Mechanical/Thermal Energy Storage & Recovery – A Case for High Performance Turbomachinery Design

2020 Thermal, Mechanical, and Chemical Energy Storage (**TMCES**) Workshop
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Message

- Turbomachines and turbomachinery systems are a key element in many energy storage schemes, and improvements in their performance and functionality have a direct link to techno-economic viability.

- A wide array of turbomachinery aero/mechanical arrangements can potentially enable practical realization of various energy storage thermodynamic cycles.

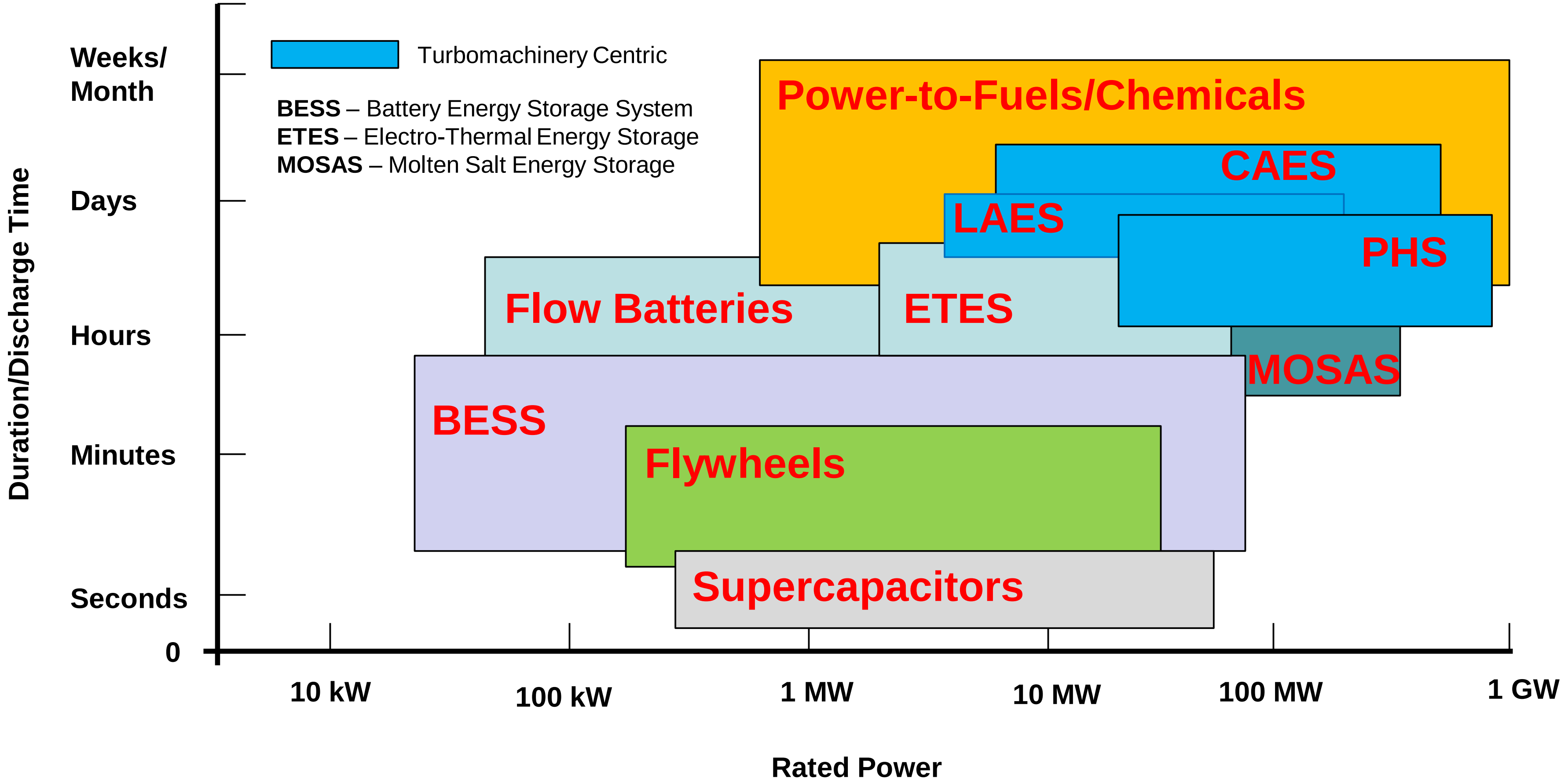
- Leveraging systems-level integrated thermodynamic cycle optimization coupled with turbomachinery topological design synthesis methods and practices from turbomachinery OEMs, may lead to significant advancements.

- Energy storage R&D programs should not only focus on "*off-the-shelf*" turbomachinery components, but also motivate the development of *novel and improved turbomachinery architectures* for maximum system benefit.

And the wheel keeps turning - innovative turbomachinery designs matter to a reduced-carbon energy supply chain!

Backdrop:
Energy Storage &
Turbomachinery Performance

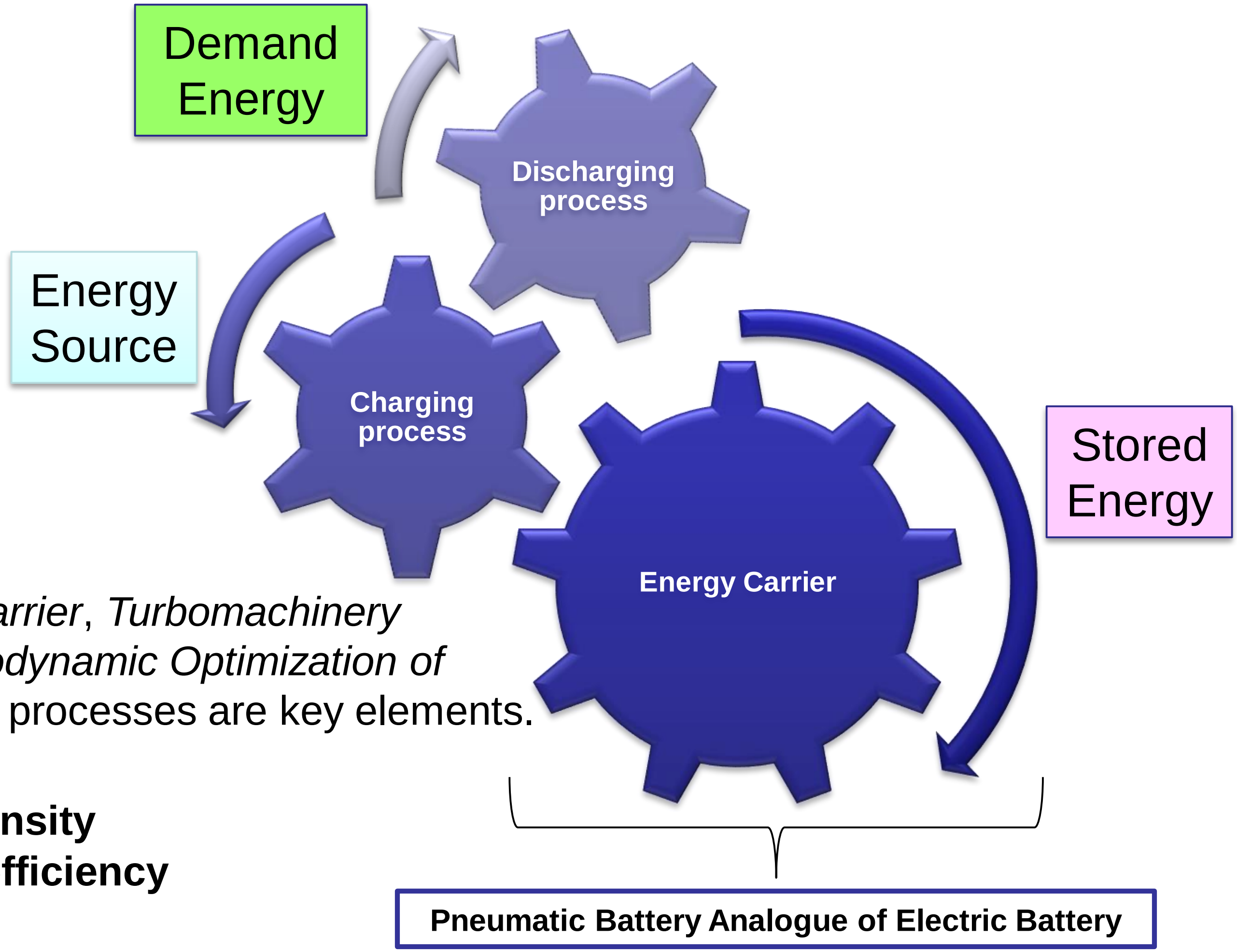
Variety of Grid-Scale Storage Solutions:
Off-the-shelf or “Clean Sheet” Turbomachinery?



