



**ASME Turbo Expo 2022 –  
Tutorial Session on Modeling  
of Turbomachinery Secondary  
Flow Systems (GT2022-83726)**

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# Agenda

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- I. Motivation
- II. Introduction
- III. Secondary Flows – Flow Path Modeling
- IV. Heat Transfer
- V. Cooling
- VI. Considerations for Integration & Automation
- VII. Conclusions



# Motivation

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- Turbomachinery design involves multi-disciplinary and inter-components interactions
  - Describe how flow behaves under a given set of boundary conditions.
- Looking at each element individually is not sufficient enough to get accurate performance and operation for the entire system
- Iterative work is required
- Need accurate modeling of secondary flows for power production, axial loads, leakages, etc.
- Local phenomena can then be broken down to understand how to design, analyze, optimize and troubleshoot flow behavior





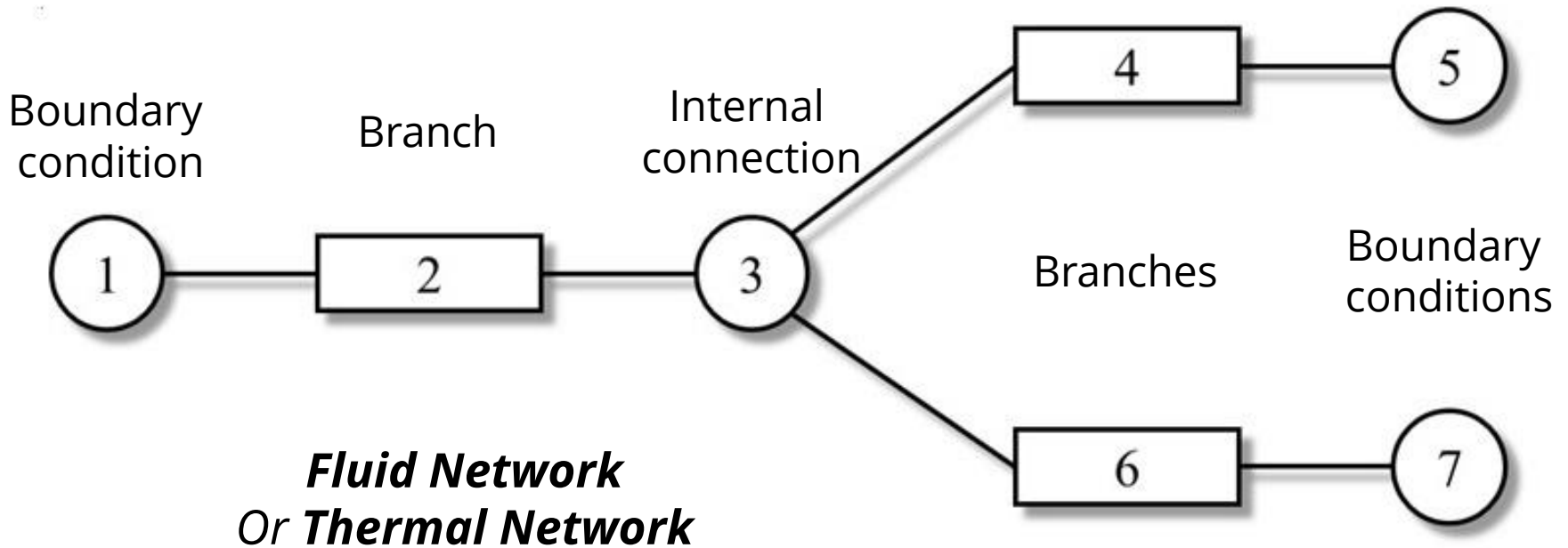
# Introduction

# Typical Thermal-Fluid Network Approach

- Goal: Model fluid and heat flow
- Approach: Represent system through different inter-connected sections of a fluid path and different parts of solid structure using 0D & 1D elements
- Fluid elements:
  - 1D elements – branches
    - Used to describe different resistances to fluid and heat flow
    - Have 2 nodes – inlet and outlet
  - 0D elements – nodes
    - Used to connect branches & setup boundary conditions
    - No restrictions on number of connections
- Thermal network: Simulates heat flow through solid structure + heat exchange between fluid flow and surrounding solid surfaces

# Typical Thermal-Fluid Network Approach

- Local initial state of fluid necessary to start calculation
- Main calculated parameters
  - Branches - mass & heat flow rates
  - Nodes - pressure and/or temperatures
  - Remaining parameters can be determined from these

