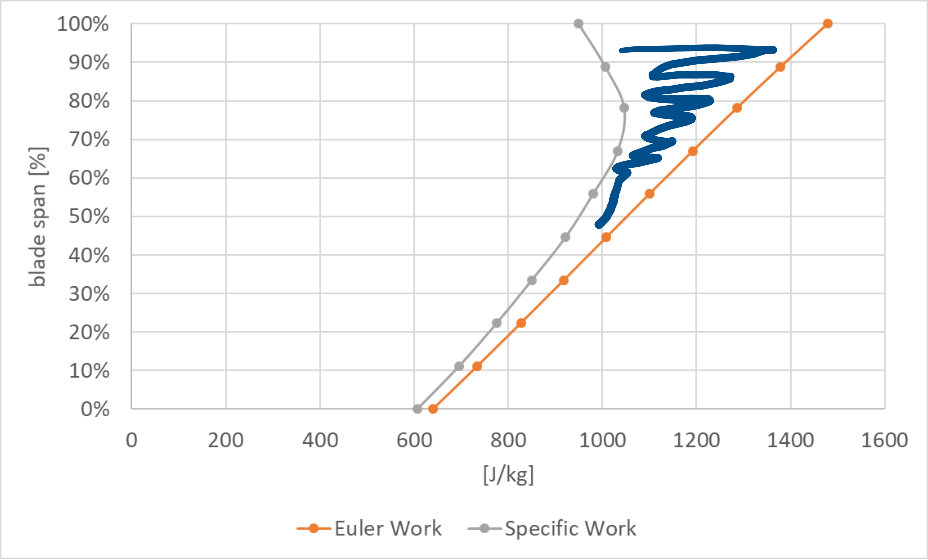

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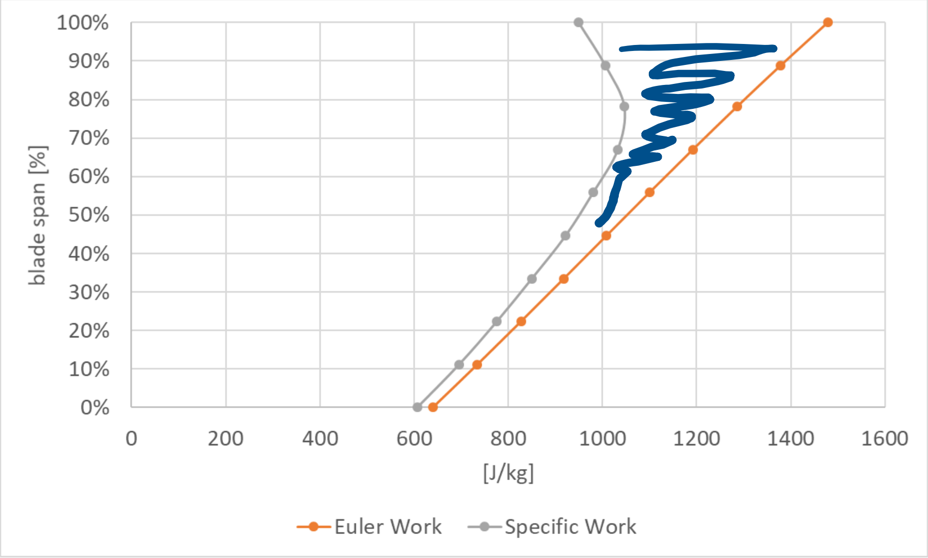
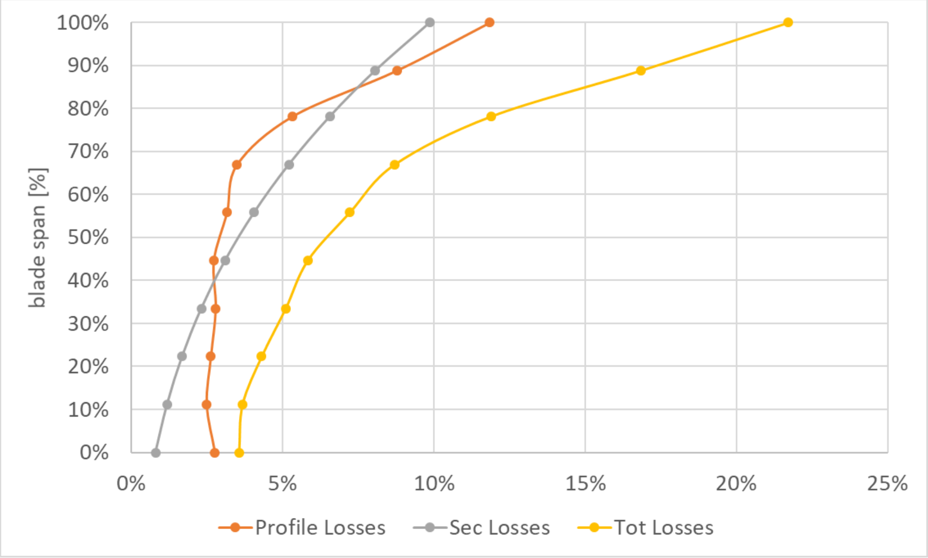
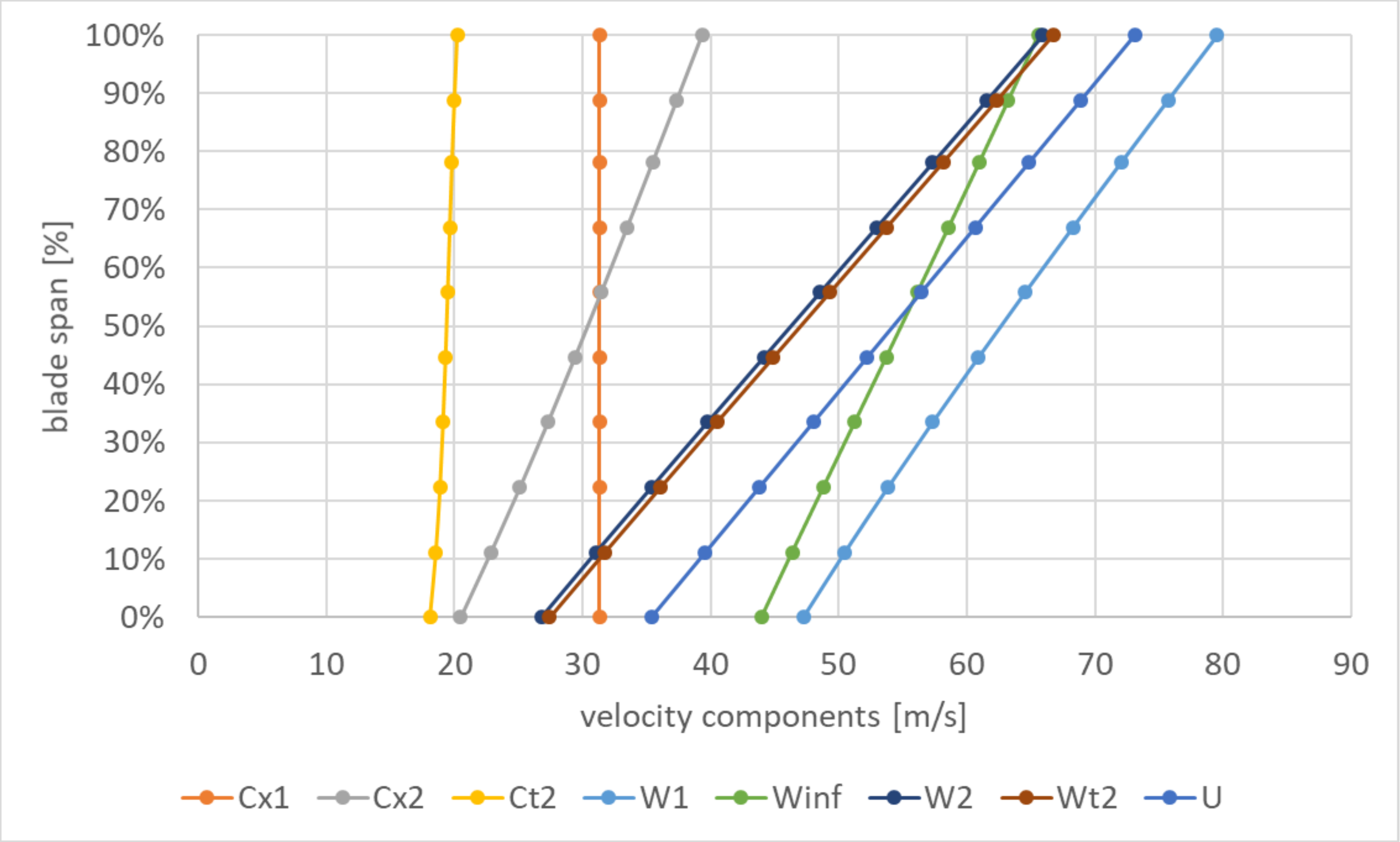




Moreover you can derive radial distribution of a number of quantities, such as velocity components:



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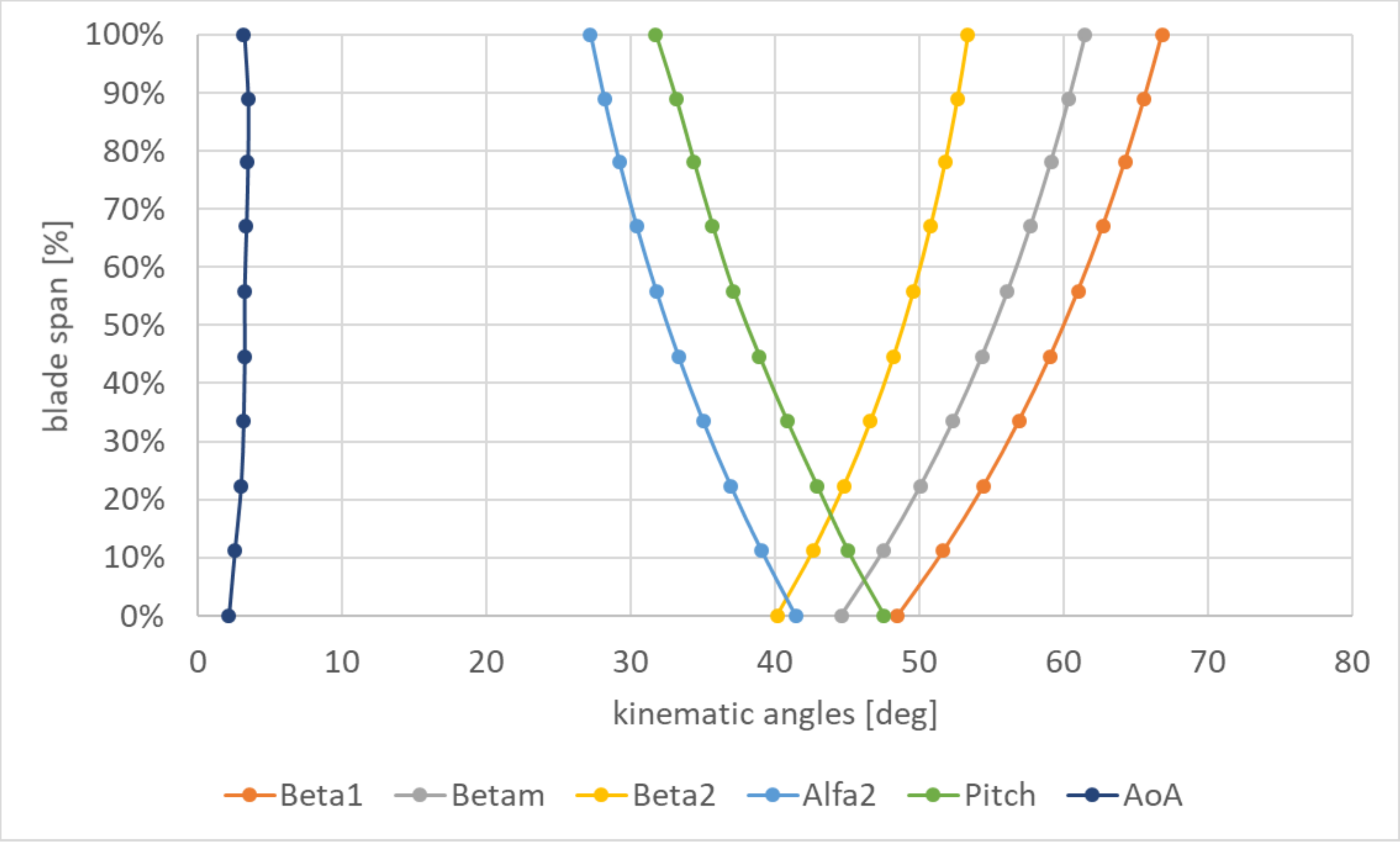
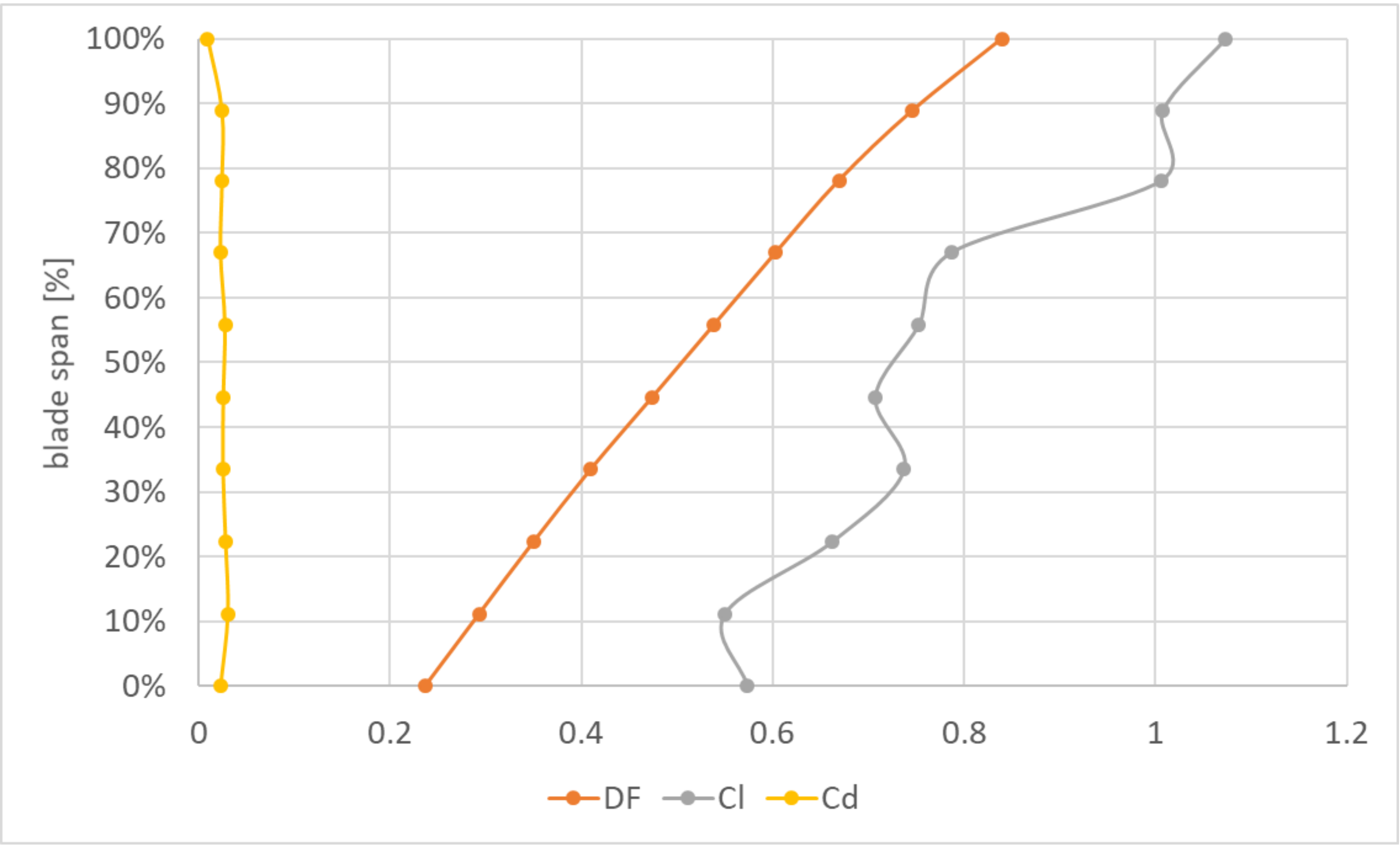
Moreover you can derive radial distribution of a number of quantities, such as kinematic angles:



Moreover you can derive radial distribution of a number of quantities, such as DF and lift and drag along the blade:



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Wake effects in blade cascade

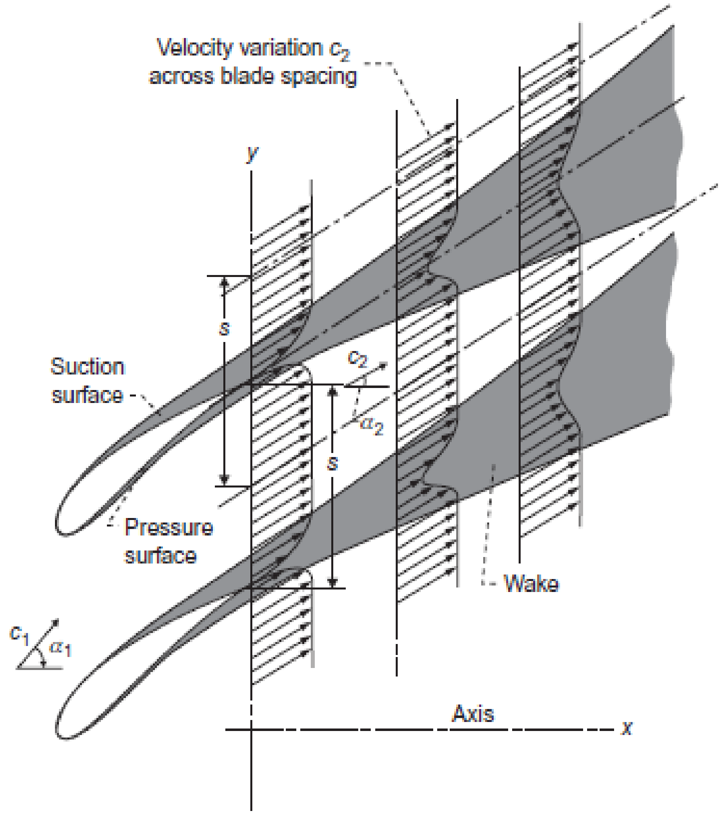

FIGURE 3.7

The flow through a blade cascade and the formation of the wakes.

(From Johnson and Bullock, 1965)

da *Dixon*, 2014

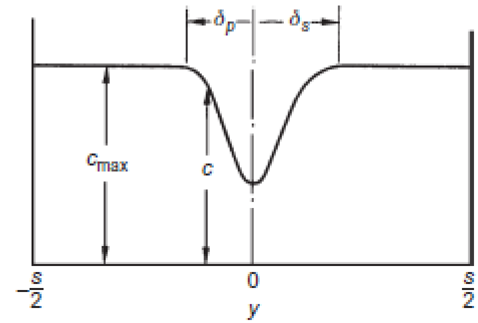

FIGURE 3.15

Blade wake downstream of the exit of a compressor blade cascade.

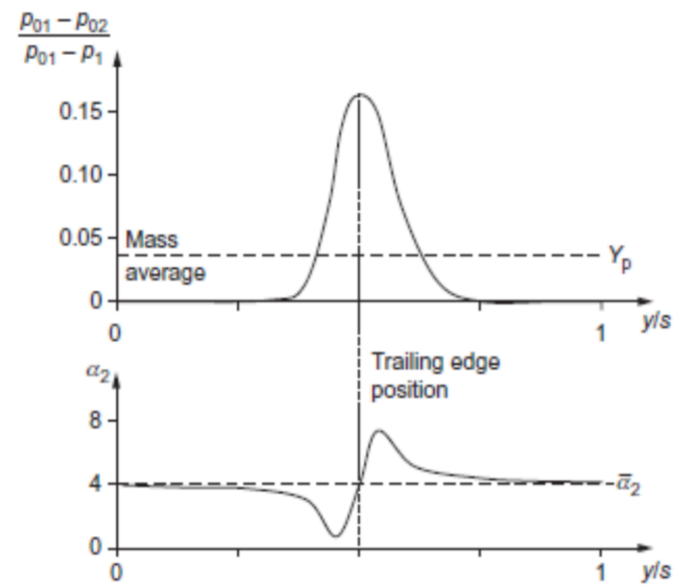

FIGURE 3.9

Typical traverse results for a compressor cascade.

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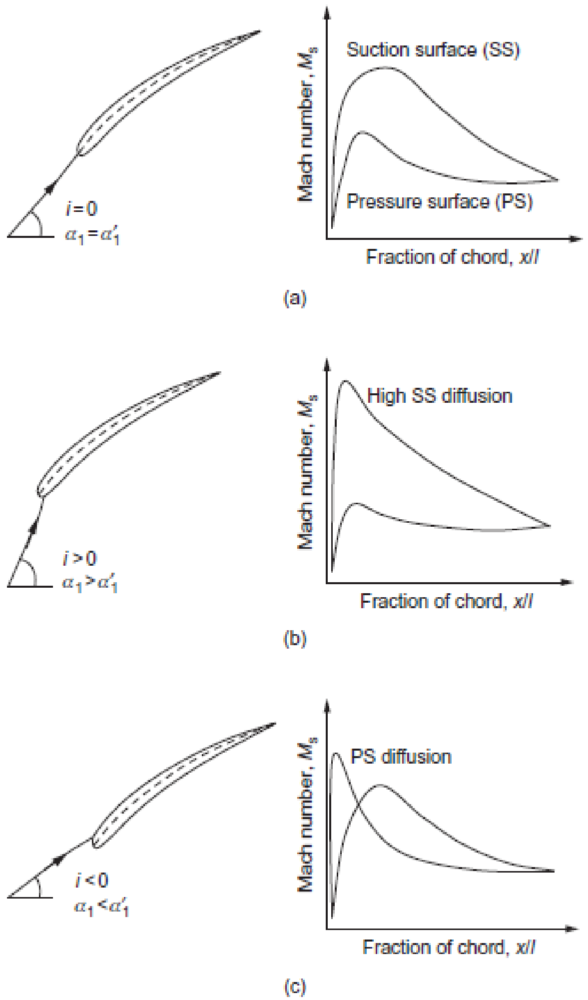


FIGURE 3.14

Effect of incidence on the surface velocity distributions around a compressor blade: (a) design incidence, (b) positive incidence, and (c) negative incidence.