Energy Storage Affordability: Balancing the techno-economics of power density, efficiency, and reliability…

Turbomachinery Centric Mechanical/Thermal Energy Storage

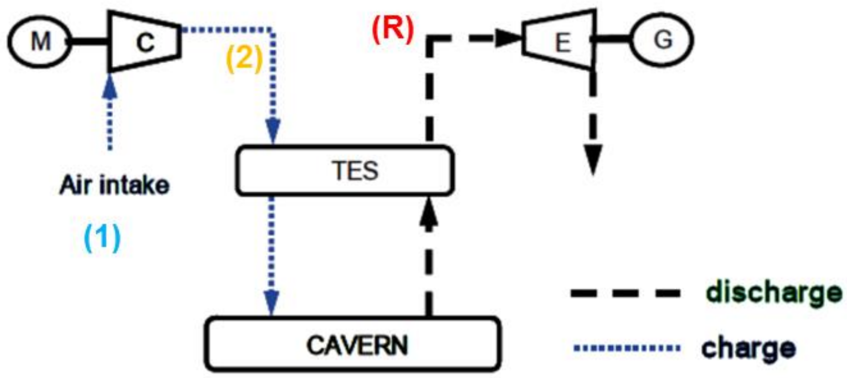
"Power cycles split into charging & discharging processes"



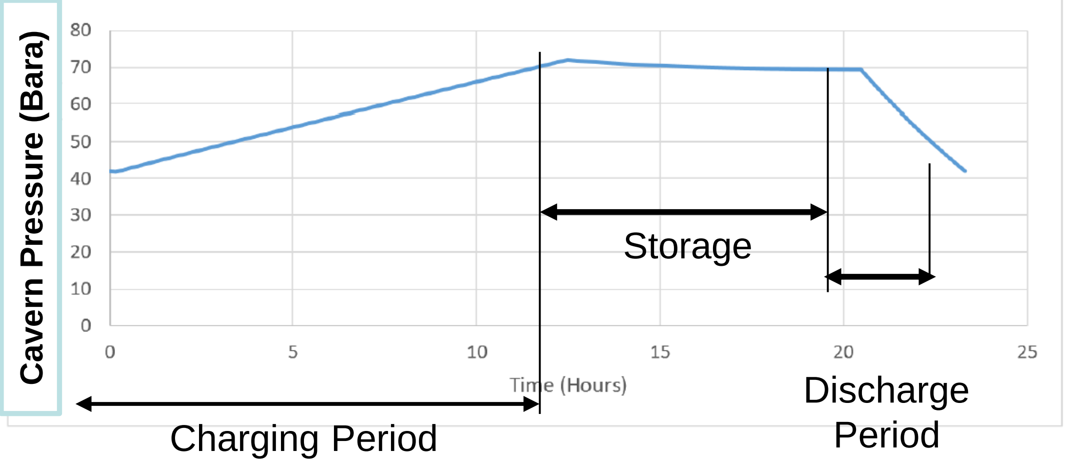
- For a chosen *Energy Carrier*, *Turbomachinery Technology* and *Thermodynamic Optimization of Charging & Discharging* processes are key elements.
- Figures of merit:

  - High power density
  - High storage efficiency
  - Affordability

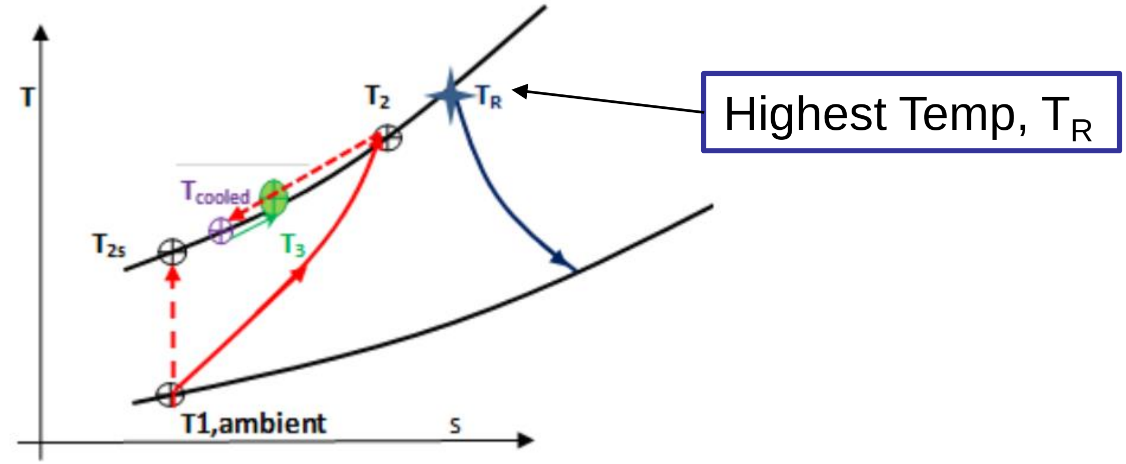
Basic Principle of Adiabatic CAES: Example of "Split" Brayton Cycle



Idealized Cavern Pressure Variation Profile Over an Operational Cycle



CAES Thermodynamic Optimization:
Importance of Turbomachinery Performance



Energy Recovery Efficiency:

$$E.R.E. = \frac{\Sigma \dot{W}_{out}\, \Delta Time_{discharging}}{\Sigma \dot{W}_{in} \Delta Time_{charging} + \dot{Q}_{reheat}\, \Delta Time_{discharging}}$$

where:

$$\dot{W}_{in} = \dot{W}_{net\ compressor} + \dot{Q}_{auxiliary\, heat\, input}\ f$$

$$0 \leq f \leq 1, \quad f = fraction\ of\ time\ thet\ Auxiliary\ Reheater\ is\ used$$

Idealized Analysis
Japikse & Di Bella, 2018

$$R_c = \frac{(T_3 - T_{1,ambient})}{(T_2 - T_{1,ambient})} \qquad Rr = \frac{(T_R - T_3)}{(T_2 - T_3)}$$

$$E.R.E._{net} = \frac{T_R}{T_{ambient}} \frac{B\, \eta_{turbine}\, \eta_{compressor}}{A\,[1 + Rr - Rr\, Rc]}$$

$$A = (Pr, compressor)^{\left(\frac{k-1}{k}\right)} - 1 \qquad B = 1 - \left(\frac{1}{Pr, turbine}\right)^{\left(\frac{k-1}{k}\right)}$$

Why High Efficiency Turbomachinery?

Energy Recovery Efficiency (ERE):

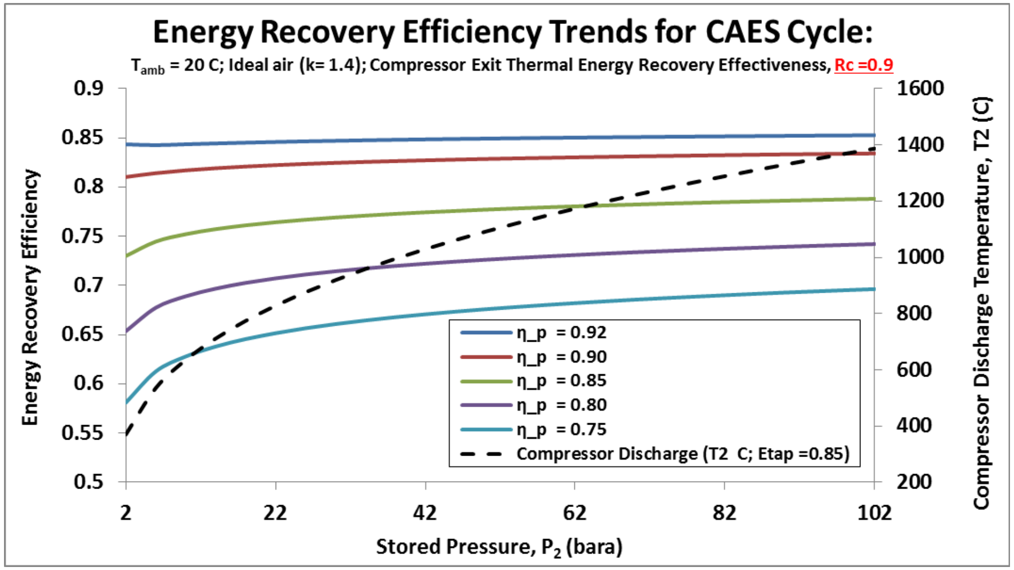
- Ideal air
- No system pressure drop
- No external heating (T_R / T_3 = 1)