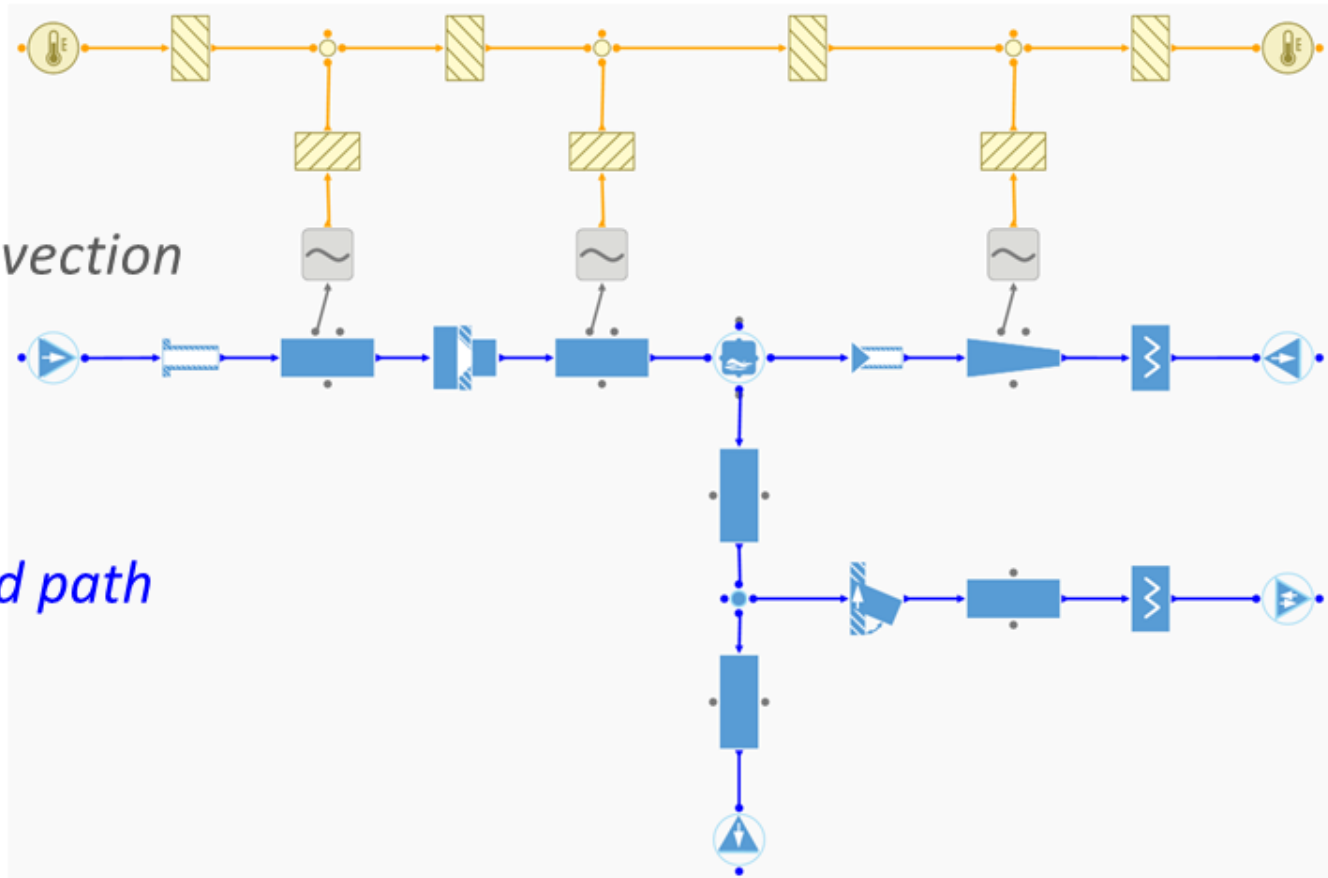


# Thermal-Fluid Network Example

*solid structure*

*convection*

*fluid path*

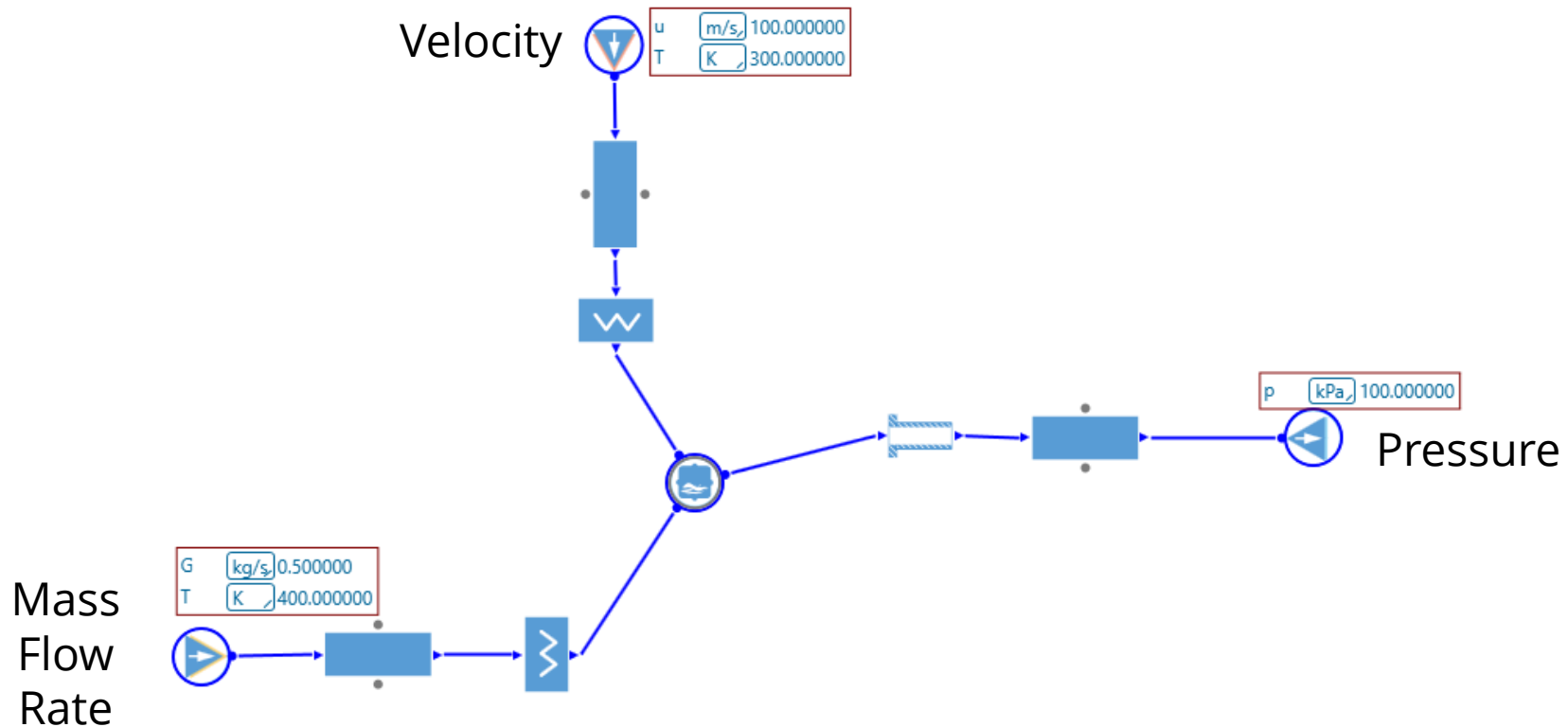


# Advantages of 1D networks

- 3D systems can be discretized and calculated as a thermal-fluid network to account for compressor bleed.
- Benefits:
  - 3D geometry is not required
  - Reduced computation time (compared with 2D-3D CFD)
  - Intuitive setup (drag-and-drop)
  - Different modeling levels
  - Scripting ability allows for automation

# Flexibility of Problem Formulation

- Different boundary condition elements allow specifying different constraints onto the model depending on the end goal
- Allows setting and sizing the system based on given constraints



# Modeling of fluids

- Different types of fluids can be modelled:
  - Single-species vs. mixtures
  - Ideal vs. real gas
  - Compressible vs. incompressible
  - Accounting for phase change vs. not
  - Etc.

$$p = \rho \cdot R_{specific} \cdot T;$$

$p$  – pressure;

$R_{specific}$  – specific gas constant.

Equation of state for ideal gas

- Example for incompressible fluids without phase change
  - Applicable to lubricating oils, liquid fuels, hydraulic fluids, etc.
  - Fluid properties required include enthalpy, entropy, heat capacity, density, thermal conductivity & viscosity
  - Fluid properties take into account temperature dependency using different models (ex. Krage equation) for a given reference pressure

$$C_p = \sum_{i=0}^4 (A_i \cdot T^i) = A_0 + A_1 \cdot T + A_2 \cdot T^2 + A_3 \cdot T^3 + A_4 \cdot T^4$$

# Steady vs. Unsteady Calculations

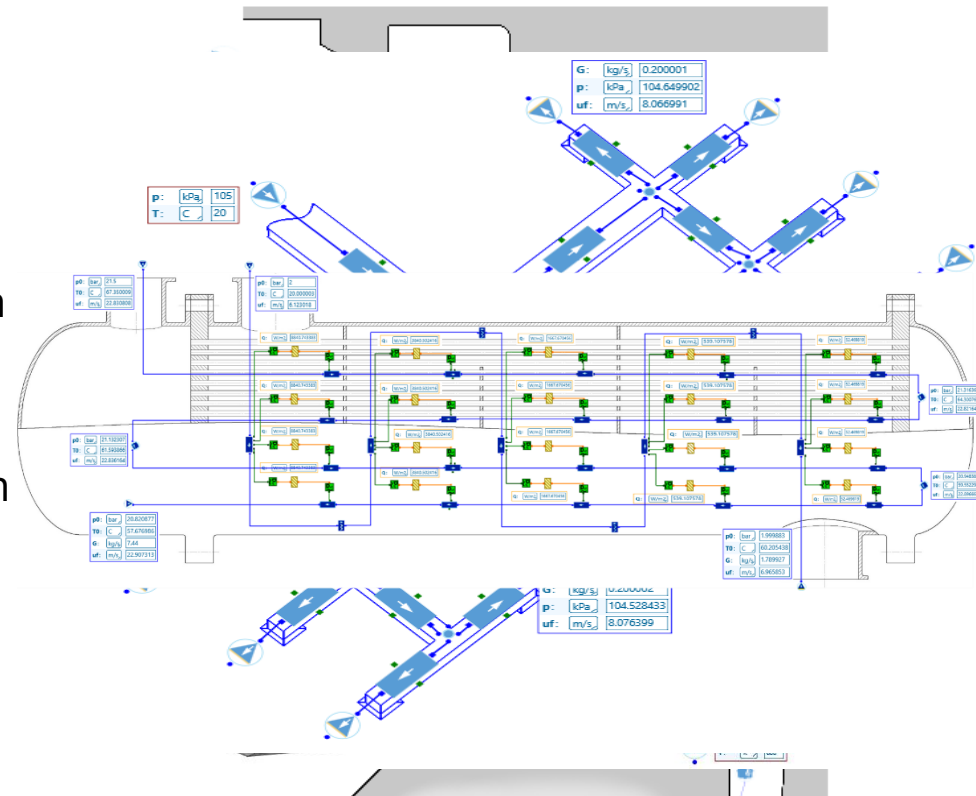
- Steady-state calculation
  - Used for established processes
- Unsteady calculation
  - Used for transient processes – properties and possibly BCs change over time
    - Filling/emptying of a chamber
    - Turbine start-up
    - Change of cycle load and therefore component operating conditions
    - Etc.
- 0D elements (nodes) are non-inertial components vs. 1D elements like chamber inside of which flow can accumulate and takes time moving in and out of
- Non-inertial elements propagate changes of state immediately, and their inertial properties, such as volume or length, are considered as negligible small.

# Turbomachinery Application Examples

## ➤ 1D thermal-fluid network modeling and analysis tool

### ➤ Applications:

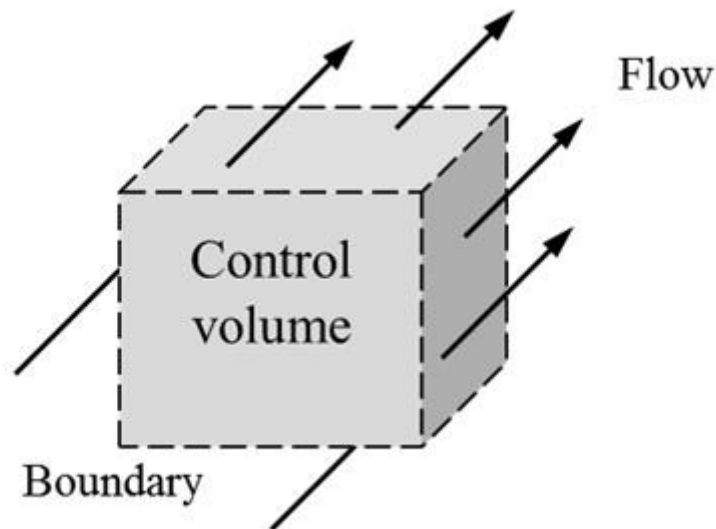
- Pipeline modeling
- Separation and mixing
- Convection modeling
- Gas turbine blade cooling system
  - Nozzles
  - Rotors
- Gas turbine secondary air system
- Steam turbine secondary flows
- Turbopump secondary flows
- Ventilation system
- Heat exchangers
- Other thermal-fluid systems



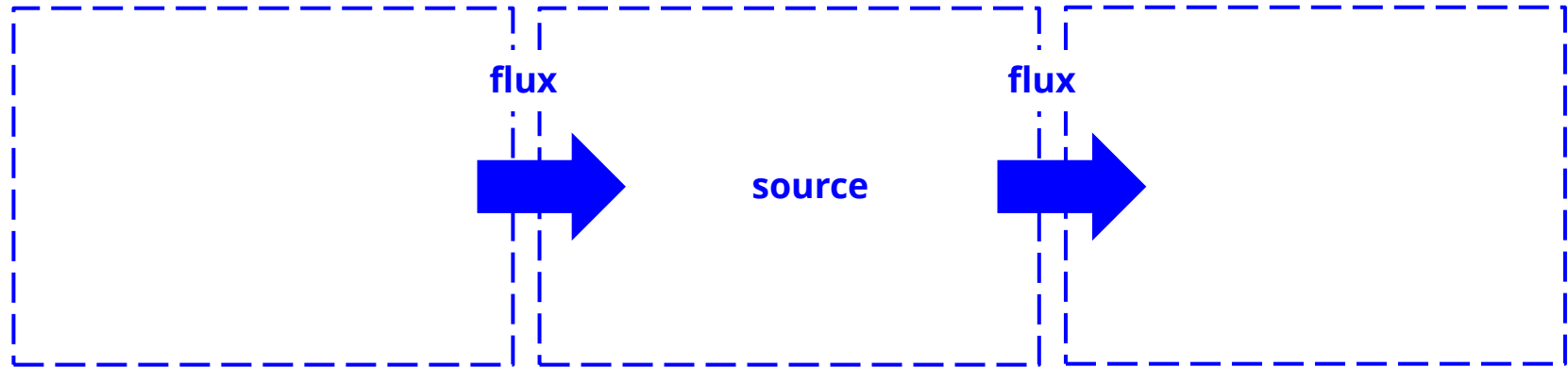
# Secondary Flows – Flow Path Modeling

# Basic Equations

- Conservation of mass, momentum & energy in fixed control volume
  - Rate of change of a physical quantity inside control volume (CV) depends on net flux of this quantity through the control volume boundary and source of this quantity inside CV or on its surface
- Finite volume solver

