Turbomachinery for LRE

Pumps

Centrifugal Pumps

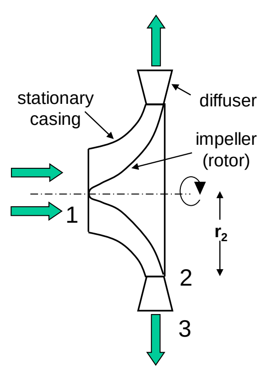
- Commonly employed in rocket engines
  - high flowrate pumps can be axial
  - liquid enters axially, leaves radially
- General Euler relation still holds

$$\dot{W} = \dot{m}\left[(Uc_\theta)_2 - (Uc_\theta)_1\right]$$

  - for no input swirl

$$\dot{W}_P = \dot{m}U_2 c_{\theta_2}$$

impeller tip speed (U_2) liquid tangential velocity at tip (c_{θ_2})



Velocity Change and Pump Work

- Changing ref. frame

$$c_{\theta_2} = U_2 + w_{\theta_2} = U_2 - \overset{= c_{r_2}}{\cancel{w_{r_2}}} \tan\beta_2$$

$$c_{r2} = \frac{\dot{m}}{\rho_2 A_2} \qquad A_2 = 2\pi r_2 b$$

$$\Rightarrow \dot{W}_P = \dot{m} U_2^2 \left(1 - \frac{c_{r2}}{U_2} \tan\beta_2 \right)$$

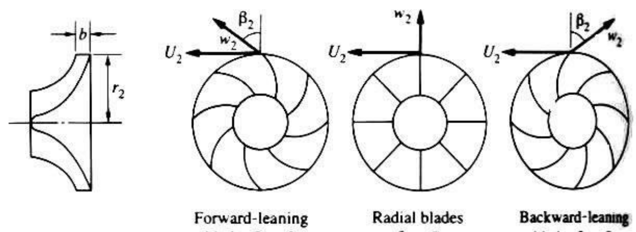
if incomp. liquid, ρ~constant (small T change) $\frac{\dot{m}}{\rho A_2} = \frac{\dot{m}}{\rho A_1}\frac{A_1}{A_2}$

$$\dot{W}_P = \dot{m} U_2^2 \left(1 - \Phi \tan\beta_2 \right)$$

note: LH_2 compressible $\Rightarrow \Phi = \frac{c_{z1}}{U_2}\frac{A_1}{A_2}$

$$\Phi \equiv \frac{c_{r2}}{U_2}$$

flow coeff. for centrifugal machine



From Hill and Peterson

Pump Work and Pressure Rise

- Work also related to enthalpy change

$$\dot{W}_P = \dot{m}\Delta h_{o_{13}} \Rightarrow \Delta h_{o_{13}} = U_2^2 \left(1 - \Phi \tan\beta_2 \right)$$

- From Gibb's eq'n (s state eqn.)

$$T_o ds_o = dh_o - dp_o / \rho$$

if isentropic $dh_o = dp_o / \rho$

if ~incomp. $\Delta h_o \cong \Delta p_o / \rho$

- So if isen. (and incomp.) $\frac{\Delta p_{o_s} / \rho}{U_2^2} = \left(1 - \Phi \tan\beta_2 \right)$

- To include irrev. - use adiabatic efficiency

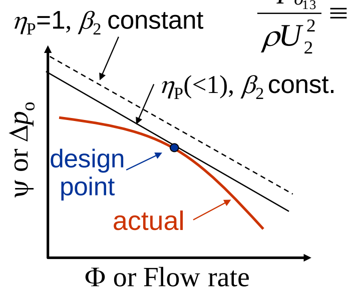
$$\eta_P \equiv \frac{\Delta h_{o_{ideal}}}{\Delta h_o} = \frac{\Delta p_o / \rho}{\Delta h_o} = \frac{\Delta p_o}{\rho \Delta h_o} = \frac{\Delta p_o}{\Delta p_{o_s}}$$

$$\frac{\Delta p_{o_{13}}}{\rho U_2^2} \equiv \psi = \eta_p \left(1 - \Phi \tan\beta_2 \right)$$