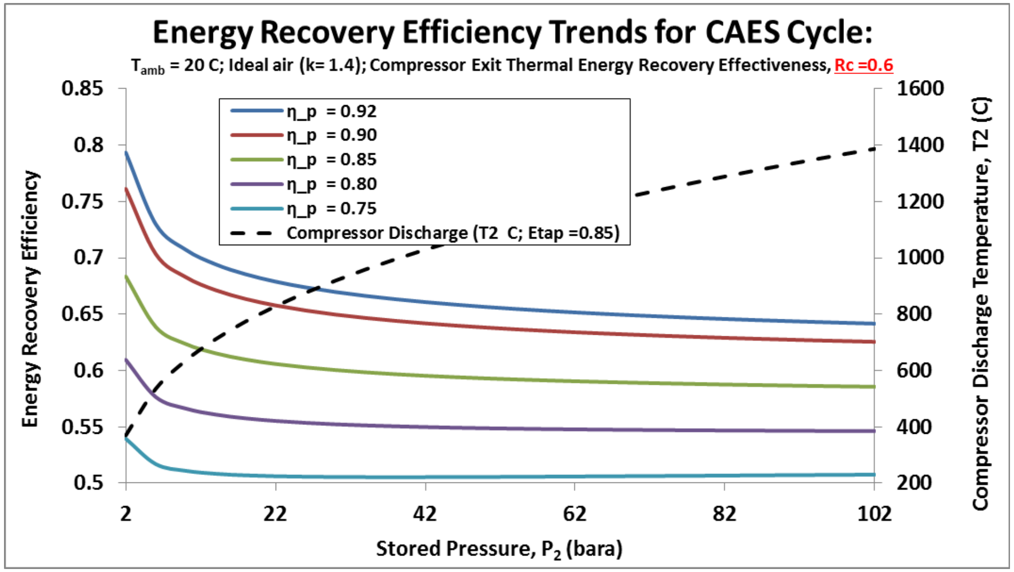
$$ERE = \left\{1 + R_c \frac{A}{\eta_c}\right\} \frac{B}{A} \eta_c \eta_t$$

Thermal Enhancement — Mechanical Storage Efficiency

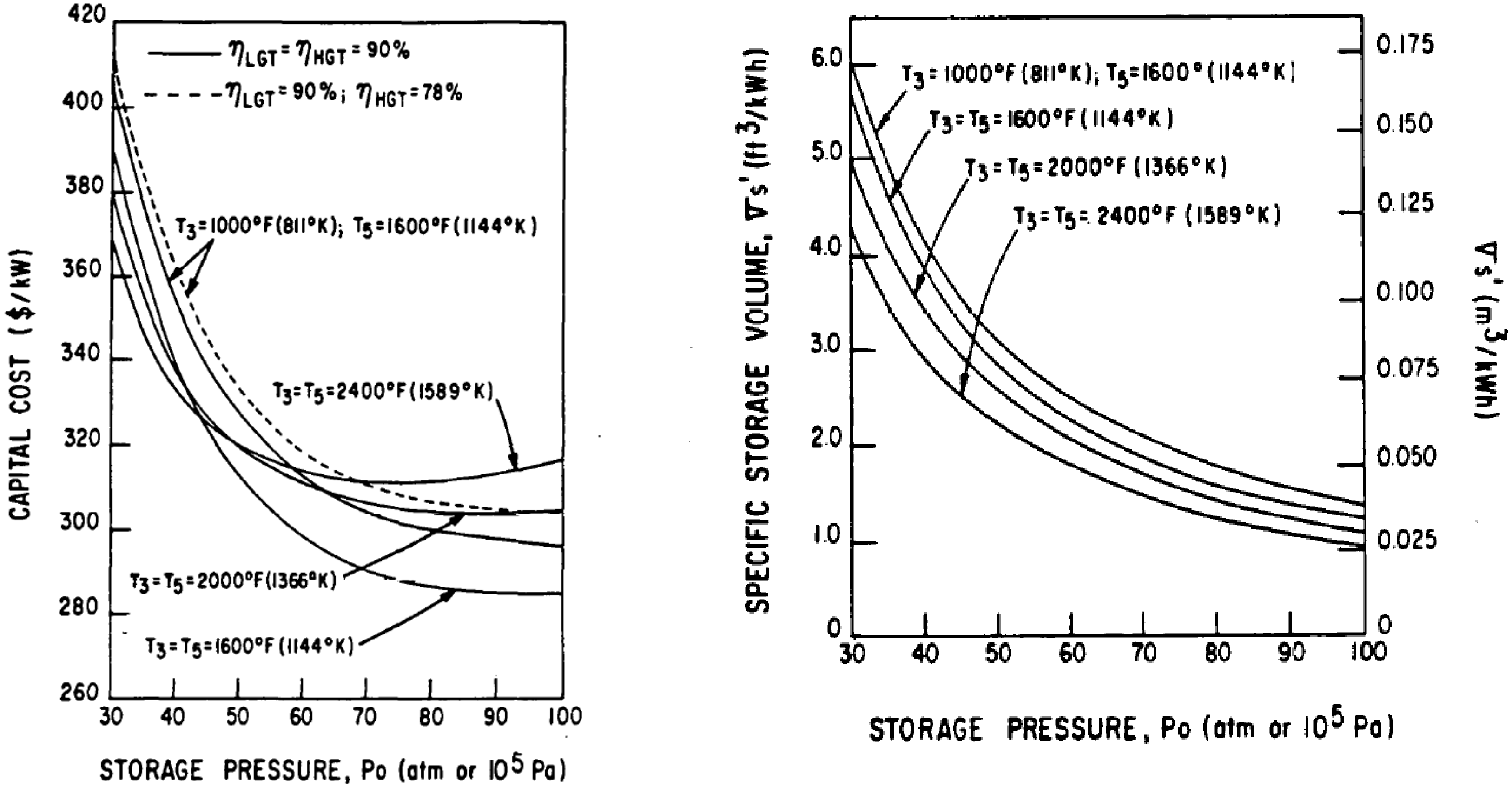
If $R_c = \eta_c$, $ERE = \eta_c \eta_t = R_c \eta_t$

- For turbomachinery polytropic efficiency, $\eta_p \geq R_c$, ERE is nearly independent of storage pressure.
- For compressor exit energy recovery effectiveness, R_c = 0.6, ERE decreases with storage pressure.
- High stored pressure can adversely impact compressor off-design matching and performance





Design Storage Pressure Impacts Cavern Volume, Turbomachinery Arrangement, and Capital Cost



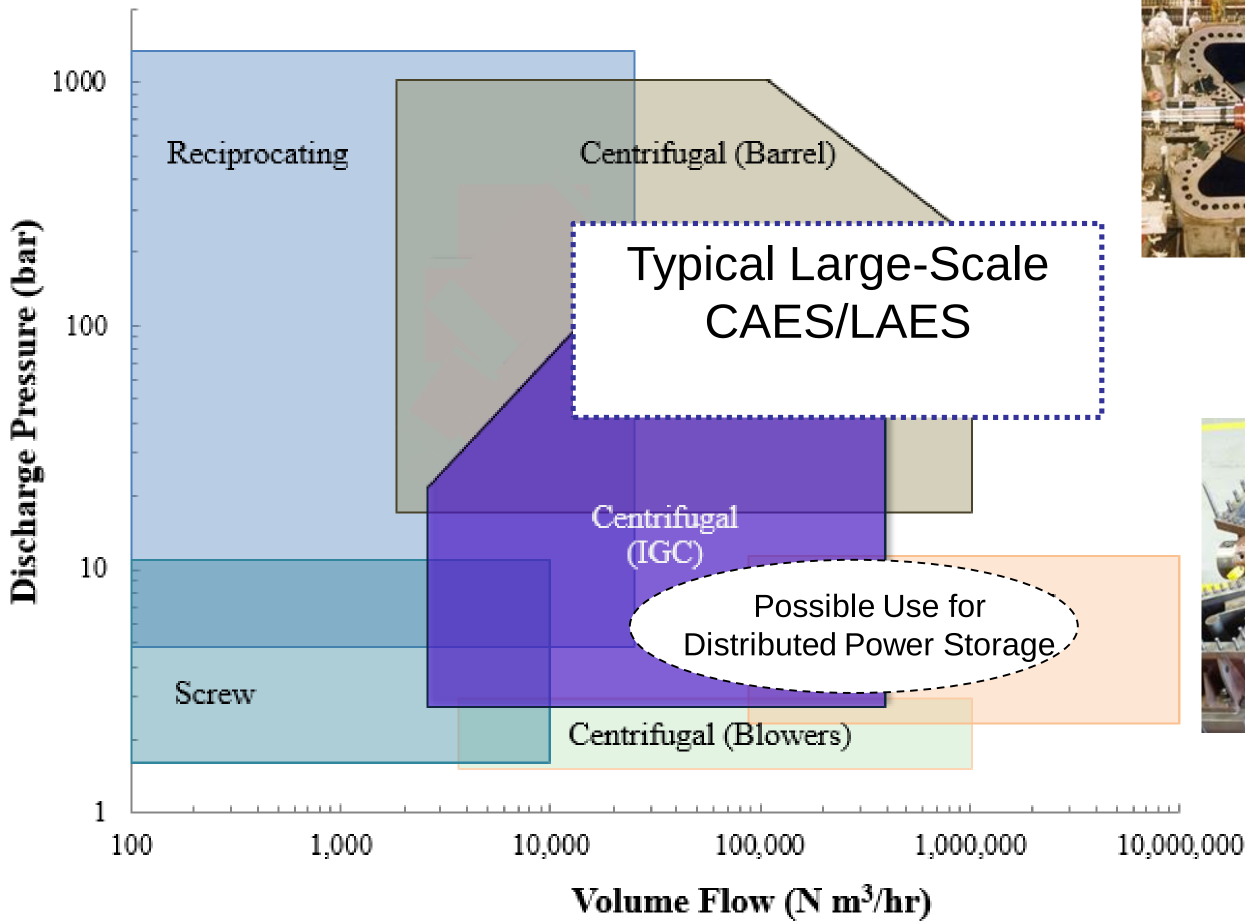
* Kartsounes and Kim, 1978, Argonne National Lab