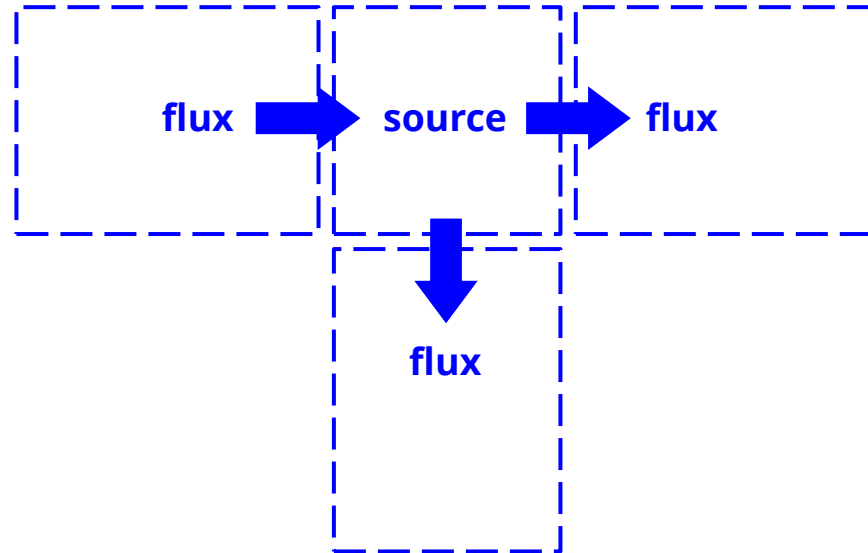


# Basic Equations in Fluid Branch Elements



$$\left( \begin{array}{c} \text{RATE OF CHANGE} \\ \textit{inside volume} \end{array} \right) + \left( \begin{array}{c} \text{FLUX} \\ \textit{through boundary} \end{array} \right) = \left( \begin{array}{c} \text{SOURCE} \\ \textit{inside volume} \end{array} \right)$$

# Basic Equations in Fluid Node Elements



$$\left( \begin{array}{c} \text{RATE OF CHANGE} \\ \textit{inside volume} \end{array} \right) + \left( \begin{array}{c} \text{FLUX} \\ \textit{through boundary} \end{array} \right) = \left( \begin{array}{c} \text{SOURCE} \\ \textit{inside volume} \end{array} \right)$$

# Basic Equations - Form

Rate of change of desired quantity

Net flux due to convective transport through boundary surface of control volume

$$\begin{aligned}
 & \frac{\partial}{\partial t} \int_V \rho \cdot dV \\
 & \frac{\partial}{\partial t} \int_V \rho \cdot \vec{u} \cdot dV \\
 & \frac{\partial}{\partial t} \int_V \rho \cdot E \cdot dV
 \end{aligned}
 = - \oint_S \rho \cdot \vec{u} \cdot d\vec{S} + s_m;$$

$$= - \oint_S \rho \cdot \vec{u} \cdot \vec{u} \cdot d\vec{S} + s_{mom};$$

$$= - \oint_S \rho \cdot E \cdot \vec{u} \cdot d\vec{S} + \oint_S k \cdot \vec{\nabla} T \cdot d\vec{S} + s_e$$

Source terms

Normal vector in any point of control volume surface directed outward

Energy flux due to heat diffusion in fluid flow

# Conservation of Ang. Momentum

- The total angular momentum of a system remains constant unless acted on by an external torque
  - Ensures accurate prediction of swirling flow through rotating components, such as cooling flow channels in rotating turbine blades

$$\frac{\partial}{\partial t} \int_V \rho \cdot \vec{u} \cdot dV = - \oint_S \rho \cdot \vec{u} \cdot \vec{u} \bullet d\vec{S} + s_{mom};$$

$t$  – time;

$\rho$  – density;

$V$  – volume;

$u$  – velocity;

$S$  – surface area;

$s$  – source;

$E$  – total energy per unit mass;

$k$  – thermal conductivity coefficient;

$T$  – temperature.

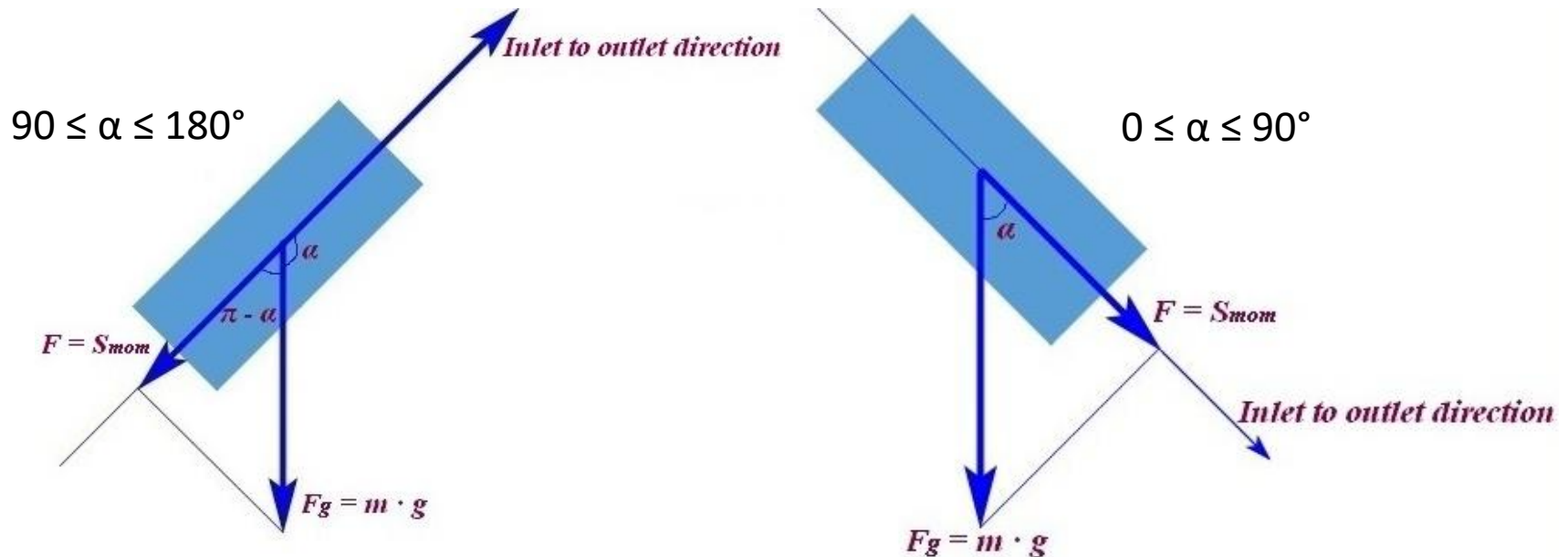
# Gravity

- Requires adding effect to source term of momentum conservation equation

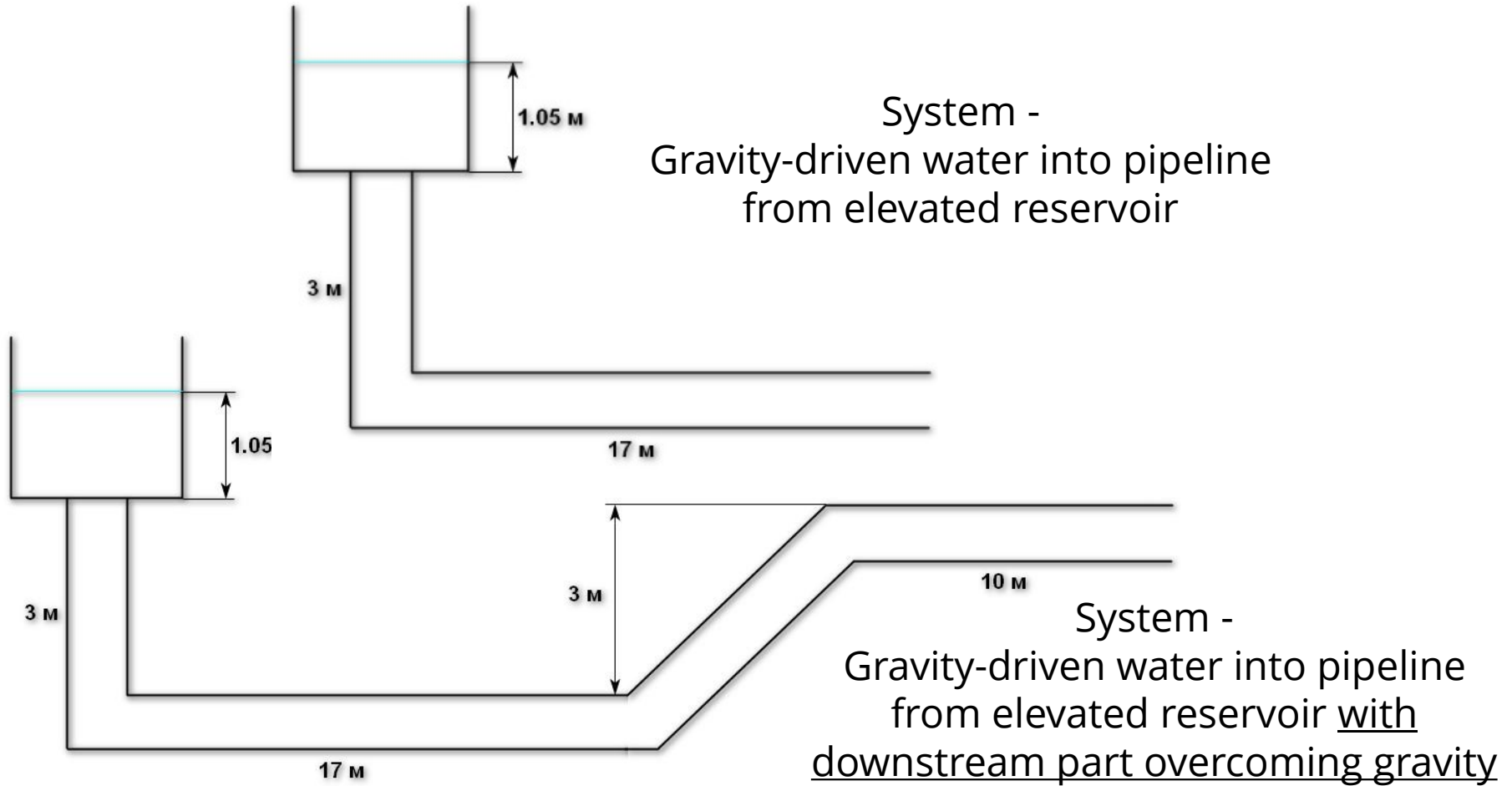
$$\frac{\partial}{\partial t} \int_V \rho \cdot \vec{u} \cdot dV = - \oint_S \rho \cdot \vec{u} \cdot \vec{u} \cdot d\vec{S} + s_{mom};$$