da *Dixon*, 2014

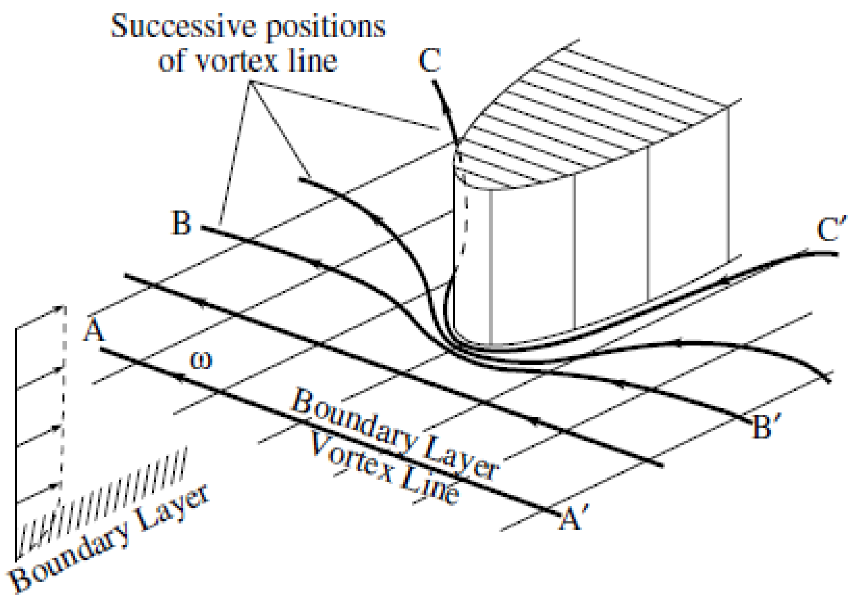
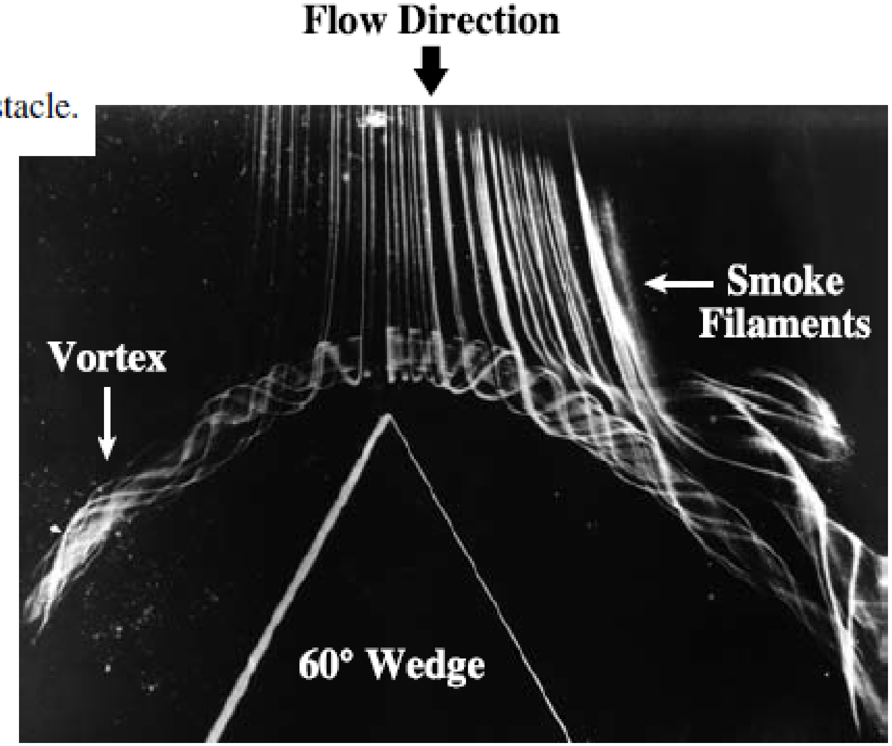


Figure 3.11: Boundary layer vortex lines wrapping round an obstacle.

da *Greitzer*, 2004

Interaction between boundary layer and bluff body with development of horseshoe vortex



Smoke flow: visualization of a horseshoe vortex
upstream of a 60° wedge in a channel (Schwind, 1962)

da *Greitzer*, 2004

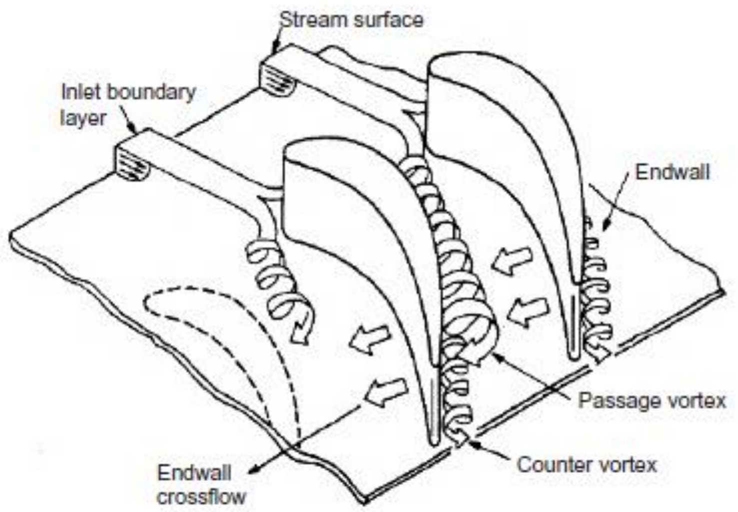


FIGURE 6.19

Secondary flow structure within a blade passage.

(*Adapted from Langston, 1980*)

da *Dixon*, 2014

Secondary motions in a turbine blade passage

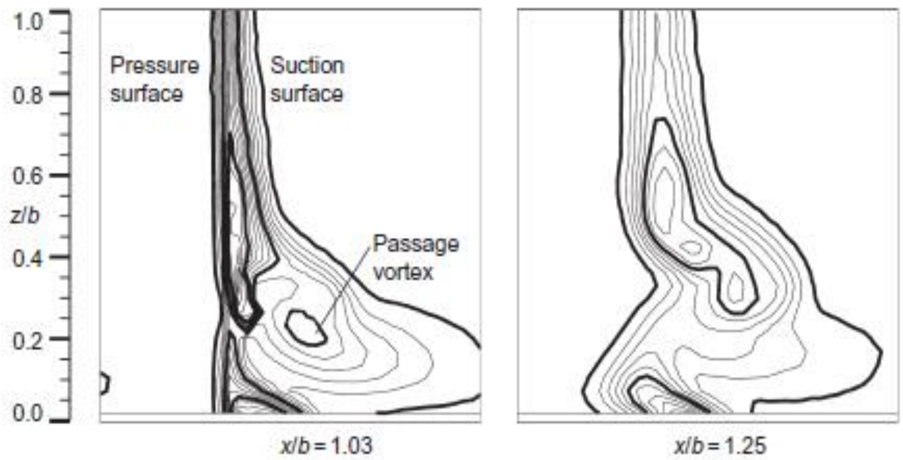


FIGURE 6.22

Stagnation pressure contours measured downstream of a turbine cascade: (a) just downstream of the trailing edge and (b) a quarter-chord downstream of the trailing edge.

(Adapted from Pullan & Harvey, 2008)

da *Dixon*, 2014

Passage vortex along the suction surface

Corner separation

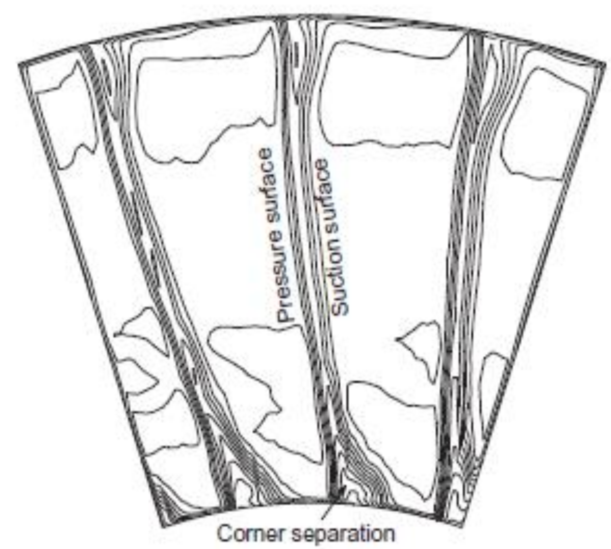


FIGURE 6.21

Stagnation pressure contours measured downstream of a rig compressor stage.

da *Dixon*, 2014

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Tip leakage vortex and induced endwall vortex in axial compressor

Tip leakage vortex in axial flow compressor

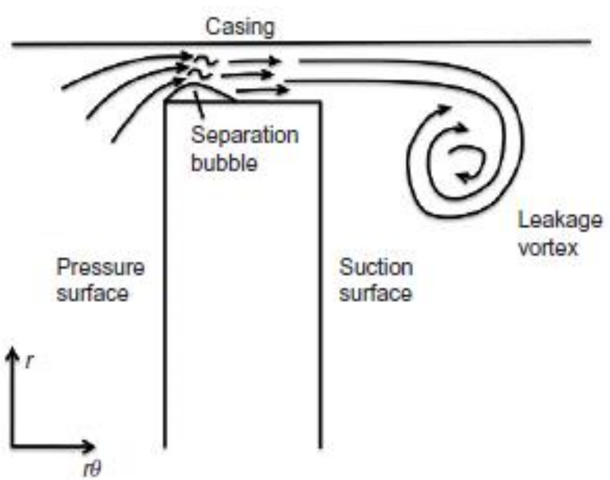
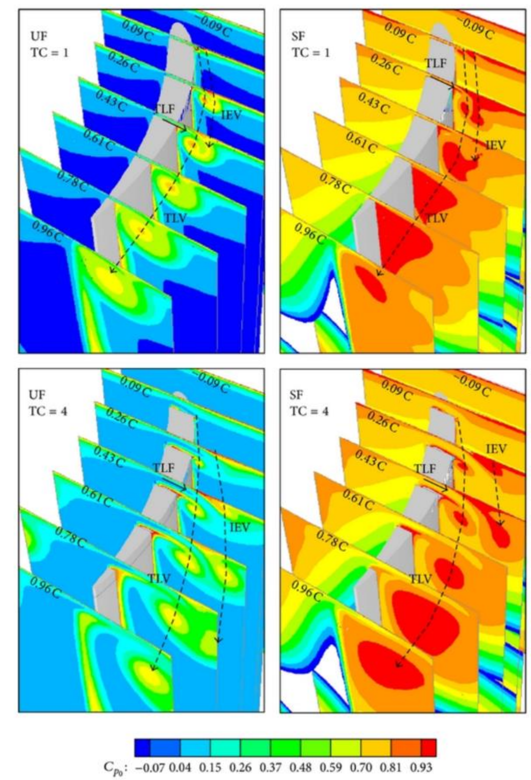


FIGURE 6.23

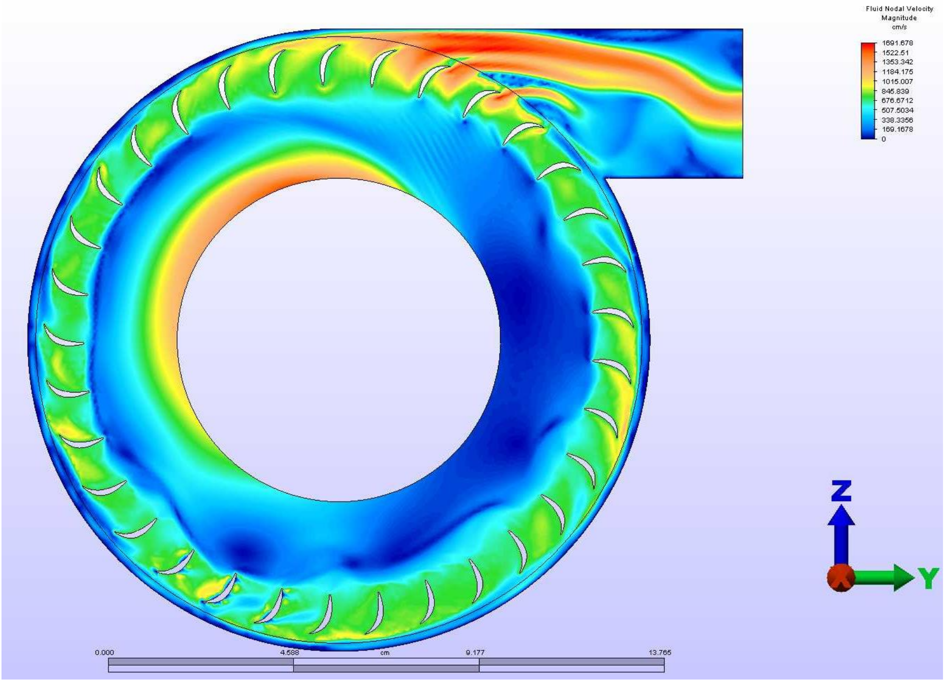
Schematic of tip leakage flow.