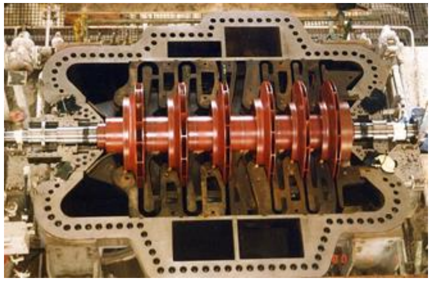
Some Turbomachinery Design & Development Challenges for TMCES

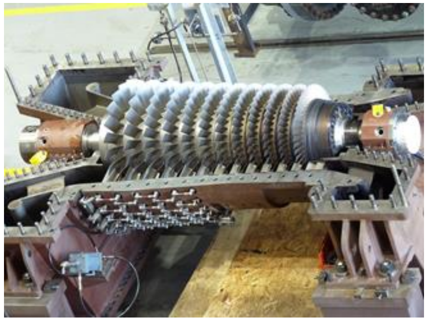
- System Performance (efficiency) & Operability (variability)
  - Efficient & flexible architectures, including blading shape
  - Off-design performance matching & flow range
- Cyclic Operation
  - Startups & shutdowns
  - Fatigue life
  - Rotary inertia
- High Pressures and Temperatures
  - Materials
  - Clearances, seals, and bearings
  - Thrust management
  - Rotor assembly
  - Equipment protection in hostile environment
- Hostile Environment
  - Internal
  - External
  - Freezing concerns in expander

These challenges are shared by OEMs across various turbomachinery applications sectors!

Zones of Applicability for Typical Industrial Compressor Types*





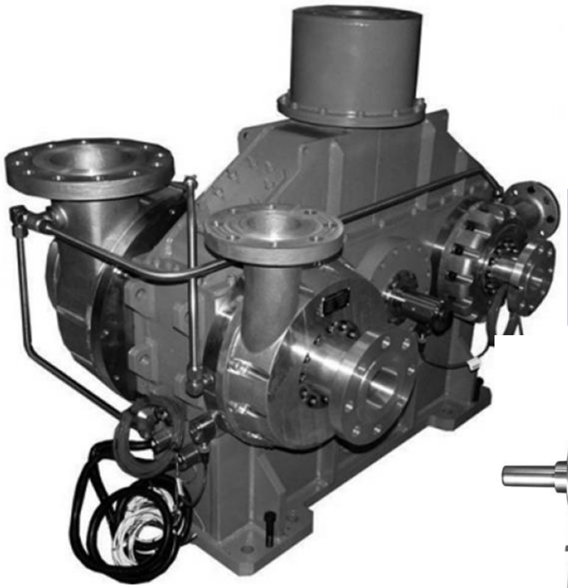


*Copied from Wygant et al., Asia Turbomachinery & Pump Symposium 2016

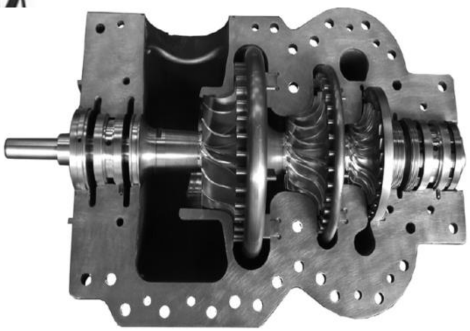
Turbine/Expander Options

- Typical off-the-shelf turbine options for CAES or high-pressure applications:
  - Two modules on single spool: HPT & LPT
    - HPT modified from axial steam turbine design
    - LPT from a conventional power generation gas turbine engine
- Incremental technology upgrade
  - Integrally Geared Multi-stage Radial Inflow Expander

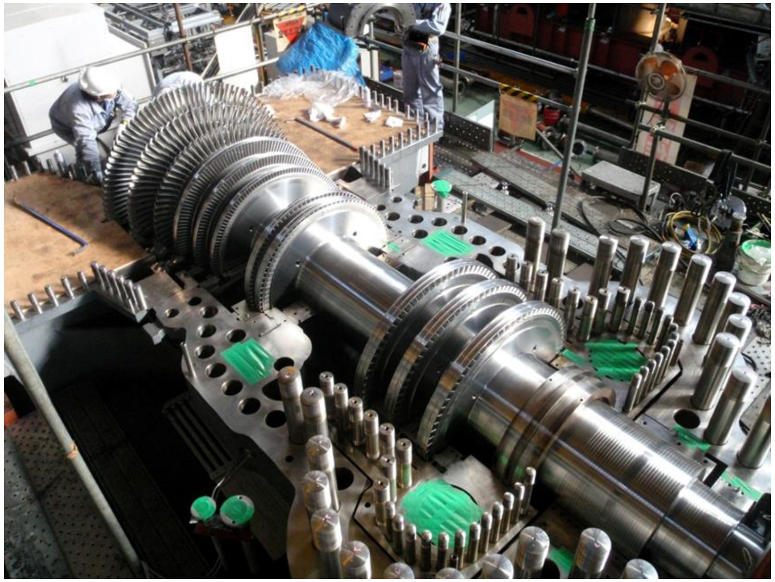
Highview Power Storage



Multi-stage Radial Inflow



Elliott Steam Turbine



Turbomachinery Design Practice in Support of Energy Storage System Development

Turbomachinery Design Phases
(from discovery to mastery of a particular solution)

Technology Portfolio

Specifications

Business Intelligence

Soft Computing: Discovery

<u>Design Synthesis:</u>

- ✓Configuration & Architecture
- ✓Design Intent & Performance Potential
- ✓Development Risks
- ✓Life Cycle Costs

<u>Design Development:</u>

- ✓Optimization – “Commercial Optimum”
- ✓Functional Verification
- ✓Risk Mitigation Tactics
- ✓Prototype Manufacturing

Hard Computing: Mastery

Design Validation

Similarity Considerations for Compressor Design, Selection & Performance Correlation

<u>Specified Design Operating Conditions</u>

- Pressure rise or polytropic head, $\boldsymbol{h_p}$
- Suction volume flow rate, $\boldsymbol{Q_0}$
- Gas suction speed of sound & kinematic viscosity, $\boldsymbol{a_0}$, ν_0
- Impeller mean diameter, $\boldsymbol{D_{2m}}$
- Impeller structural material specific strength, $\boldsymbol{\sigma/\rho_s}$
- Rotor shaft speed, $\boldsymbol{\Omega}$

Generalized Dimensionless Specific Speed, $\mathbf{n}_i = \dfrac{\Omega\, Q_0^{0.5}}{(v_i\ )^{1.5}}$

$\boldsymbol{v_i}$ is a set of effective velocities, expressed in terms of specified operating conditions, linked to certain forces which determine the action (kinematic & dynamic) of the machine. For example:

$$v_\sigma = (\sigma/\rho_s)^{0.5}\ ;\ v_s = (h_p)^{0.5}$$

***<u>A Few Dimensionless Turbomachinery Operating Conditions</u>*:**

Basic Specific Speed, $n_s = \dfrac{\Omega\, Q_0^{0.5}}{h_p^{0.75}}$ Specific Diameter, $d_s = \dfrac{D_{2m}\, h_p^{0.25}}{Q_0^{0.5}}$ Stress Specific Speed, $n_\sigma = \dfrac{\Omega\, Q_0^{0.5}}{(\sigma/\rho_s)^{0.75}}$

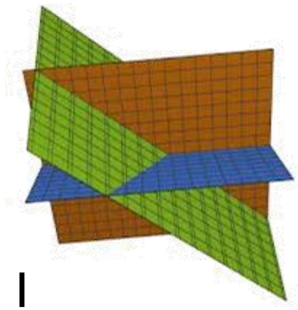
Viscocity Specific Speed, $n_\nu = \dfrac{\Omega\, Q_0^{0.5}}{(\Omega \nu_0\ )^{0.75}}$ Compressibility Specific Speed, $n_a = \dfrac{\Omega\, Q_0^{0.5}}{a_0^{1.5}}$

Broad Statement of the Design Problem

Stage design process can be mathematically characterized as:

$$\mathbf{n_i} = F_i \, (p_1, p_2, p_j, \ldots p_m)$$

Dimensionless Operating Conditions — **Physical Functional Relation** — **Dimensionless Design Parameters**



$i = 1, 2, \ldots, l$; $m > l$

p_j is a set of dimensionless *geometric, kinematic, and dynamic design parameters*
Since $m > l$, certain "design choices" need to be made in order to close this underdetermined system.