$$F = -m \cdot g \cdot \cos(\pi - \alpha)$$

- Orientation of channel vs. flow direction matters

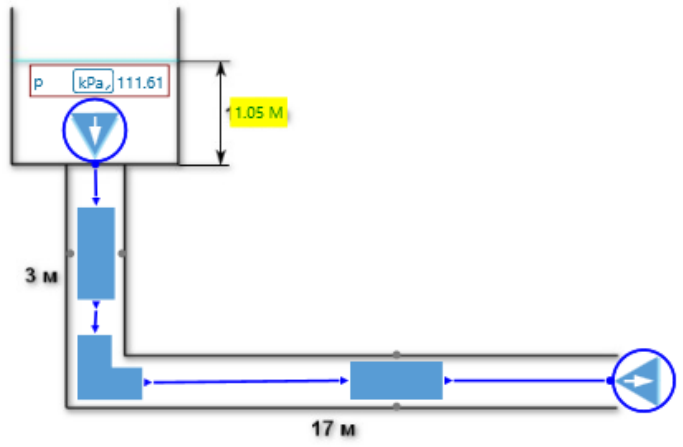


# Gravity Example - System



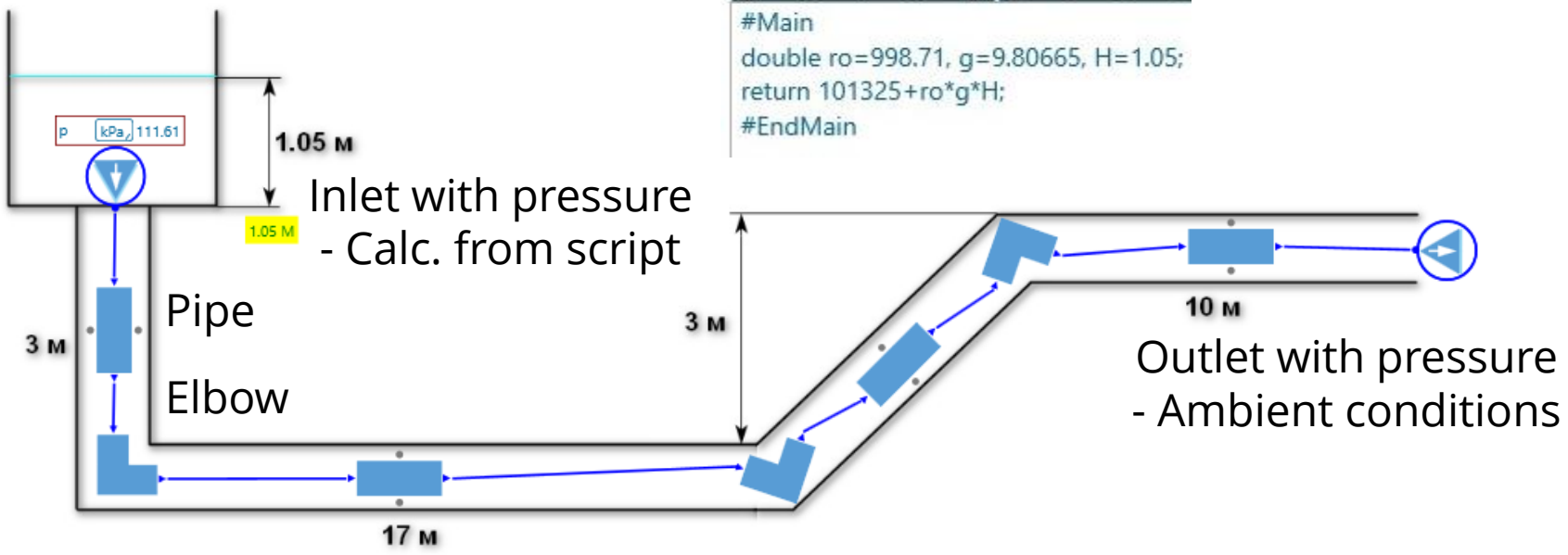
# Gravity Example - Results

| Case # | Elbows Angle | Mass Flow Rate |
|--------|--------------|----------------|
| Case 1 | N/A          | 3827 kg/h      |
| Case 2 | 45°          | 1394 kg/h      |
| Case 2 | 60°          | 1402 kg/h      |
| Case 2 | 90°          | 1389 kg/h      |



```

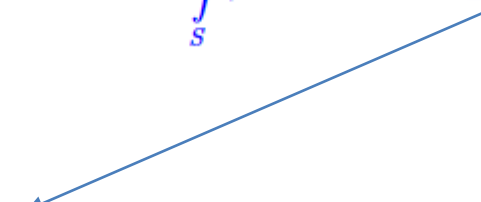
Script editor inletPressure.axsc
#Main
double ro=998.71, g=9.80665, H=1.05;
return 101325+ro*g*H;
#EndMain
    
```



# Centrifugal Force

- Inertial force appearing to act on all objects when viewed in a rotating frame of reference
- Requires adding effect to source term of momentum conservation equation

$$\frac{\partial}{\partial t} \int_V \rho \cdot \vec{u} \cdot dV = - \oint_S \rho \cdot \vec{u} \cdot \vec{u} \cdot d\vec{S} + s_{mom};$$


$$F = 0.5 \cdot \rho \cdot A \cdot \omega^2 \cdot (R_2^2 - R_1^2)$$

$F$  – centrifugal force, source term ( $s_{mom}$ ) in the momentum conservation equation;

$\rho$  – fluid density;

$A$  – cross section area of the channel;

$\omega$  – angular velocity;

$R_1$  – inlet radial distance;

$R_2$  – outlet radial distance.

# Centrifugal Force - Example

1000 rpm

5000 rpm

Flow direction ↓

|       |                   |         |
|-------|-------------------|---------|
| Angle | deg               | 45.000  |
| Rd1   | mm                | 200.00  |
| Rd2   | mm                | 100.00  |
| ua    | m/s               | 317.448 |
| rho   | kg/m <sup>3</sup> | 1.40    |
| uf    | m/s               | 317.275 |
| G     | kg/s              | 0.8704  |
| ut    | m/s               | 10.472  |
| p0    | kPa               | 190.28  |

Centripetal flow

|       |                   |         |
|-------|-------------------|---------|
| Angle | deg               | 45.000  |
| Rd1   | mm                | 200.00  |
| Rd2   | mm                | 100.00  |
| ua    | m/s               | 315.676 |
| rho   | kg/m <sup>3</sup> | 1.37    |
| uf    | m/s               | 311.303 |
| G     | kg/s              | 0.8348  |
| ut    | m/s               | 52.360  |
| p0    | kPa               | 186.24  |

Flow direction ↑

|       |                   |         |
|-------|-------------------|---------|
| Angle | deg               | 135.000 |
| Rd1   | mm                | 100.00  |
| Rd2   | mm                | 200.00  |
| ua    | m/s               | 317.879 |
| rho   | kg/m <sup>3</sup> | 1.40    |
| uf    | m/s               | 317.189 |
| G     | kg/s              | 0.8734  |
| ut    | m/s               | 20.944  |
| p0    | kPa               | 191.24  |

Centrifugal flow

|       |                   |         |
|-------|-------------------|---------|
| Angle | deg               | 135.000 |
| Rd1   | mm                | 100.00  |
| Rd2   | mm                | 200.00  |
| ua    | m/s               | 333.372 |
| rho   | kg/m <sup>3</sup> | 1.47    |
| uf    | m/s               | 316.498 |
| G     | kg/s              | 0.9105  |
| ut    | m/s               | 104.720 |
| p0    | kPa               | 211.40  |

Centrifugal effect in upward channel increases:

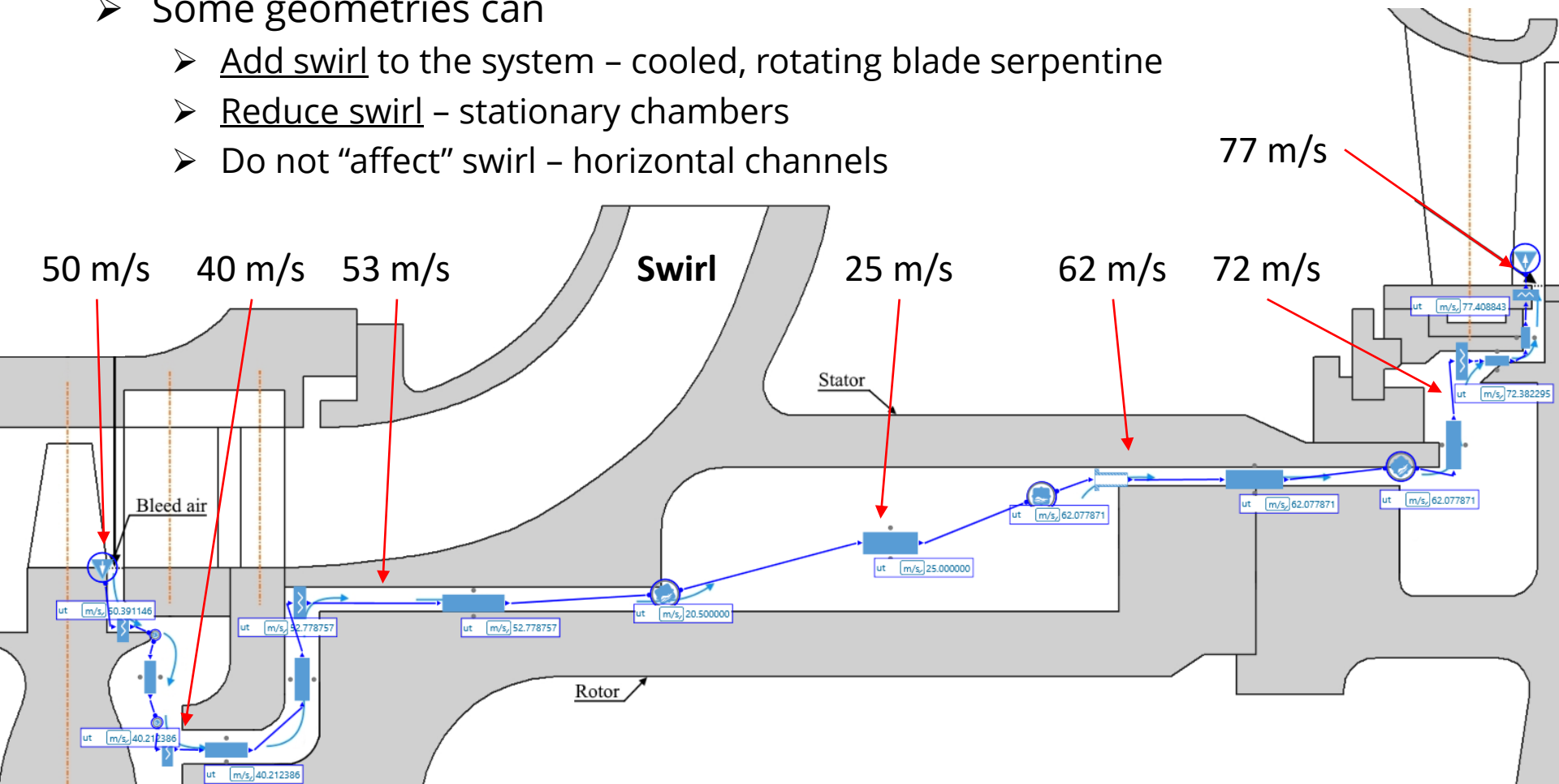
- Density
- Mass flow rate
- Tangential velocity
- Total pressure

The higher the rotation speed, the bigger the effect -  $f(\omega^2)$

Example - Cooled blade channels

# Swirl

- Amount of local tangential velocity in the system
- Some geometries can
  - Add swirl to the system – cooled, rotating blade serpentine
  - Reduce swirl – stationary chambers
  - Do not “affect” swirl – horizontal channels

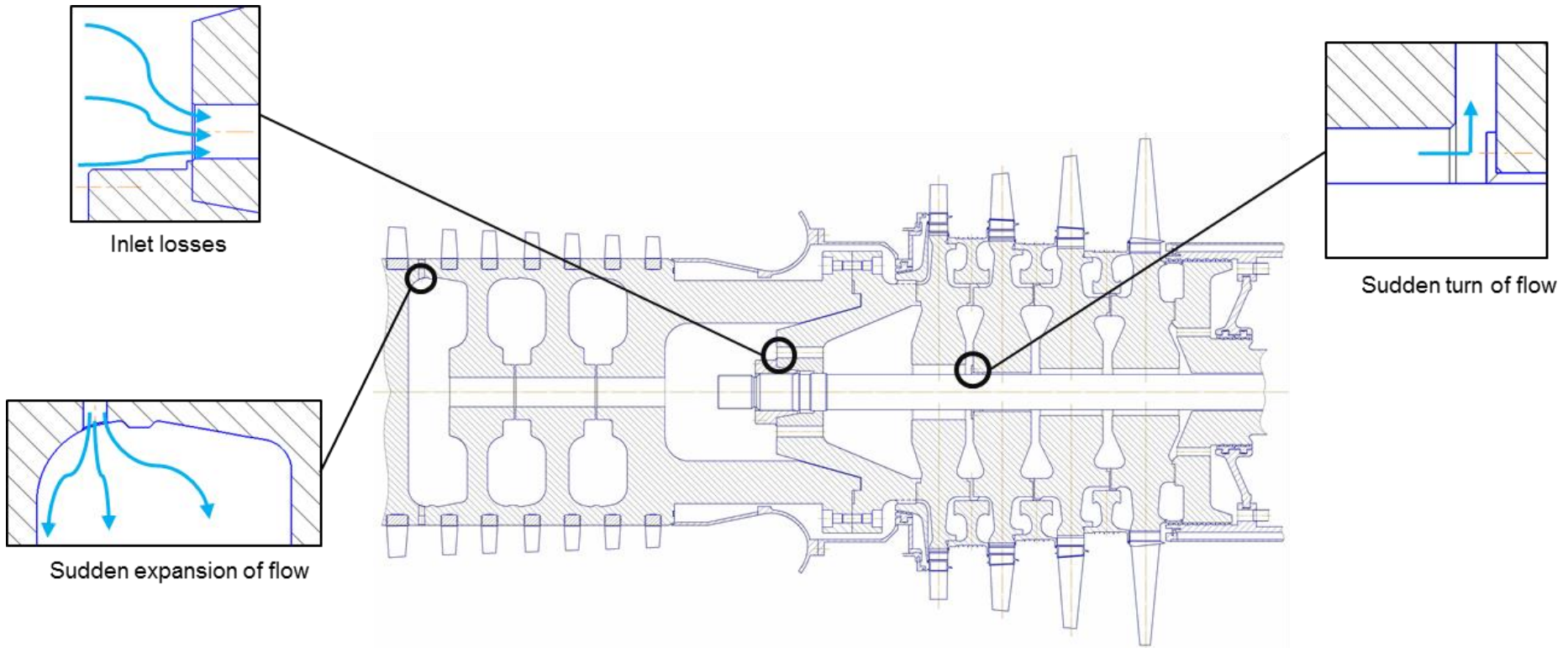


# Losses in Turbomachinery Systems

# Losses

- Various types and sources of losses present in secondary flows:
- Hydraulic losses
  - Inlets/contractions
  - Expansions
  - Bends
  - Flow splitting and mixing
- Friction losses
  - Disk windage
- Aerodynamic losses
  - Mixing losses of main turbine flow and cooling flows
  - Correct incoming swirl from injections
- Pumping losses

# Hydraulic Losses in Secondary Flow Path

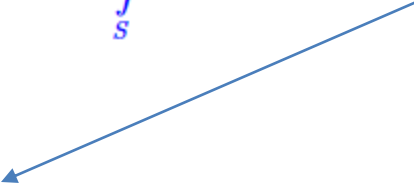


- Hydraulic losses include: inlet/outlet, expansions/contractions, orifices, etc.
- Allow determining accurate cooling flow parameters for component/system performances.

# Hydraulic Resistance Force

- Force acting in opposite direction of flow motion and corresponding to a flow restriction having multiple possible sources:
  - Surface roughness
  - Cross-section reduction
  - Etc.
- Requires adding effect to source term of momentum conservation equation

$$\frac{\partial}{\partial t} \int_V \rho \cdot \vec{u} \cdot dV = - \oint_S \rho \cdot \vec{u} \cdot \vec{u} \bullet d\vec{S} + s_{mom};$$


$$F_r = 0.5 \cdot \zeta \cdot \rho \cdot u^2 \cdot A$$

$\zeta$  – resistance coefficient;

$\rho$  – fluid density;

$u$  – fluid flow velocity;

$A$  – cross section area of the channel.