Dixon, 2014

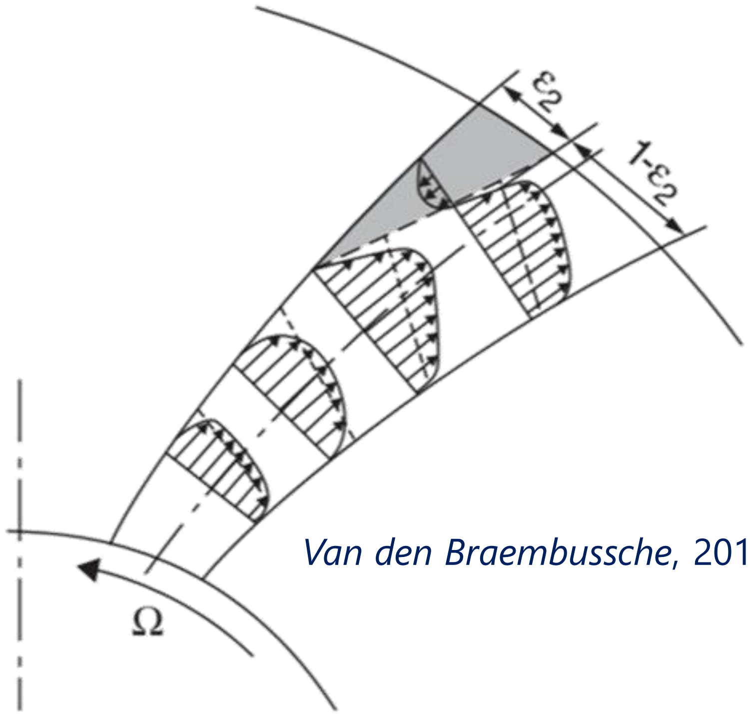


Varpe and *Pradeep*, 2013

Separation in centrifugal fans and compressors



acousticflowsolutions.com



Van den Braembussche, 2019

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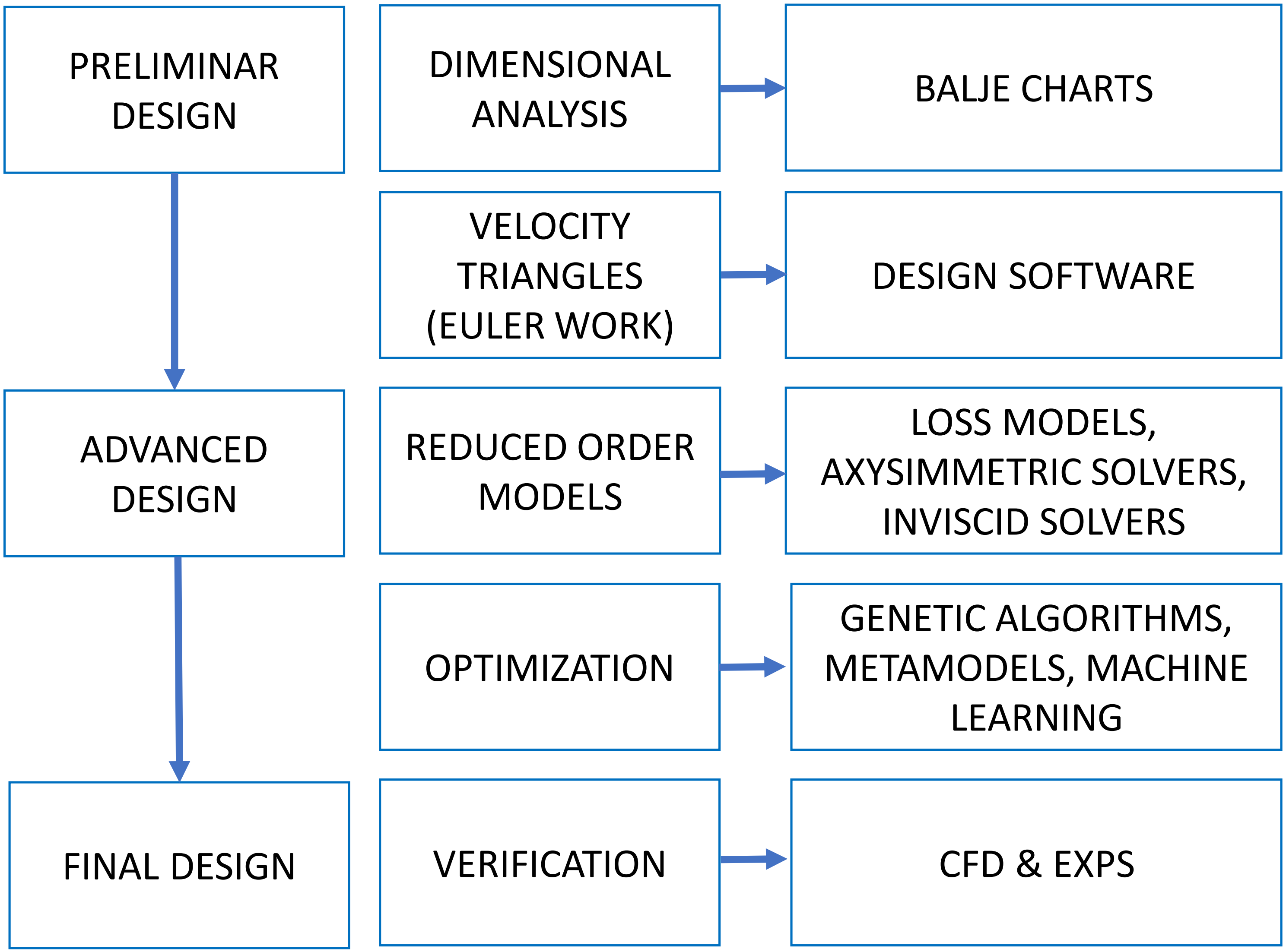
PRELIMINAR DESIGN
DIMENSIONAL ANALYSIS
BALJE CHARTS
VELOCITY TRIANGLES (EULER WORK)
DESIGN SOFTWARE
ADVANCED DESIGN
REDUCED ORDER MODELS
LOSS MODELS, AXYSIMMETRIC SOLVERS, INVISCID SOLVERS
OPTIMIZATION
GENETIC ALGORITHMS, METAMODELS, MACHINE LEARNING
FINAL DESIGN
VERIFICATION
CFD & EXPS

Dimensional analysis

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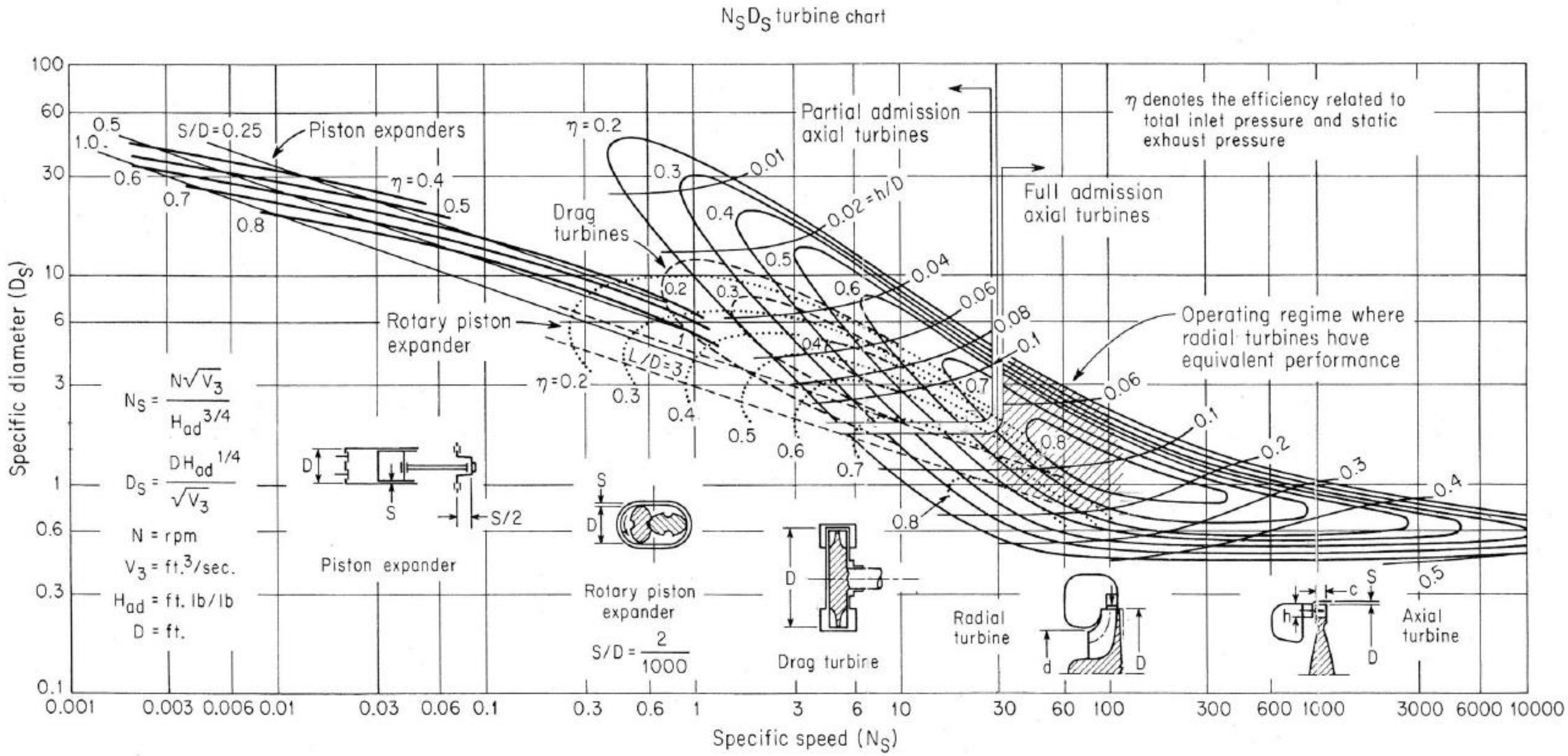
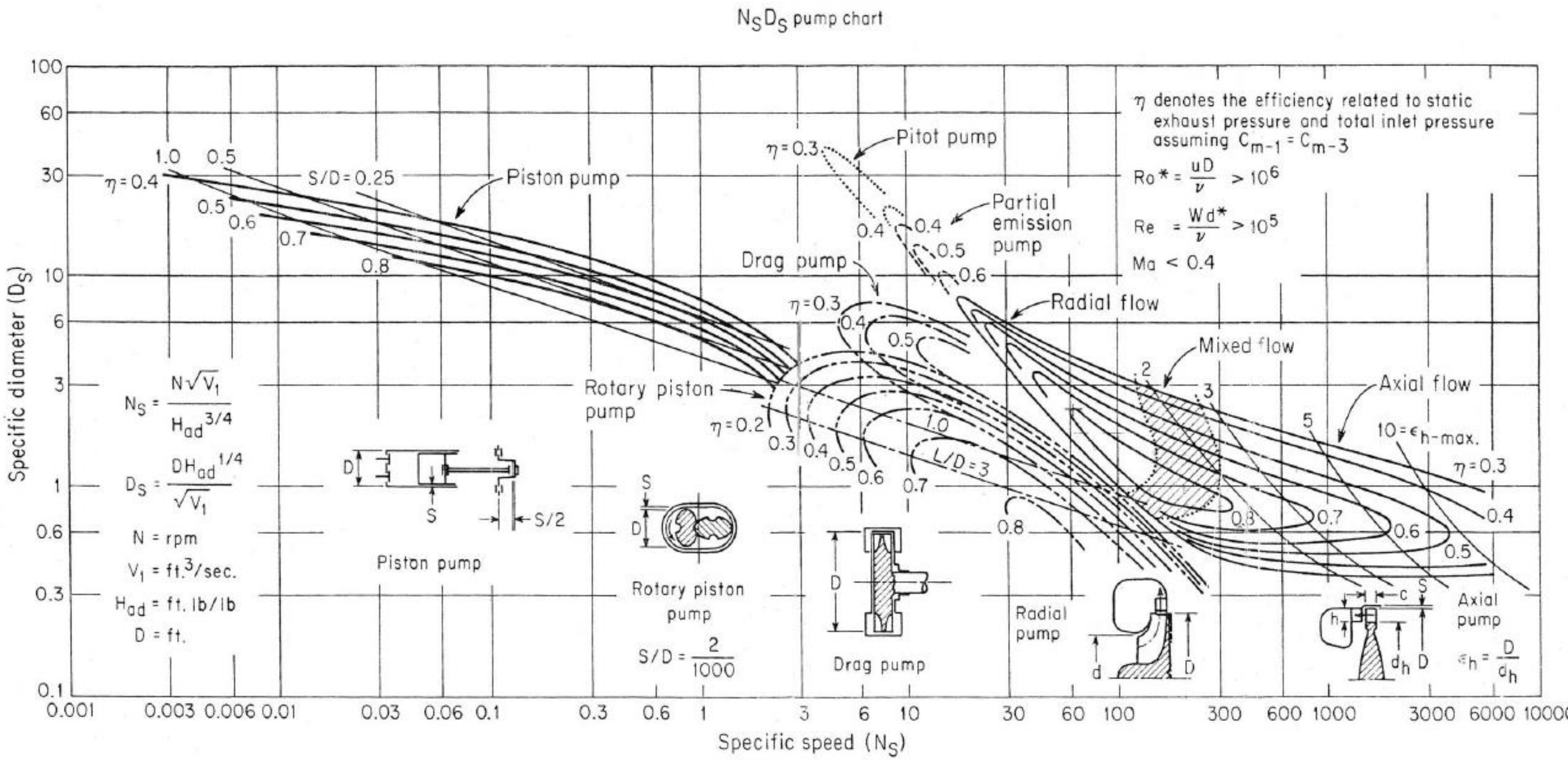
Balje charts (turbines)