$\mathbf{F_i}$ is a set of functional relationships between a desired set of design parameters, as dictated by physical balancing principles:

$R_F\,(\mathbf{W}_F, G(\mathbf{X} : \mathbf{p_j})) = 0$ - Model Representation of Fluid Dynamics Constraint
$R_S\,(\mathbf{W}_S, G(\mathbf{X} : \mathbf{p_j})) = 0$ - Model Representation of Structural Dynamics Constraint
$G(\mathbf{X} : \mathbf{p_j})$ is the domain boundary geometric shape with geometric coordinates **X** and flow/structural states $\mathbf{W}_F$ **&** $\mathbf{W}_S$

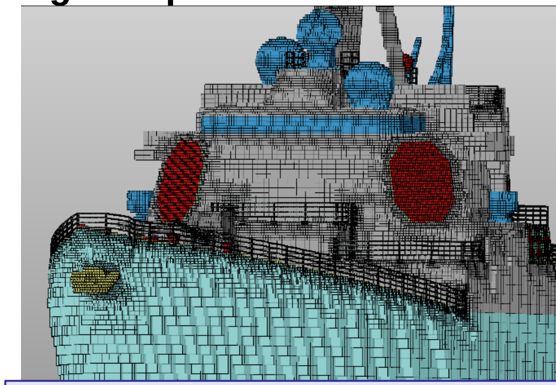
$G(\mathbf{X} : \mathbf{p_j})$ can be parameterized in terms of the geometric subset of p_j
$\mathbf{W}_F$ & $\mathbf{W}_s$ are linked to kinematic and dynamic subset of p_j

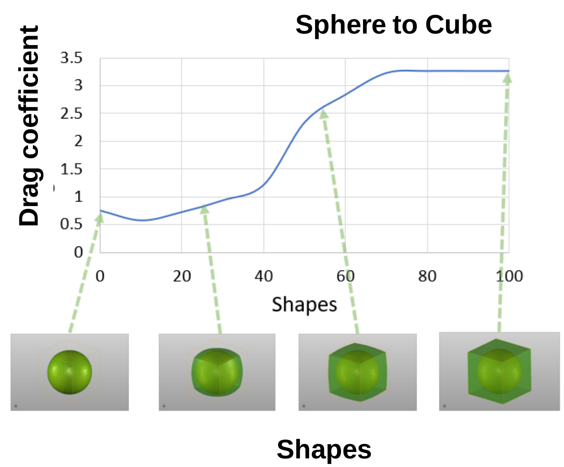
Determine the set of design parameters and associated geometric shape such that the desired set of design operating conditions is achieved…

Geometric Shaping is the Beginning & End of Turbomachinery Design:

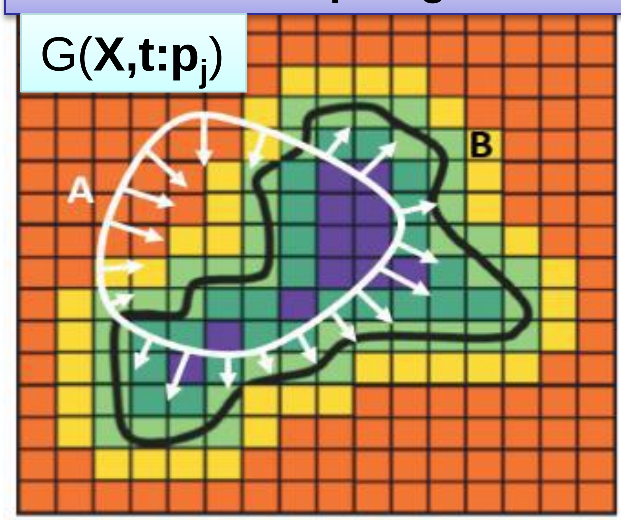
Digital Geometry & Morphing Between Different Body Shapes

Integer Representation of Geometry as Opposed to Traditional NURBS/BREP Construct





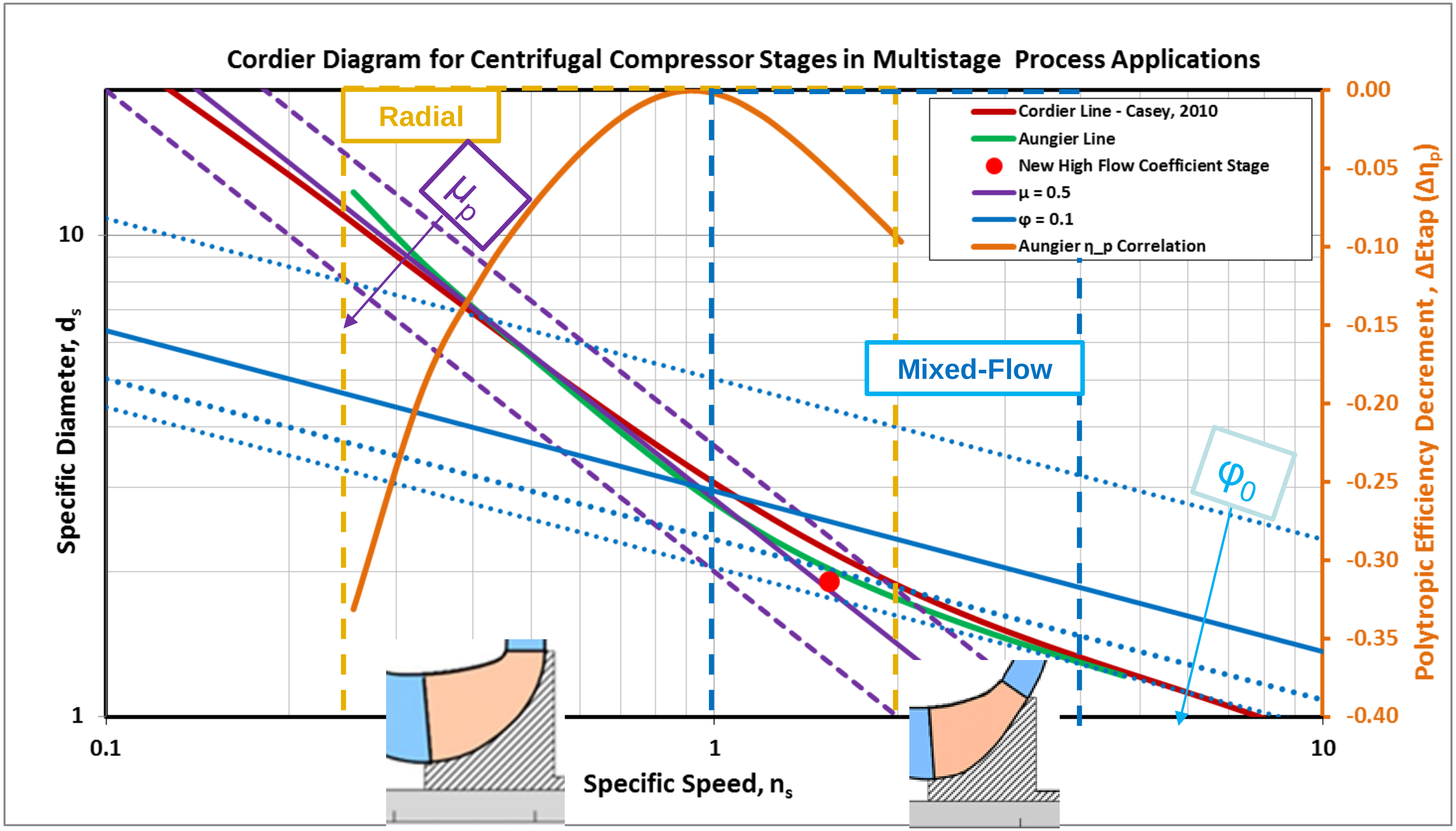
Geometric Morphing Process



- Morphing is based on parametric part representation and can be used as the main process in a solution to the design problem
- Three main step:
  - ❖ Blade Shape Genomic Mapping
  - ❖ Blade Shape Re-parenting
  - ❖ Adaptation of Technological Features