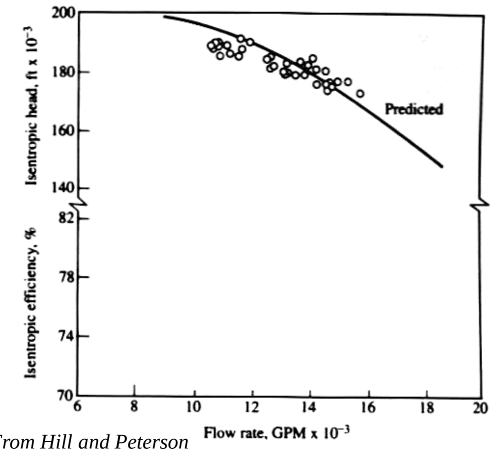
loading coeff.

Pump Characteristics

$$\frac{\Delta p_{o_{13}}}{\rho U_2^2} \equiv \psi = \eta_p \left(1 - \Phi \tan \beta_2\right)$$



- Efficiency losses
  - boundary layers
  - stall: β_2 not set by blade when (ψ,Φ) too far from design (ψ_{design}, Φ_{design})
  - secondary flows…



From Hill and Peterson

Nondimensional Parameters

- For axial turbine, we saw turbine characteristics often shown normalized as
  - flowrate $\dot{m}\sqrt{RT_o}/\left(p_o D^2\right)$ to account for choking effects
  - rot. speed or blade Mach number $\Omega D/\sqrt{\gamma R T_o}$
- Similarly for pumps (not just centrifugal)

$$\Phi = \frac{c_{r2}}{U_2} \propto \frac{\overbrace{Q}^{\text{Vol. Flowrate}}/D^2}{\underbrace{N}_{\text{RPM}}\underbrace{D}_{\text{Diam.}}} = \frac{Q}{ND^3} \equiv \Phi^*$$

can be dimensionless if N→Ω

Typically 0.4-0.7 at design condition

$$\psi = \frac{\Delta p_o/\rho}{U^2} \propto \frac{Hg_o}{(ND)^2} \equiv \psi^*$$

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Specific Speed

- From dimensional analysis, can combine these to produce dimensionless (?) parameter with no size dependence

$$\frac{\Phi^{*1/2}}{\psi^{*3/4}} = \frac{Q^{1/2}}{\left(N\cancel{D}^3\right)^{1/2}} \frac{\left(N\cancel{D}\right)^{6/4}}{\left(Hg_o\right)^{3/4}} \qquad \frac{L^{3/2}t^{-3/2}}{L^{3/2}t^{-3/2}}$$

- Call this **Specific Speed**
  – scaling: e.g., for fixed N and head, larger thrust $\Rightarrow$ higher $Q \Rightarrow$ higher N_s

$$N_s = \frac{NQ^{1/2}}{\left(Hg_o\right)^{3/4}} \qquad Hg_o = \Delta p_o/\rho$$

often specified at max η point

in U.S. also dimensional version

- N in RPM
- Q in gpm sometimes ft³/s
- Drop g_o

$$\frac{RPM\sqrt{gpm}}{ft^{3/4}} \div 2731 \rightarrow \text{dimensionless}$$

$$\frac{RPM\sqrt{ft^3/s}}{ft^{3/4}} \div 128.9 \rightarrow \text{dimensionless}$$

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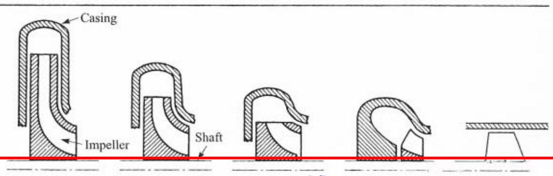
Pump Types