Balje, 1981



Balje, 1981

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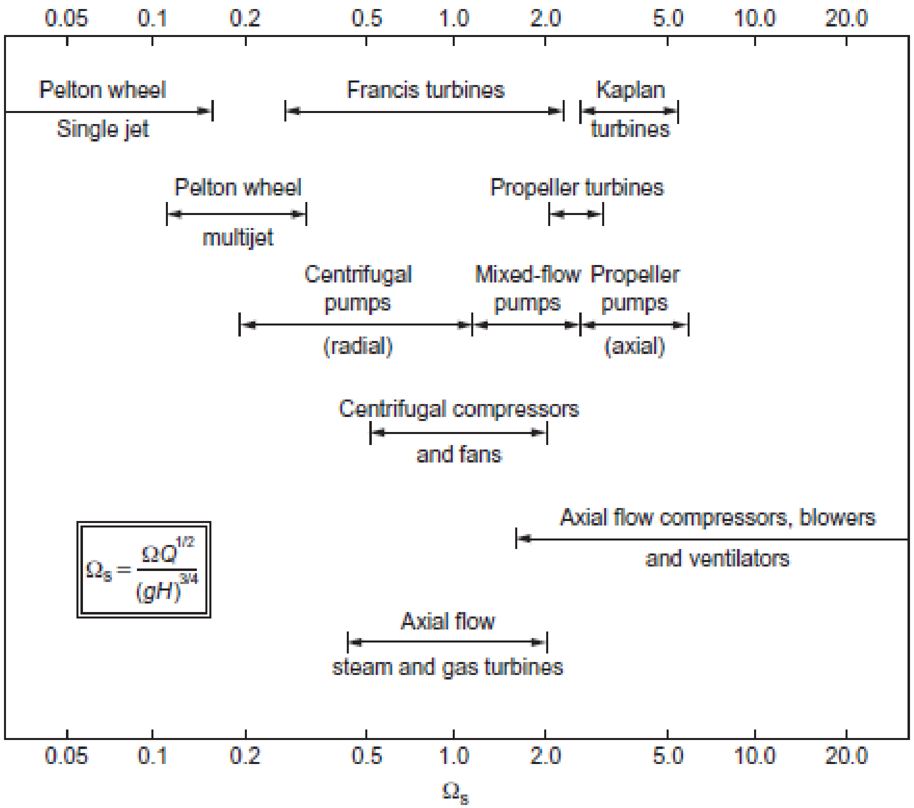


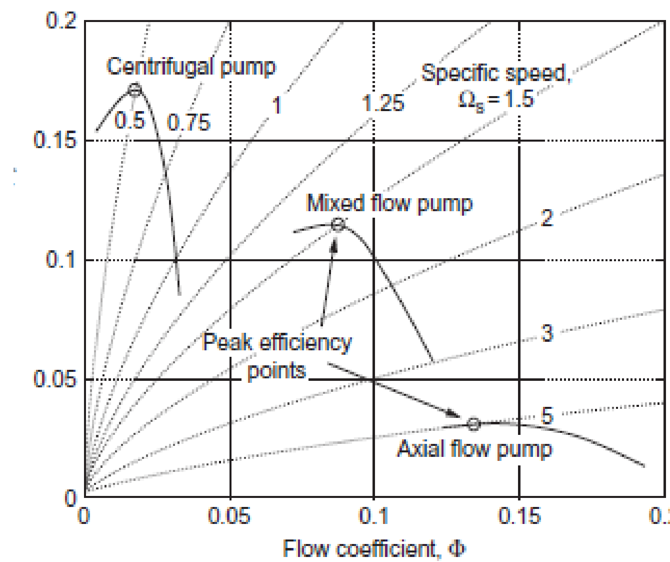


FIGURE 2.8

Range of specific speeds for various types of turbomachine.

Dixon, 2014

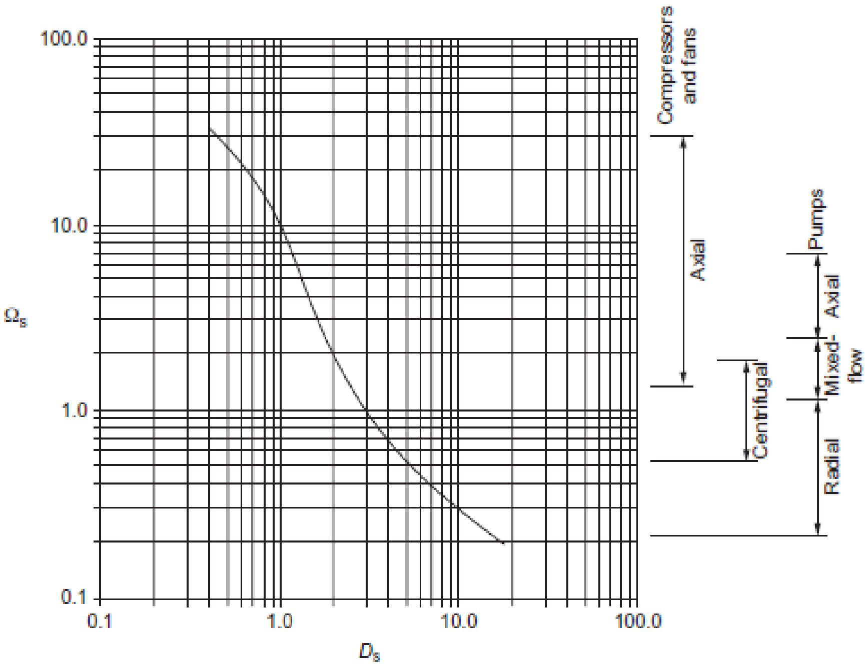


FIGURE 2.9

Cordier diagram for machine selection.

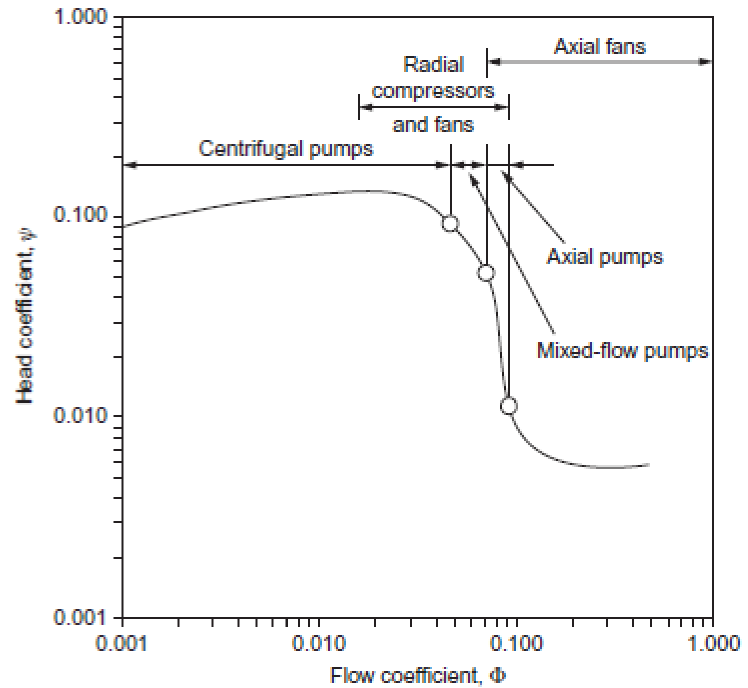


FIGURE 2.10

Chart of ψ versus Φ for various pumps and fans.

Dixon, 2014

To study the performance characteristics of turbomachinery, **a large number of parameters** is involved. Dimensional analysis reduces the variables to a number of manageable **dimensional groups**.

Usually, the properties of interest in turbomachinery are **power output**, **efficiency**, and **head**. The performance of turbomachinery depends on one or more of several parameters.

Dimensional analysis applied to turbomachinery has two uses:

1. **prediction of a prototype's performance** from tests conducted on a scale model (**similitude**)

2. **determination of the most suitable type of machine**, on the basis of maximum efficiency, for a specified range of head, speed, and flow rate.

The variables involved in engineering are expressed in terms of a limited number of basic dimensions.