The Compressor Selection Chart:

Cordier Diagram as Solution to a Particular Phase of the Design Problem



Connecting Thermodynamic Cycle with Turbomachinery Design Selection

Overall Compressor Basic Specific Speed

$$N_s = \left(\frac{1}{\eta_p}\right)^{1/2} \left\{\frac{v_s}{H_p{}^{5/2}}\right\}^{1/2} \left(\Omega\sqrt{P_{sh}}\right)$$

Expected Overall Performance — **Specified Gas & Thermodynamic Cycle Conditions** — **Driver Requirements for Given Application**

v_s – Suction Specific Volume; P_{sh} – Overall Required Shaft Power; H_p – Overall Polytropic Head

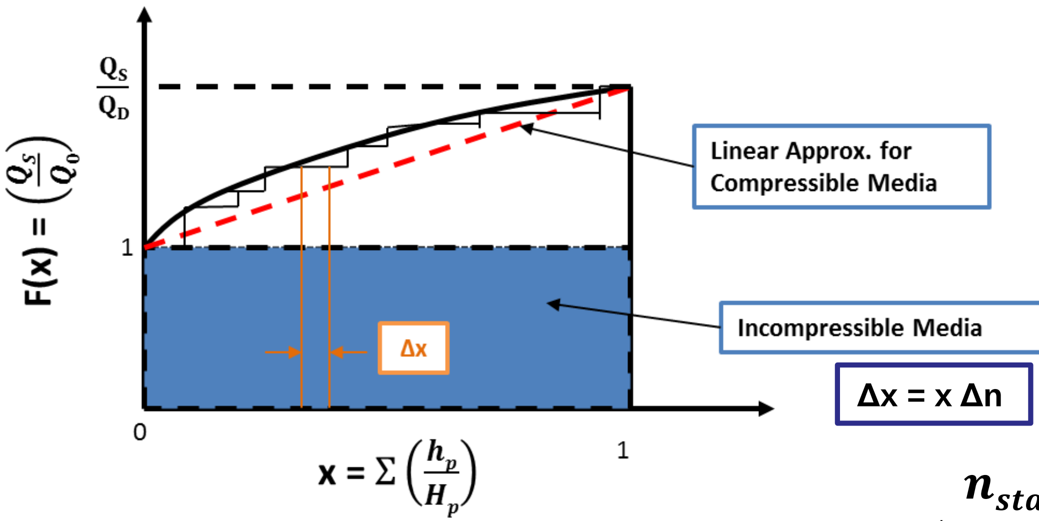
Connecting Overall Specific Speed to Individual Stage Specific Speed:

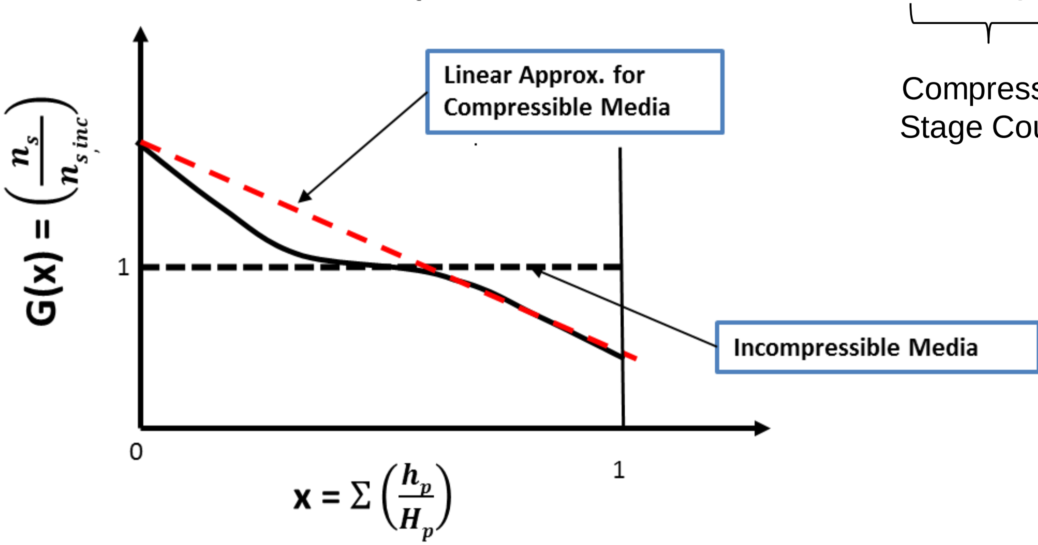
$$\frac{N_s}{n_s} = \left(\frac{Q_s}{Q_0}\right)^{1/2} \left(\frac{h_p}{H_p}\right)^{3/4}$$

$$Stage\ count, nstage_{,\ inc} = \left(\frac{n_{s\ inc}}{N'_s}\right)^{4/3}$$

Q_0 – Individual Stage Suction Volume Flow; h_p – Individual Stage Polytropic Head

Analogy Between Hypothetical Incompressible & Actual Compressible Multistage Machine





Stage count for equivalent pump with constant specific speed

$$n_{stage,\,inc} = \left(\frac{n_{s\,inc}}{N_s}\right)^{4/3}$$

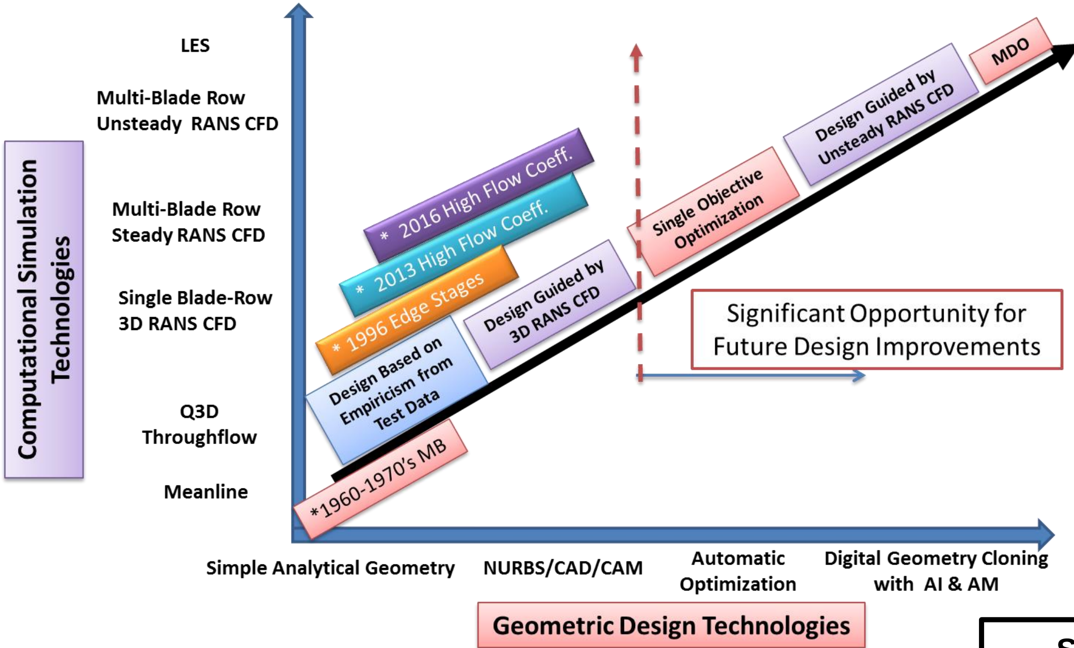
$$\underbrace{n_{stage}}_{\text{Compressor Stage Count}} = \underbrace{n_{stage,\,inc}}_{\text{Equivalent Pump Stage Count}} \underbrace{\int_0^1 \left(\frac{Q_s}{Q_0}\right)^{2/3} \left(\frac{n_s}{n_{s\,inc}}\right)^{4/3} dx}_{\text{Compressibility Correction}}$$