# Hydraulic Resistance

- One of the parameters from the hydraulic resistance force equation

$$\zeta = \lambda \cdot \frac{l}{D_h}$$

$\lambda$  – Darcy friction factor;

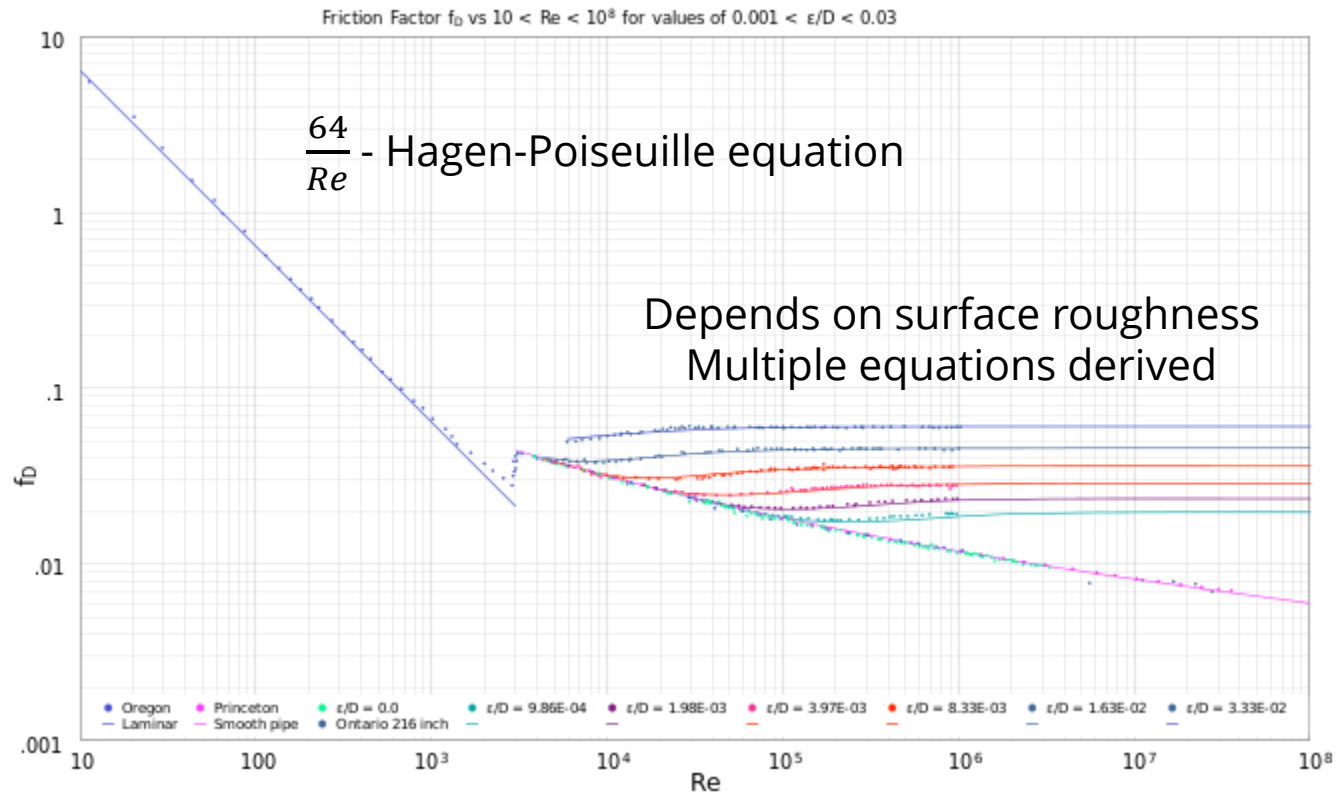
$l$  – channel length;

$D_h$  – hydraulic diameter of the channel.

- Increase in hydraulic resistance due to:
  - Smaller cross-sections
  - Longer channels
  - Higher friction per unit length (see next slide)
- Results in pressure losses as a function of square of flow velocity

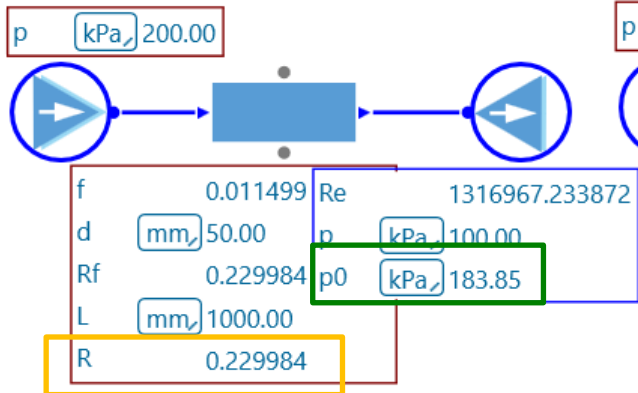
# Darcy Friction Factor

- Non-constant, dimensionless quantity used in Darcy-Weisbach eq. for calculation of friction losses in pipe flows
  - Depends on characteristics of pipe, fluid, flow regime and flow velocity (Moody diagram)

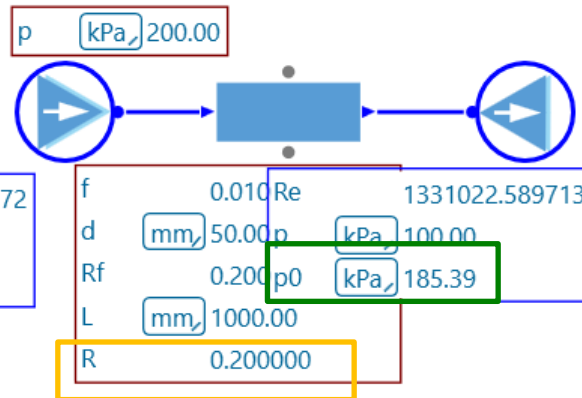


# Resistance Coefficient Examples - Roughness

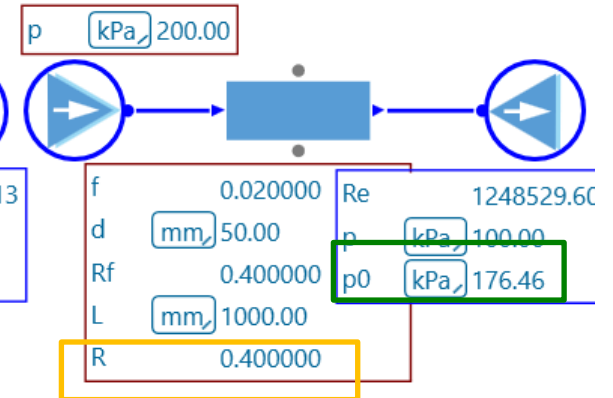
Smooth wall



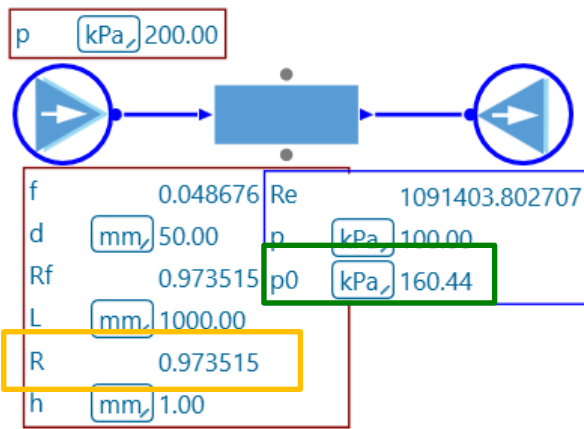
Rough wall - Darcy friction factor = 0.01



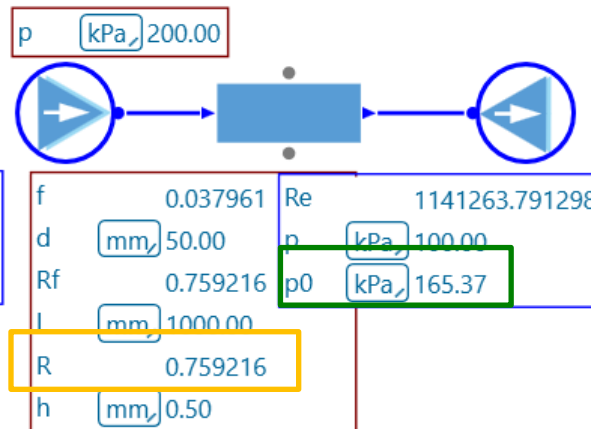
Rough wall - Darcy friction factor = 0.02