- Based on required p rise (head), flowrate and RPM limits (max ~200-700m/s impeller tip speed) $\Rightarrow N_s$
  (weaker steels – titanium)
  – can choose best pump type

TABLE 10-2. Pump Types

Adapted from Sutton

	Impeller type				
	Radial	Francis	Mixed flow	Near axial	Axial
Basic shape (half section)					



	Centrifugal/radial flow pumps		Mixed flow pumps		Axial flow pumps
Specific speed N_s U.S. nomenclature	500–1000	1000–2000	2000–3000	3000–~~6000~~ 7000	Above ~~8000~~ 7000
SI consistent units	~~0.2–0.3~~	~~0.4~~	~~0.6–0.8~~	~~1.0–2.0~~	~~Above 2.5~~
	0.18-0.37	0.37-0.73	0.73-1.1	1.1-~~2.2~~ 2.6	>~~2.9~~ 2.6
Efficiency %	50–80	60–90	70–92	76–88	75–82

to be consistent with US ranges

Specific Diameter

- Already defined normalized rotational speed

$$N_s = \frac{NQ^{1/2}}{(Hg_o)^{3/4}}$$

- Can also combine ψ* and Φ* to remove N dependence

$$\frac{\psi^{*1/4}}{\Phi^{*1/2}} = \frac{(Hg_o)^{1/4}}{(\cancel{N}D)^{2/4}} \frac{(\cancel{N}D^3)^{1/2}}{Q^{1/2}} \qquad \frac{L(L/t)^{1/2}}{L^{3/2}/t^{1/2}}$$

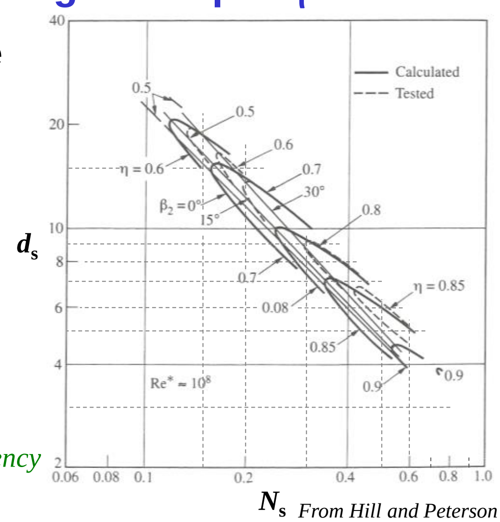
  – call this **Specific Diameter**

$$d_s = \frac{D(Hg_o)^{1/4}}{Q^{1/2}}$$

Centrifugal Pump Sizing: Example η Contours

- Each point is single point design (d_s,N_s)
- Narrow high efficiency region
  – given d_s can't change N_s much or η↓
  – high η ⇒ high N_s and low d_s

maximum efficiency near N_s=1



From Hill and Peterson

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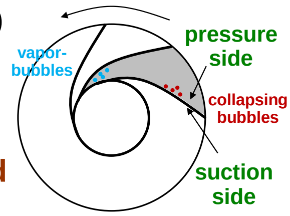
Cavitation Prevention

- If static p at inlet too low vs. fluid's vapor pressure, then as p drops over suction side of blades get vaporization (bubbles)



- Recall **Net Pos. Suction Head**

$$NPSH = \frac{p_{o_{in}} - p_{vap}}{\rho g_o}$$

min NPSH gives inlet p to prevent cavitation

- Convert to **Suction Specific Speed**

$$S_s = \frac{NQ^{1/2}}{\left(NPSH \cdot g_o\right)^{3/4}}$$

low flow rates lead to vibrations and excessive heating

- Typically S_s ~ 3.7–9.1 best designs with nearly no cavitation $S_{s,min}$ ~1.8 H_2, ~1.2 RP

  ~ 10–15 with controlled cav.

  even up to 30 *with modern inducer designs*

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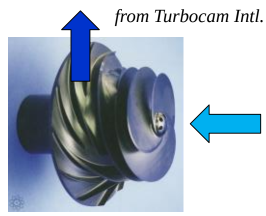
Cavitation Prevention

- What if we want to operate at RPM that gives NPSH < available head? $H_{avail} = H_{\text{tank}} + H_{\text{grav. head}} - H_{\text{line losses}}$

1. raise tank storage pressure
   – weight issue
2. add inducer
   – typically helical (screwlike) inducer to raise pressure before impeller
   – also used on axial/mixed pumps
3. separate booster pump
   – like having separate inducer on its own (low speed) shaft
   – could require extra turbine or more gearing

from Turbocam Intl.