Variation of Impeller Average Diameter

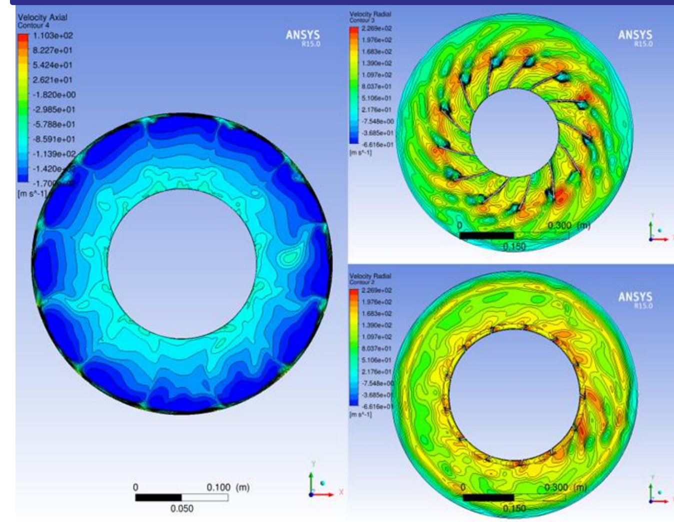
$$D_{2m} = ds\left(\frac{n_s\,Q_0}{\Omega}\right)^{1/3}$$

d_s from Cordier line given n_s

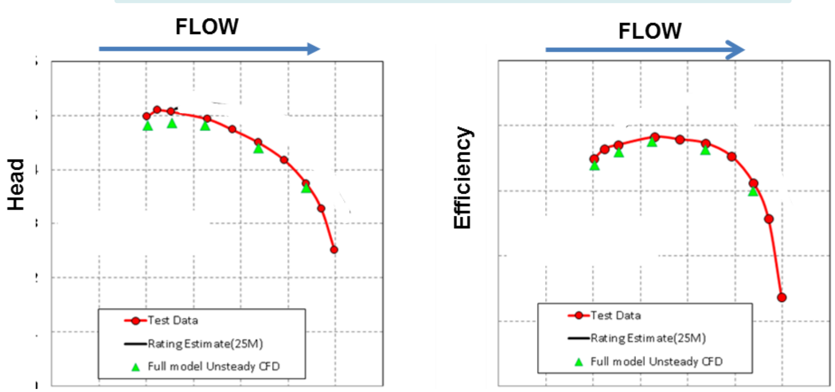
Well Anchored Design & Simulation Technologies Have Contributed Towards High-Performing Centrifugal Compressors



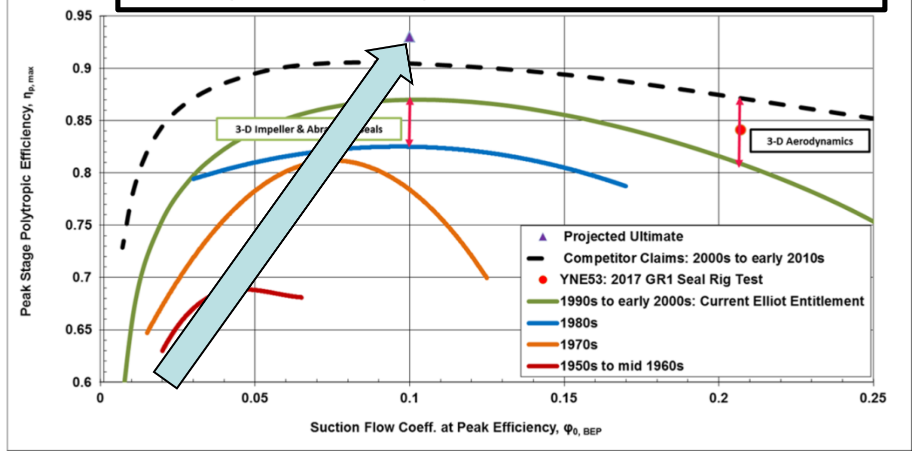
URANS Simulation of Stage Performance



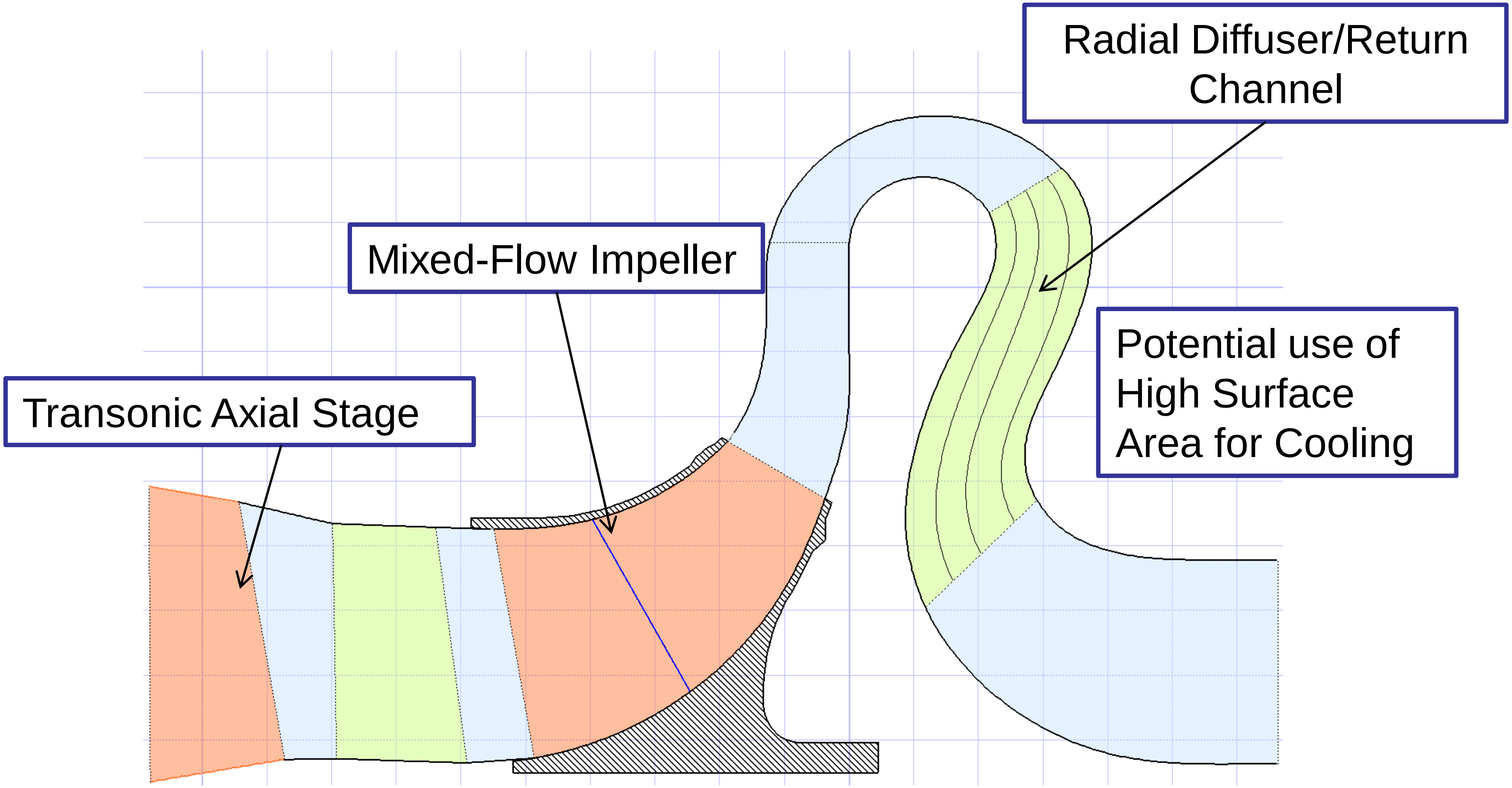
Calibration of Predictions with Test



Significant Stage Performance Evolution

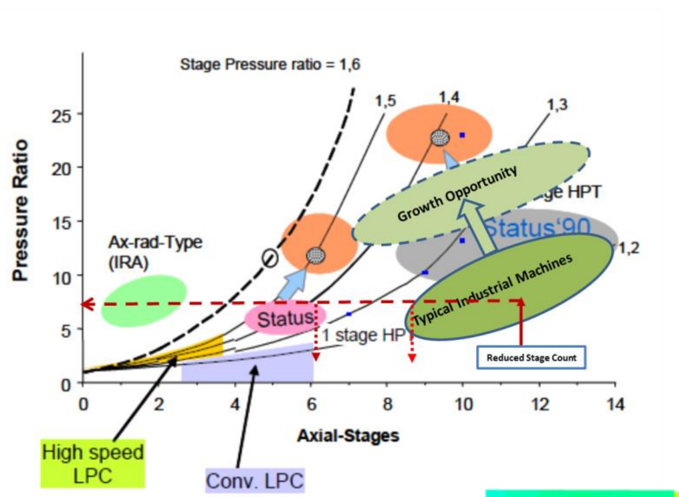


Efficient High Flow and High Pressure Ratio Compression System: Hybrid Architecture: Axial & Mixed-Flow



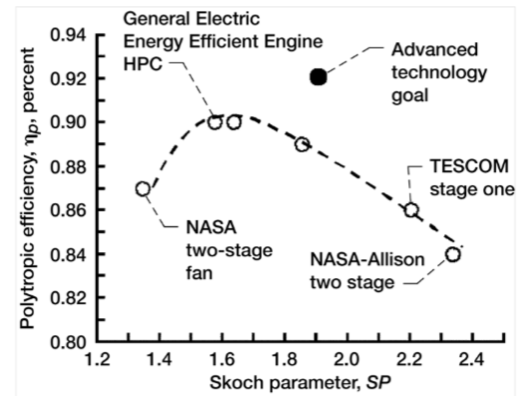
Improved Design Practice Enables Higher Stage Loadings without Penalizing Performance