Rough wall - Surface roughness = 1 mm



Rough wall - Surface roughness = 0.5 mm



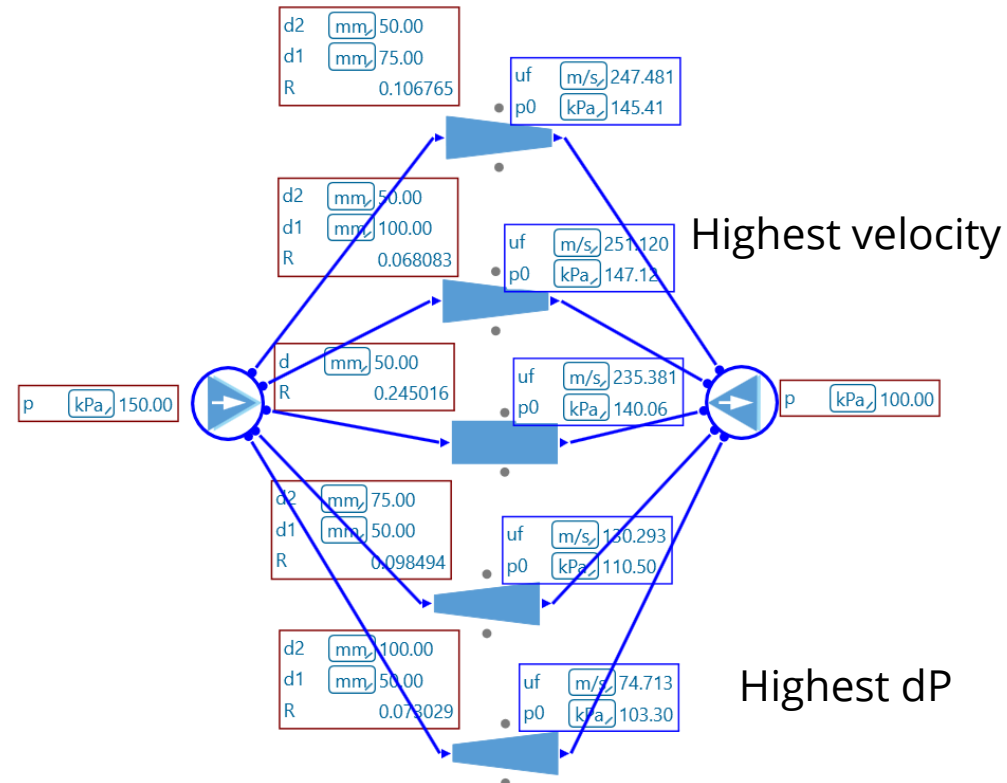
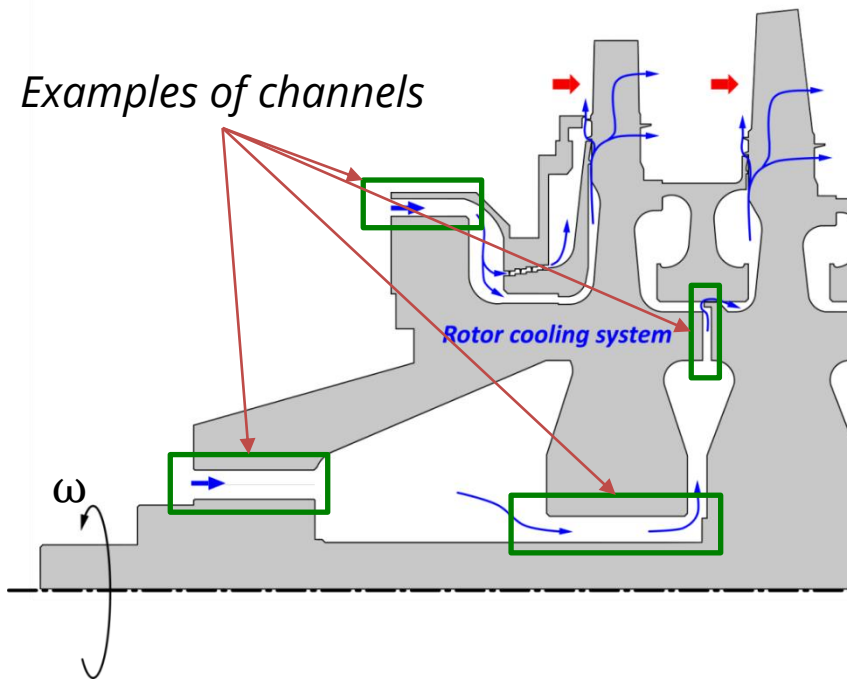
# Straight vs. Converging vs. Diverging Channel

- Pressure drop in channels

$$\Delta p = \zeta * \frac{\rho u^2}{2}$$

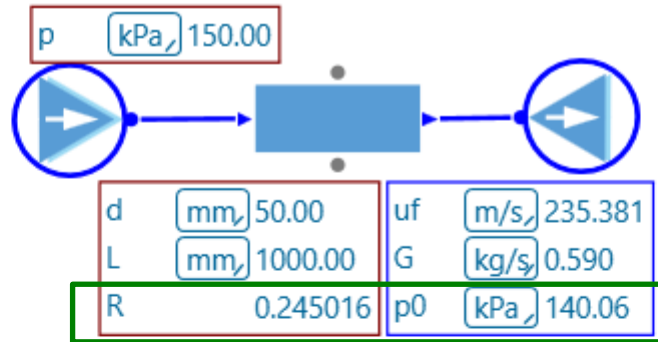
- $u^2$  – mean flow velocity between inlet and outlet flows

Examples of channels

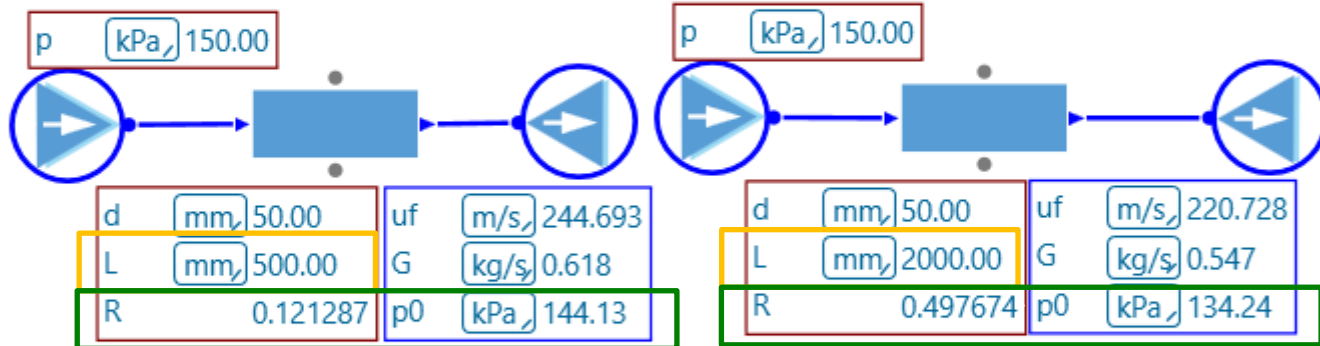


# Resistance Coefficient Examples - Geometry

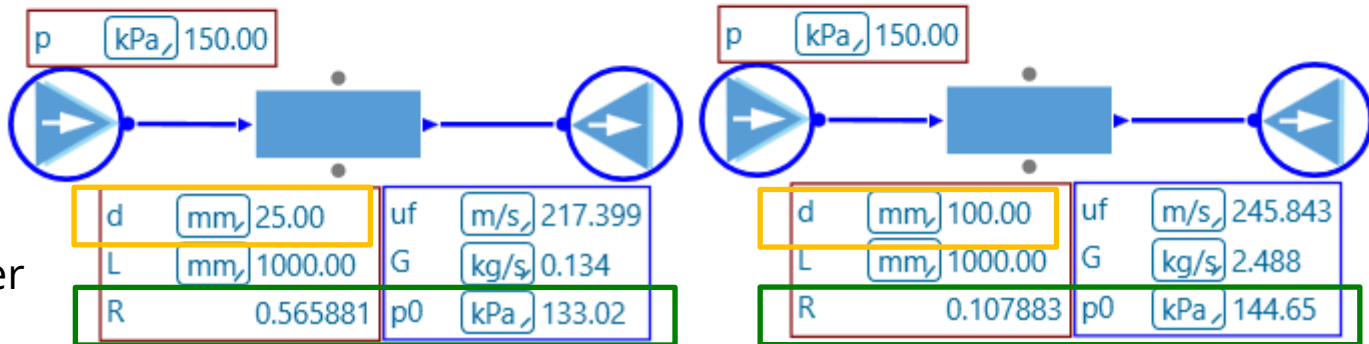
Benchmark



Diff. length



Diff. diameter



# Friction in Non-Circular Pipes

- Darcy friction factor established for circular pipes and corrected for non-circular ones

$$\lambda' = k \cdot \lambda;$$

$\lambda'$  – Darcy friction factor for non circular straight channels;

$\lambda$  – Darcy friction factor for straight circular channels at the same Reynolds number  $Re$ ;

$k$  – Darcy correction factor that takes into account the effect of the channel cross section shape.

- $\lambda' =$

- For square cross sections for laminar (0.89) vs. turbulent (1)
- Non-constant for other shapes
- Example - Depends on the aspect ratio for rectangular cross-sections (the more square the better)

| $\frac{b_0}{a_0}$                   | 0   | 0.1  | 0.2 | 0.4  | 0.6  | 0.8 | 1.0  |
|-------------------------------------|-----|------|-----|------|------|-----|------|
| $k = f\left(\frac{b_0}{a_0}\right)$ | 1.5 | 1.34 | 1.2 | 1.02 | 0.94 | 0.9 | 0.89 |

| $\frac{b_0}{a_0}$                   | 0   | 0.1  | 0.2  | 0.4  | 0.6  | 0.8  | 1.0 |
|-------------------------------------|-----|------|------|------|------|------|-----|
| $k = f\left(\frac{b_0}{a_0}\right)$ | 1.1 | 1.08 | 1.06 | 1.04 | 1.02 | 1.01 | 1.0 |

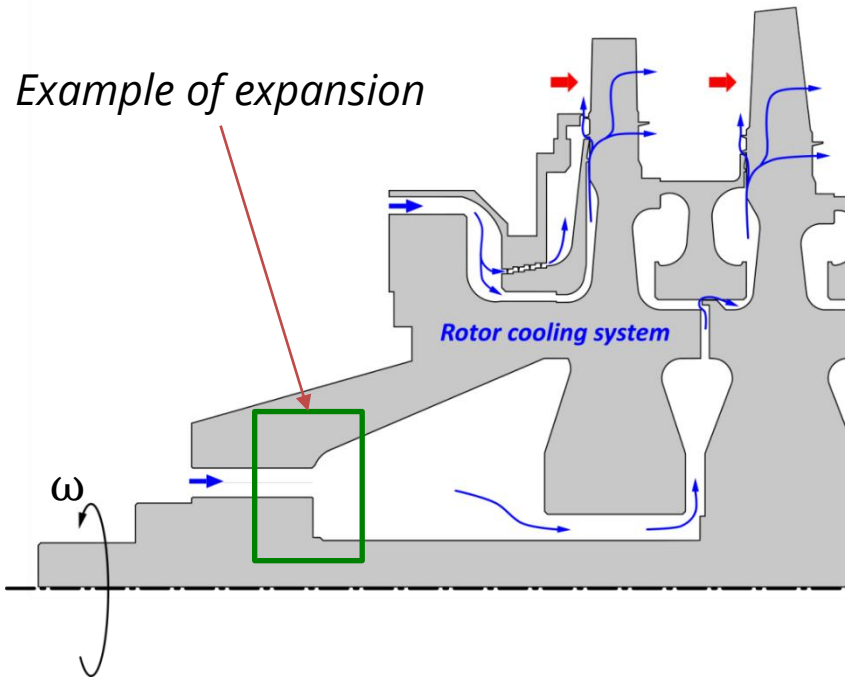
# Notes on Friction for sCO<sub>2</sub>

- Supercritical fluids by definition behave differently from the other phases.
- One major aspect is the viscosity and therefore friction
- Darcy friction factor considered laminar  $\frac{64}{Re}$  for typical fluids for  $Re < 2000$  while it holds true for  $Re < 2300$  for sCO<sub>2</sub> in the laminar regime
- Transition regime is  $2300 < Re < 3400$  for sCO<sub>2</sub> vs.  $2000 < Re < 4000$