Inducer Cavitation

cavitation bubbles, but collapsing by end of inducer

S_s=3.2

S_s=10

pump discharge pressure dropping by few %

bubbles impacting impeller

large reduction in discharge pressure (“head loss”)

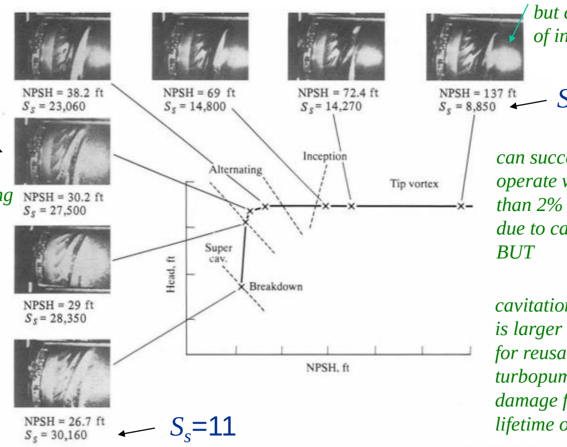


FIGURE 13.15 Experimental inducer cavitation characteristics. (Courtesy Rockwell International, Rocketdyne Division.)

can successfully operate with more than 2% head loss due to cavitation, BUT

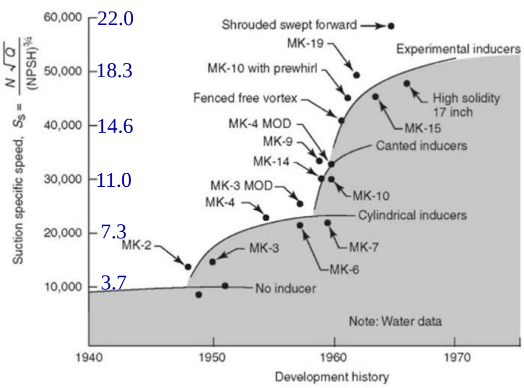
cavitation damage is larger concern for reusable turbopumps (less damage for short lifetime operation)

From Hill and Peterson

Inducer Development History

>25

- Maximum suction speed increasing with improved inducers



Rocketdyne Propulsion and Power

BOEING

SSME Low P H_2 Pump Inducer

LH_2 is compressible
$\Rightarrow$ temperature also drops as p reduced during flow acceleration
$\Rightarrow$ p_{vap} drops
$\Rightarrow$ reduced cavitation
$\Rightarrow$ higher pump specific speed or RPM for LH_2

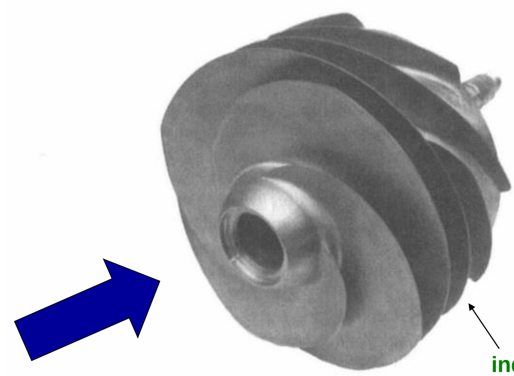


FIGURE 10–8. Fuel pump inducer impeller of the Space Shuttle main engine low-pressure fuel turbopump. It has a diameter about 10 in., a nominal hydrogen flow of 148.6 lbm/sec, a suction pressure of 30 psi, a discharge pressure of 280 psi at 15,765 rpm, an efficiency of 77%, and a suction specific speed of 39,000 when tested with water. (Courtesy of Pratt & Whitney Rocketdyne.)