Waschka et. al., ISABE-2005-1266



Similar Re, Specific Flow, and Stall Margin

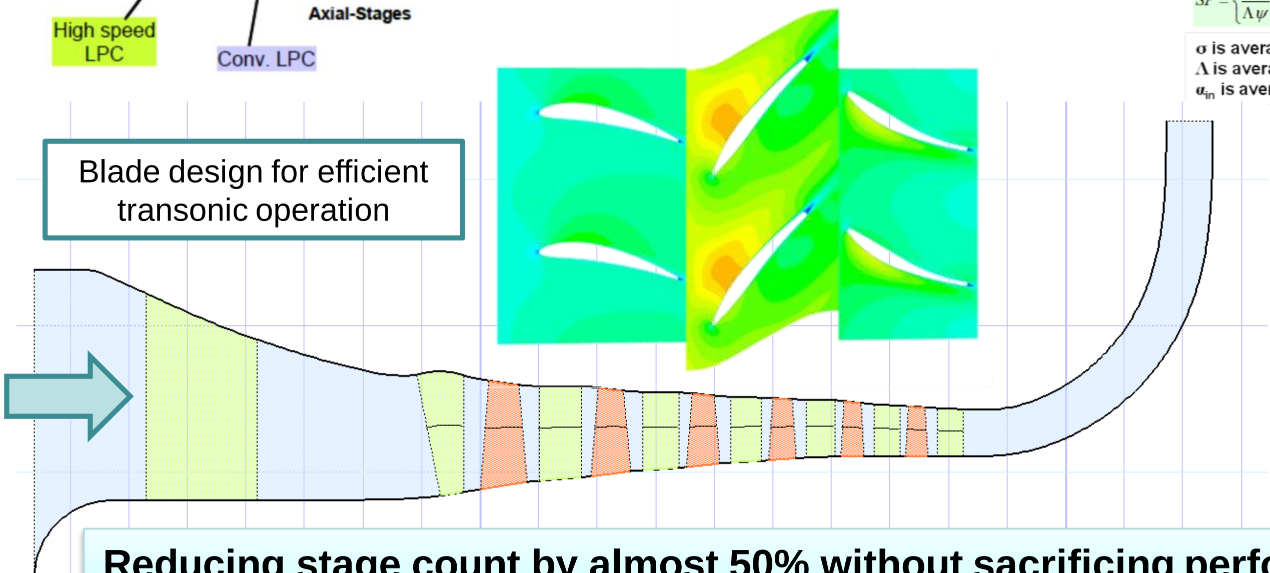
"Judicious Combination of Geometry and Vector Diagram"

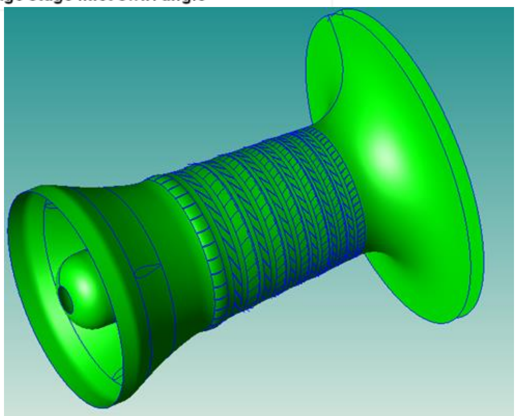


$$SP = \left\{ \frac{\sigma\phi}{\Lambda\psi} \sqrt{1 + (1/\phi - \tan\alpha_{in})^2} \right\}^{1/2}$$

σ is average solidity
Λ is average blade aspect ratio
α_{in} is average stage inlet swirl angle

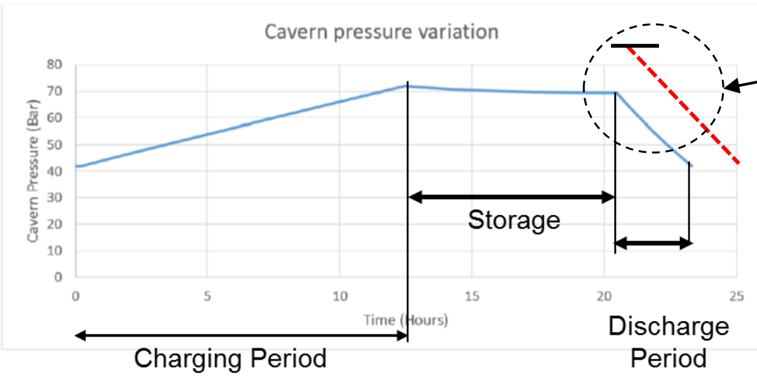
Blade design for efficient transonic operation

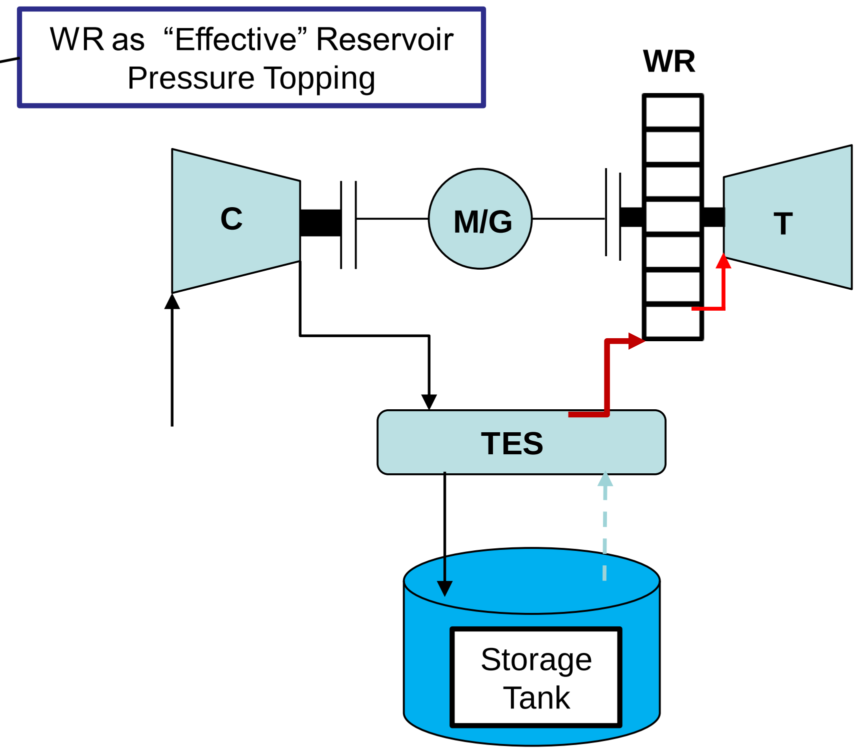




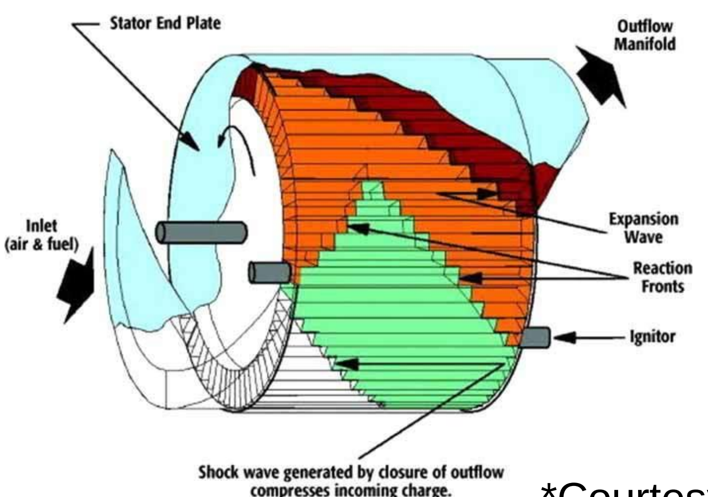
Reducing stage count by almost 50% without sacrificing performance and operability!

Integration of Wave Rotor in CAES – Synergies with Industrial Steam Turbine





<u>Wave Rotor* (WR): Pressure Exchanger</u>



*Courtesy of D. Paxon, NASA Glenn

Back to Message

- Turbomachinery performance and cost can play a major role in the practical realization of various energy storage technologies.

- Compelling evidence and reason were given for seeking out novel turbomachinery arrangements and achieving as high a turbomachinery efficiency as potentially available, taking full advantage of current computational simulation tools (involving AI) and advanced manufacturing techniques.

- Plenty of challenges remain. Multidisciplinary system-level thinking, along with careful appropriation of existing turbomachinery engineering know-how (coupled to emerging design methods) should light the path forward…

Thanks for listening!