For most engineering problems, the basic dimensions are **mass (m)**, **length (L)**, **temperature (T)** and **time (t)**

For example, the dimensions of pressure can be designated as follows:

$$P = \frac{force}{area} = \frac{mass \ \cdot acceleration}{area} = \left[\frac{mL/t^2}{L^2}\right] = \left[\frac{m}{Lt^2}\right]$$

Buckingham proved that **the number of** independent dimensionless group of variables **(dimensionless parameters) needed to correlate the unknown variables in a given process is equal to n - m**, where n is the number of variables involved and m is the number of dimensionless parameters included in the variables.

Suppose, for example, the drag force F of a flowing fluid past a sphere is known to be a function of the velocity (v) mass density (ρ) viscosity (m) and diameter (D).

We have five variables (F, v, ρ, m, and D) and three basic dimensions (L, F, and T) involved. Then, there are 5 - 3 = 2 basic grouping of variables that can be used to correlate experimental results.

1. Understand what typology of TM is the most suitable (i.e. has maximum efficiency) for a given range of Ω [rad/s], Q [m^3/s] and H [m]

2. Predict performance from a scaled prototype tested in laboratory (with some caveats)

1. You can define only two control parameters (e.g. Ω [rad/s] and Q [m^3/s] for a pump)
2. You choose a working fluid (ρ [kg/m^3] and μ [Pa s])
3. You select a size of the pump (D [m])

$$gH = f_1\left(Q, \Omega, D, \varrho, \mu, e, \frac{l_1}{D}, \frac{l_2}{D}, \dots\right)$$
$$\eta = f_2\left(Q, \Omega, D, \varrho, \mu, e, \frac{l_1}{D}, \frac{l_2}{D}, \dots\right)$$
$$P = f_3\left(Q, \Omega, D, \varrho, \mu, e, \frac{l_1}{D}, \frac{l_2}{D}, \dots\right)$$

Buckingham theorem:

6 variables $(Q, \Omega, D, \varrho, \mu, e\ )$ → 6-3=3 non-dimensional groups

3 dimensions (mass, length, time)

Head coefficient: $\psi = \frac{gH}{(\Omega D)^2} = f_4\left(\frac{Q}{\Omega D^3}, \frac{\rho \Omega D^2}{\mu}, \frac{e}{D}\right)$

Efficiency: $\eta = f_5\left(\frac{Q}{\Omega D^3}, \frac{\rho \Omega D^2}{\mu}, \frac{e}{D}\right)$

Power coefficient: $\Pi = \frac{P}{\varrho \Omega^3 D^5} = f_6\left(\frac{Q}{\Omega D^3}, \frac{\rho \Omega D^2}{\mu}, \frac{e}{D}\right)$

Flow coefficient: $\varphi = \frac{Q}{\Omega D^3} \propto \frac{c_m}{U}$

Reynolds Number: $Re = \frac{\rho \Omega D^2}{\mu}$

Normalized roughness: $\frac{e}{D}$

All these coefficients are dimensionless

BUT

MANY US/UK CHARTS ARE NOT

BEWARE EVERY TIME TO CHECK

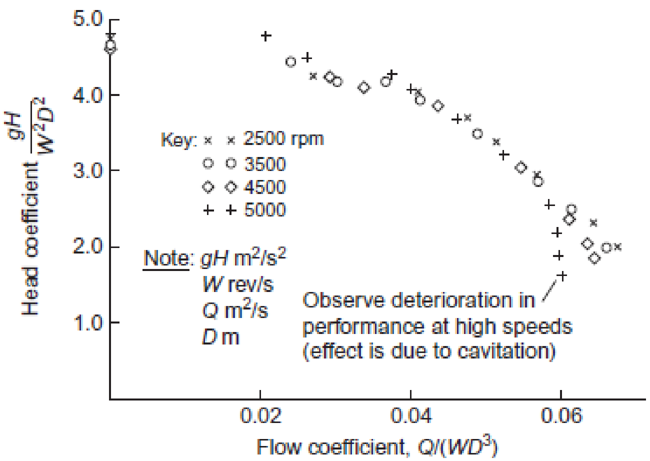


FIGURE 2.2

Dimensionless head–volume characteristic of a centrifugal pump.

da *Dixon*, 2014

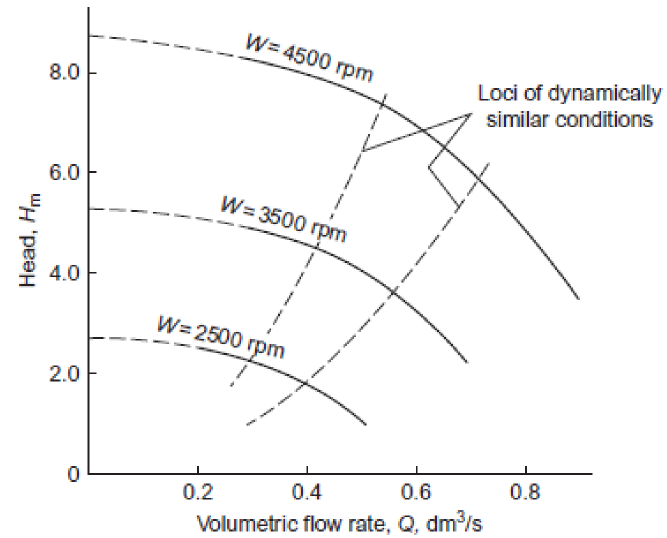


FIGURE 2.3

Extrapolation of characteristic curves for dynamically similar conditions at $\Omega = 3500$ rpm.

da *Dixon*, 2014

$$\varphi_1 = \frac{Q}{\Omega D^3} = cost$$
$$\psi_1 = \frac{gH}{(\Omega D)^2} = cost$$
$$\Pi_1 = \frac{gH}{(\Omega D)^2} = cost$$

cancel D → $\Omega_s = \frac{\varphi_1^{1/2}}{\psi_1^{3/4}} = \frac{\Omega Q^{1/2}}{(gH)^{3/4}}$

cancel Ω → $D_s = \frac{\psi_1^{1/4}}{\varphi_1^{1/2}} = \frac{D(gH)^{1/4}}{Q^{1/2}}$

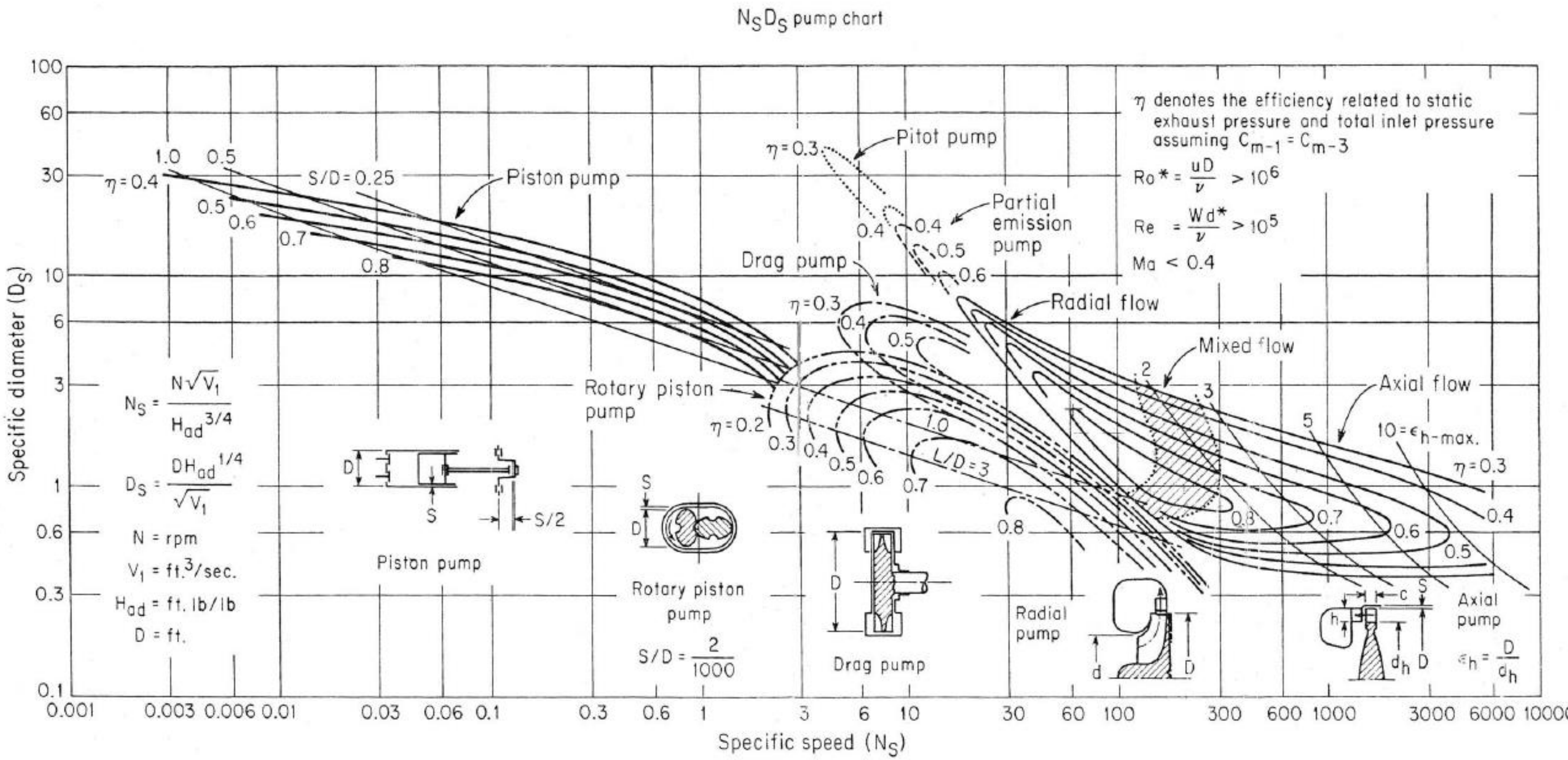
For hydraulic turbines we use power specific speed: $\Omega_{sp} = \frac{\Pi_1^{1/2}}{\psi_1^{5/4}} = \frac{\Omega(P/\varrho)^{1/2}}{(gH)^{5/4}}$

Notice that: $\frac{\Omega_{sp}}{\Omega_s} = \sqrt{\eta}$

and that: Ω_s, Ω_{sp} and D_s are dimensionless, but in some books and graphs they aren't

In different books and manuals flow, head and power coefficients can have different definitions so be advised

Balje charts (pumps, incompressible flows)



Balje, 1981

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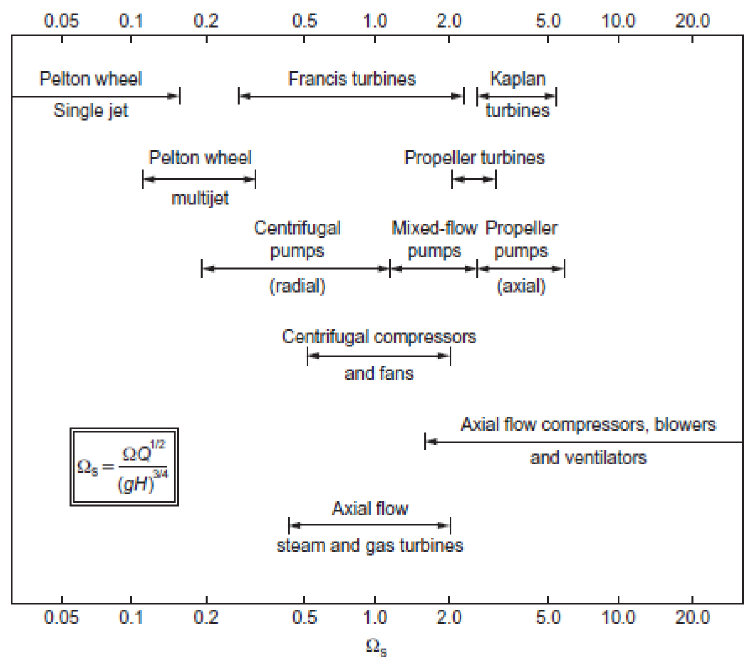


FIGURE 2.8

Range of specific speeds for various types of turbomachine.