S_s=14.3

From Sutton

Pump Design Example

- Consider preliminary design requirements for liquid oxygen pump
  – fluid properties: 1150 kg/m^3
  – flow: 257 kg/s (0.2235 m^3/s)
  – inlet: 85K, 1 bar
  – outlet: 120 bar (Head = 1055m)
- Constraints
  – stage loading coeff. 0.4-0.7
  – assume zero swirl at inlet, incompressible

Pump Design Example

- Estimate blade speed (single-stage pump)
  - pick maximum ψ for minimum blade speed

$$U_2 = \sqrt{\frac{\Delta p_o}{\rho\psi}} = \sqrt{\frac{119\times10^5\,N/m^2}{1150\,kg/m^3\,(0.7)}} = 122\,m/s$$

What if LH_2? 70 kg/m^3 490 m/s

reasonable (<200-400) ⇒ feasible with standard alloys.

- Pump type
  - can we use centrifugal, $N_{s,max}$ ~ 0.4-0.7
  - for 0.4 $$N = \frac{N_s\left(Hg_o\right)^{3/4}}{Q^{1/2}} = \frac{0.4\left(1055m\ 9.8077\,m/s^2\right)^{3/4}}{\left(0.2235\,m^3/s\right)^{1/2}} = 870\frac{rad}{s} = 8300\ rpm$$

Pump Design Example

- With N_s=0.4, good effic. for d_s ~ 6.5 (~86%)

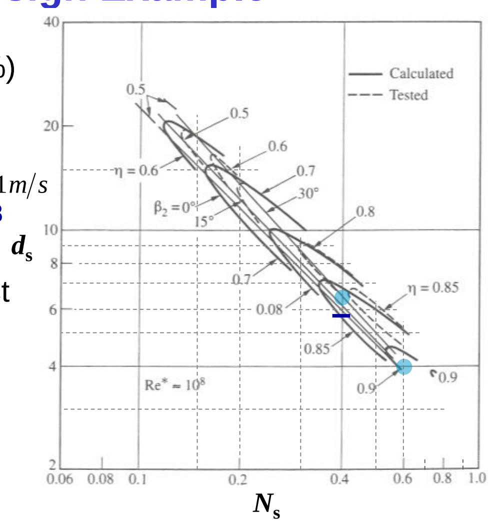
$$D = \frac{d_s Q^{1/2}}{\left(Hg_o\right)^{1/4}} = 0.30m$$ *not too big*

$$U_2 = (D/2)\Omega = (0.15)870\,m/s = 131\,m/s$$

$\Rightarrow \psi$=0.68

- Can make smaller by reducing η; small effect
- Can also increase N_s
  - N_s=0.6, d_s=4 (η>90%??)

  $\Rightarrow$ 12,500 *rpm*, 0.18*m*, *U*=118 *m/s*, ψ=0.75 *too high?*



Cavitation Check

- Check to see if need booster pump or inducer

$$NPSH = \frac{p_{o_{in}} - p_{o_{vap}}}{\rho g_o} = \frac{(1-0.567)10^5\,Pa}{1150\,kg/m^3\,(9.81\,m/s^2)} = 3.8m$$

- For N_s=0.6

$$S_s = \frac{NQ^{1/2}}{(NPSH \cdot g_o)^{3/4}} = 41$$

$$p_{vap}(bar) = 10^{3.85845 - \frac{325.675}{T(K) - 5.667}}$$

webbook.nist.gov/cgi/cbook.cgi?ID=C7782447&Mask=4

  - WILL NEED booster pump or higher storage pressure (too high for inducer)
  - or will need to reduce rpm to level where we can use inducer (e.g., S_s<25 $\Rightarrow$ N_s<0.36, close to our 1st choice, but slightly larger pump)