(*From Csanady, 1964*)

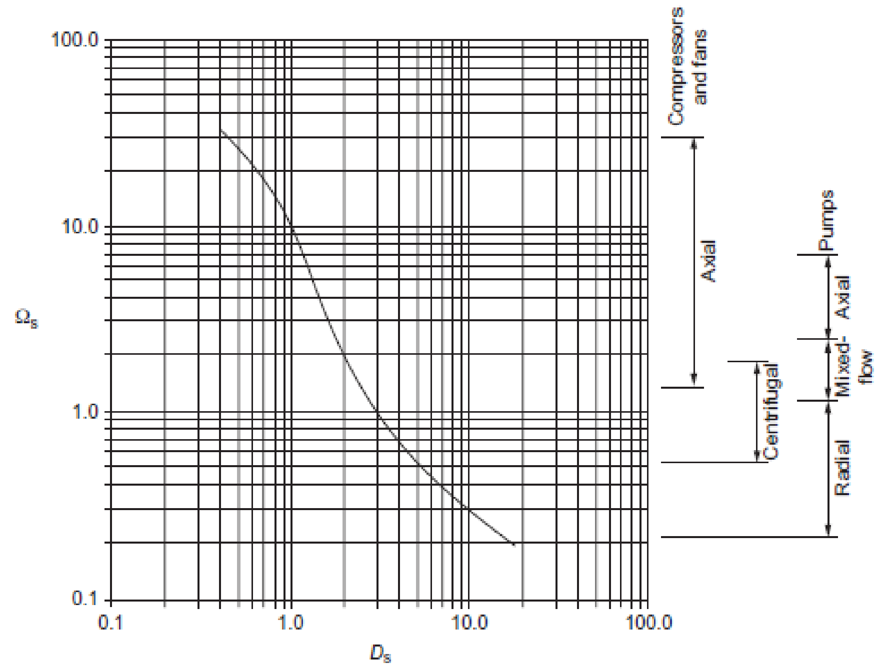


FIGURE 2.9

Cordier diagram for machine selection.

Dixon, 2014

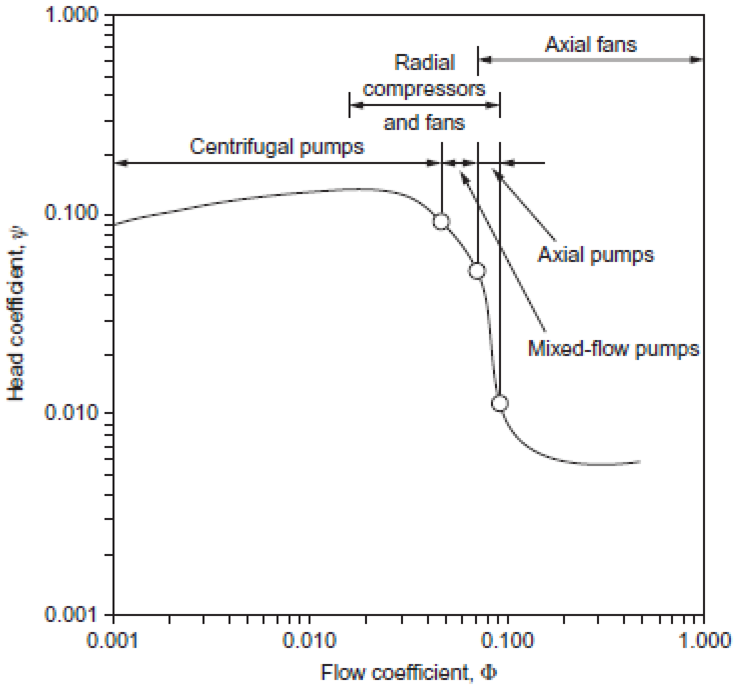
$$\Omega_s = \frac{\varphi_1^{1/2}}{\psi_1^{3/4}} = \frac{\Omega Q^{1/2}}{(gH)^{3/4}}$$

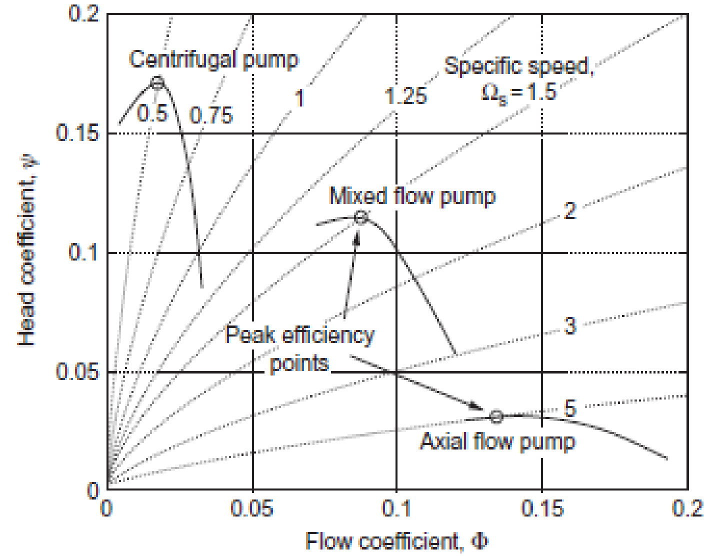
$$D_s = \frac{\psi_1^{1/4}}{\varphi_1^{1/2}} = \frac{D(gH)^{1/4}}{Q^{1/2}}$$

$$\longrightarrow$$

$$\varphi = \frac{1}{\Omega_s \, D_s^3}$$

$$\psi = \frac{1}{\Omega_s^2 D_s^2}$$





da *Dixon*, 2014

1. You now need to define <u>four</u> control parameters (e.g. Ω rad/s], $\dot{m}$ [kg/s], a_{01}[-] and γ [–]), where a_{01} is the stagnation speed of sound and γ the ratio of specific heats
2. You choose a working fluid (ρ [kg/m^3] and μ [Pa s])
3. You select a size of the pump (D [m])
4. Instead of *H* we use stagnation enthalpy Δh_{0s} [J] change as in an ideal adiabatic process this is equal to the work done per unit mass of fluid

$$\Delta h_{0s} = f_1(\mu, \Omega, D, \rho_{01}, a_{01}, \gamma)$$

$$\eta = f_2(\mu, \Omega, D, \rho_{01}, a_{01}, \gamma)$$

$$P = f_3(\mu, \Omega, D, \rho_{01}, a_{01}, \gamma)$$

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Head coefficient: $\psi = \frac{\Delta h_{0s}}{(\Omega D)^2} = f_4\left(\frac{\dot{m}}{\rho_{01}\Omega D^3}, \frac{\rho_{01}\Omega D^2}{\mu}, \frac{\Omega D}{a_{01}}, \gamma\right)$

Efficiency: $\eta = f_5\left(\frac{\dot{m}}{\rho_{01}\Omega D^3}, \frac{\rho_{01}\Omega D^2}{\mu}, \frac{\Omega D}{a_{01}}, \gamma\right)$

Power coefficient: $\Pi = \frac{P}{\rho_{01}\Omega^3 D^5} = f_6\left(\frac{\dot{m}}{\rho_{01}\Omega D^3}, \frac{\rho_{01}\Omega D^2}{\mu}, \frac{\Omega D}{a_{01}}, \gamma\right)$

Nondimensional mass flow: $\frac{\dot{m}}{\rho_{01}\Omega D^3}$

Reynolds Number: $Re = \frac{\rho_{01}\Omega D^2}{\mu}$

Blade Mach Number: $\mathrm{Ma} = \frac{\Omega D}{a_{01}}$

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Head coefficient: $\psi = \frac{\Delta h_{0s}}{a_{01}^2} = f_4\left(\frac{\dot{m}}{\rho_{01} a_{01} D^2}, \frac{\rho_{01} a_{01} D}{\mu}, \frac{\Omega D}{a_{01}}, \gamma\right)$

Efficiency: $\eta = f_5\left(\frac{\dot{m}}{\rho_{01} a_{01} D^2}, \frac{\rho_{01} a_{01} D}{\mu}, \frac{\Omega D}{a_{01}}, \gamma\right)$

Power coefficient: $\Pi = \frac{P}{\rho_{01} a_{01}^3 D^2} = f_6\left(\frac{\dot{m}}{\rho_{01} a_{01} D^2}, \frac{\rho_{01} a_{01} D}{\mu}, \frac{\Omega D}{a_{01}}, \gamma\right)$

Nondimensional mass flow: $\frac{\dot{m}}{\rho_{01} a_{01} D^2}$

Reynolds Number: $Re = \frac{\rho_{01} a_{01} D}{\mu}$

Blade Mach Number: $\mathrm{Ma} = \frac{\Omega D}{a_{01}}$

For a perfect gas we can use more convenient relationships:

$$\frac{p_{02}}{p_{01}}, \eta, \frac{\Delta T_0}{T_{01}} = f\left(\frac{\dot{m}\sqrt{\gamma R T_{01}}}{p_{01} D^2}, \frac{\Omega D}{\sqrt{\gamma R T_{01}}}, Re, \gamma\right)$$

Why more convenient? Because it is easy to measure inlet and exit stagnation quantities in a TM

Moreover if the device handles a single gas and operates at high Re in a narrow range:

$$\frac{p_{02}}{p_{01}}, \eta, \frac{\Delta T_0}{T_{01}} = f\left(\frac{\dot{m}\sqrt{\gamma R T_{01}}}{p_{01} D^2}, \frac{\Omega D}{\sqrt{\gamma R T_{01}}}\right)$$

Flow capacity: $\frac{\dot{m}\sqrt{\gamma R T_{01}}}{p_{01} D^2}$

In industry, for TM of known size and working fluid it is customary to omit γ, *R*, C_p, *D* (but ratios on RHS are not dimensionless anymore)

$$\frac{p_{02}}{p_{01}}, \eta, \frac{\Delta T_0}{T_{01}} = f\left(\frac{\dot{m}\sqrt{T_{01}}}{p_{01}}, \frac{\Omega}{\sqrt{T_{01}}}\right)$$

And to refer them to standard sea level conditions using corrected speed and corrected flow:

$$\frac{p_{02}}{p_{01}}, \eta, \frac{\Delta T_0}{T_{01}} = f\left(\frac{\dot{m}\sqrt{\theta}}{\delta}, \frac{\Omega}{\sqrt{T_{01}}}\right)$$

with:

$$\theta = \frac{T_{01}}{T_a} \text{ and } \delta = \frac{p_{01}}{p_a}$$

Efficiency can be rewritten for compressors and turbines as:

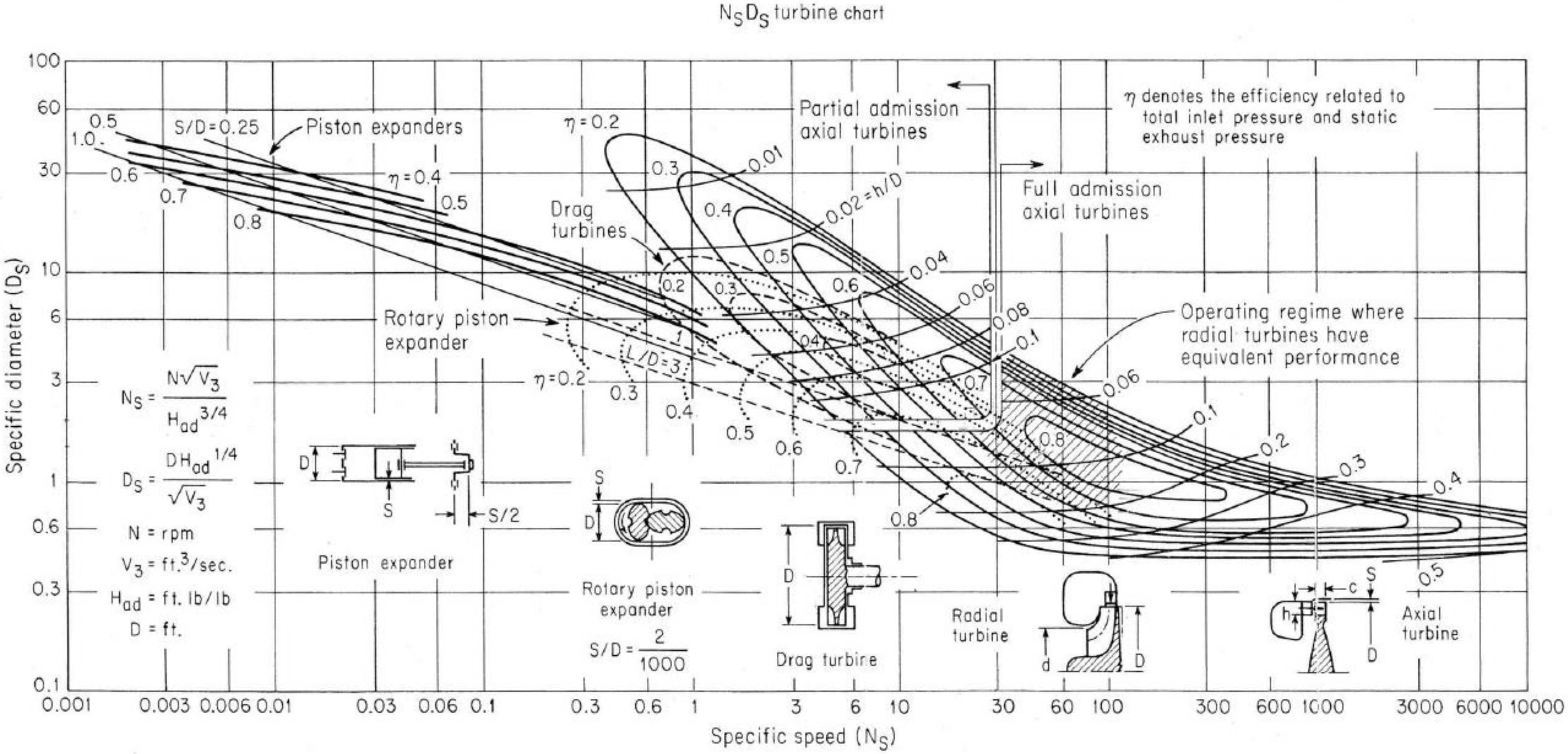
$$\eta_C = \frac{\Delta h_{0s}}{\Delta h_0} = \frac{\left(p_{02}/p_{01}\right)^{\gamma/\gamma-1} - 1}{\Delta T_0/T_{01}}$$

$$\eta_T = \frac{\Delta h_0}{\Delta h_{0s}} = \frac{\Delta T_0/T_{01}}{\left(p_{02}/p_{01}\right)^{\gamma/\gamma-1} - 1}$$

Flow coefficient (not to be confused with non-dimensional mass flow) and stage loading:

$$\varphi = \frac{\dot{m}}{\rho_{01} A_1 U} = f\left(\frac{\dot{m}\sqrt{c_p T_{01}}}{D^2 p_{01}}, \frac{\Omega D}{\sqrt{\gamma R T_{01}}}\right)$$

$$\psi = \frac{\Delta h_0}{U^2} = f\left(\frac{\dot{m}\sqrt{c_p T_{01}}}{D^2 p_{01}}, \frac{\Omega D}{\sqrt{\gamma R T_{01}}}\right)$$



Balje, 1981

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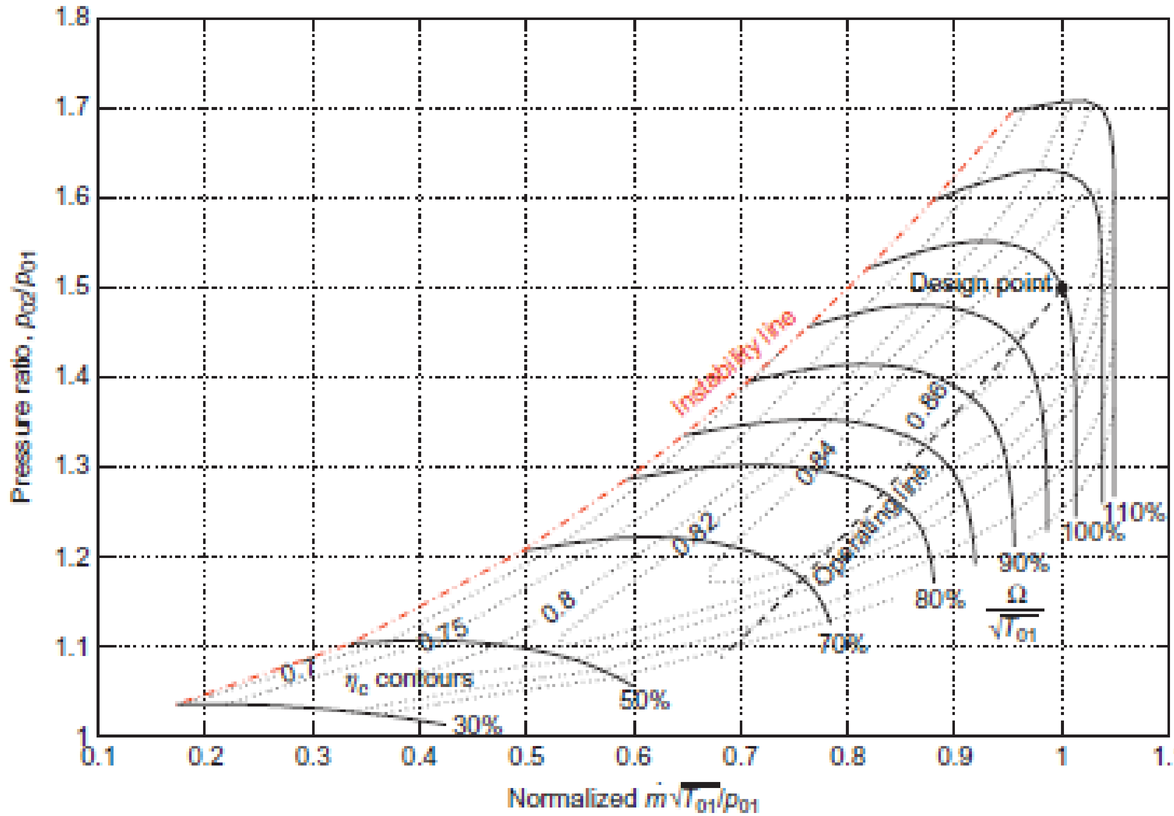

FIGURE 2.4

Characteristic map of a transonic fan for a civil aircraft jet engine.

Dixon, 2014

$$SM = \frac{pr_s - pr_0}{pr_0}$$

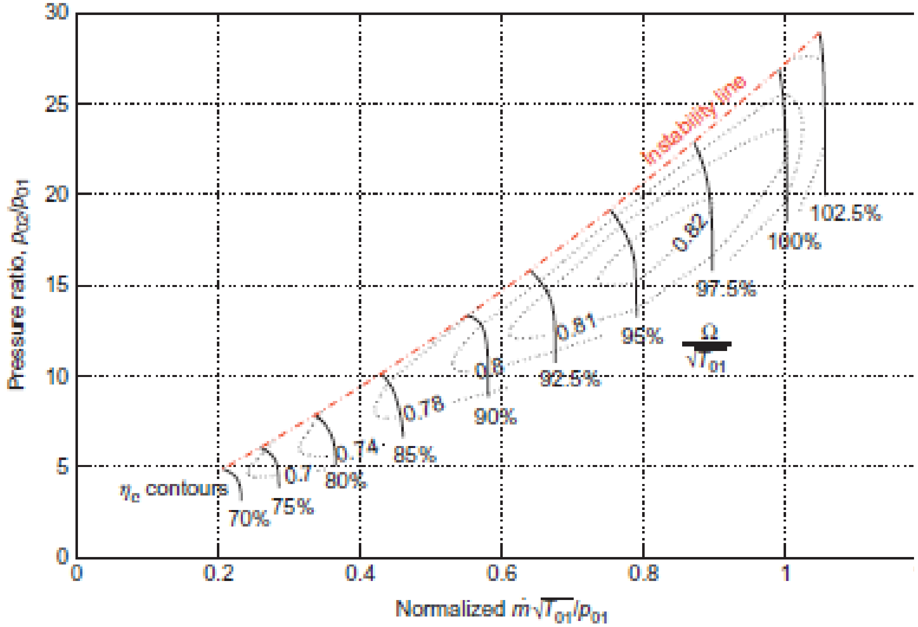

FIGURE 2.5

Performance map of a 10-stage high-speed axial compressor.

(Adapted from Cline

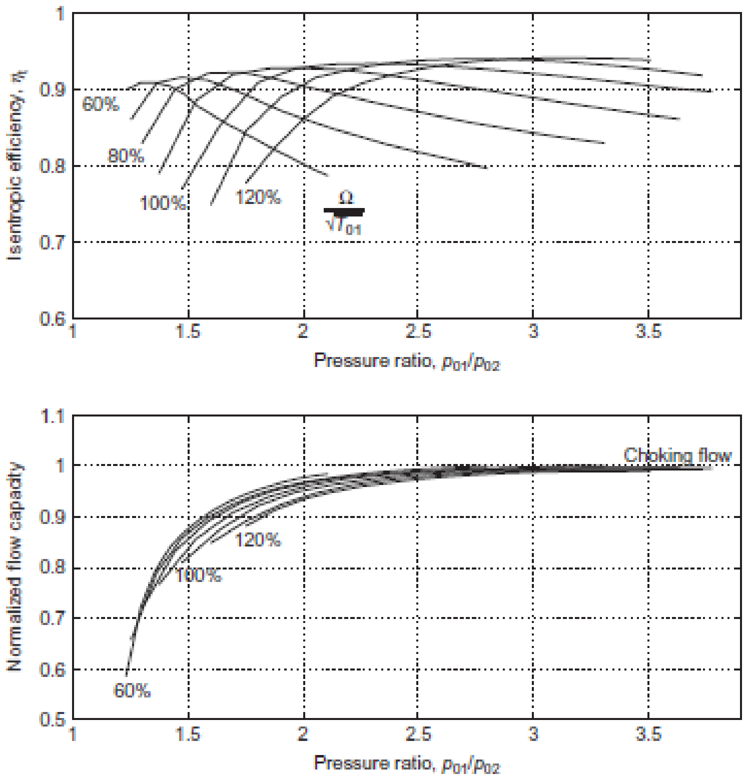

FIGURE 2.6

Overall characteristic of a two-stage high-speed axial turbine.

Dixon, 2014