# Sudden Expansions

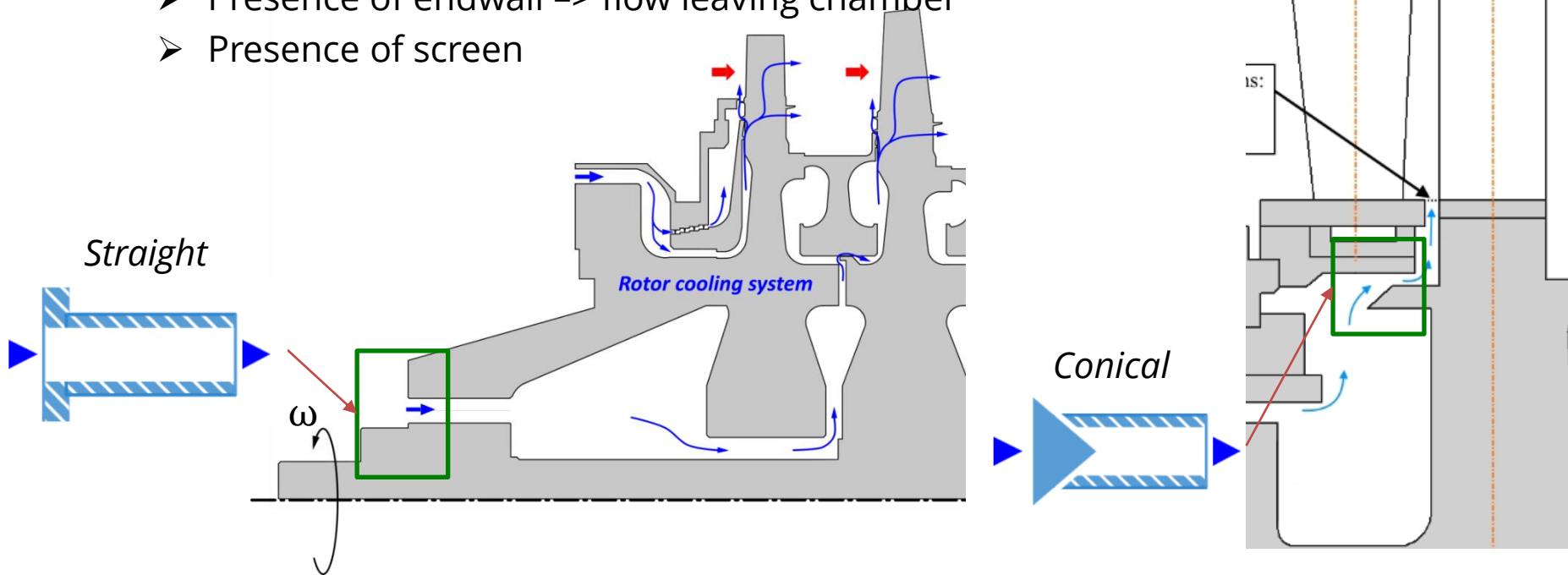
- Typically modeled differently depending on geometry of upstream channel
  - Short vs. long channel – length/diameter < or > than 10
  - Resistance coefficient based on Re

*Example of expansion*



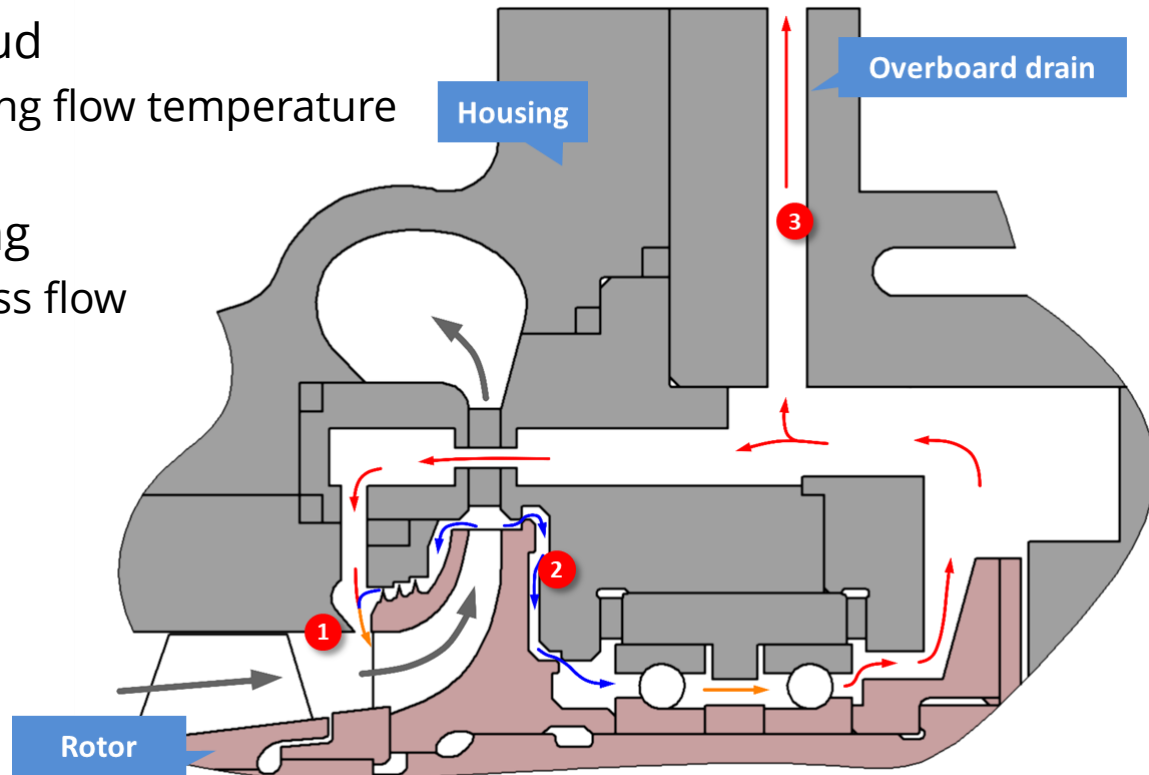
# Fluid Entrances

- Model flow transition from chamber to channel
- Different resistances depending on Re and shape of channel transition
  - Straight
  - Rounded
  - Conical
  - Presence of endwall → flow leaving chamber
  - Presence of screen



# Turbopumps & Turbochargers Example

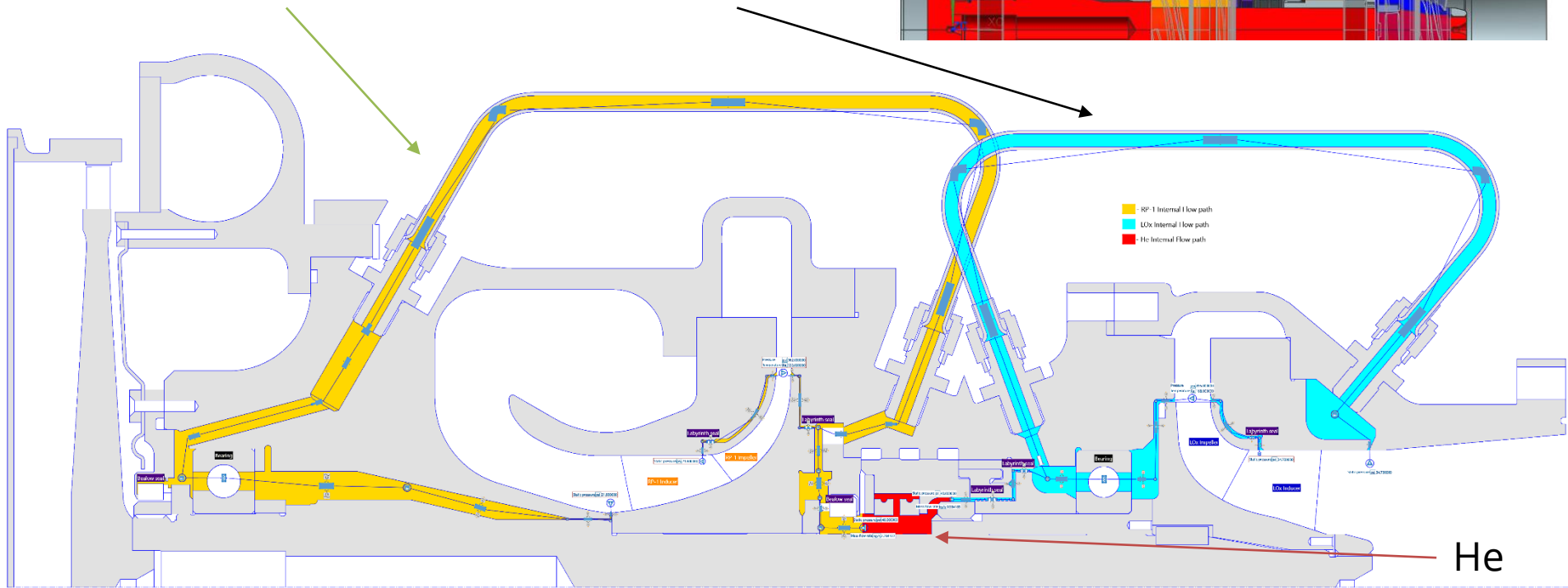
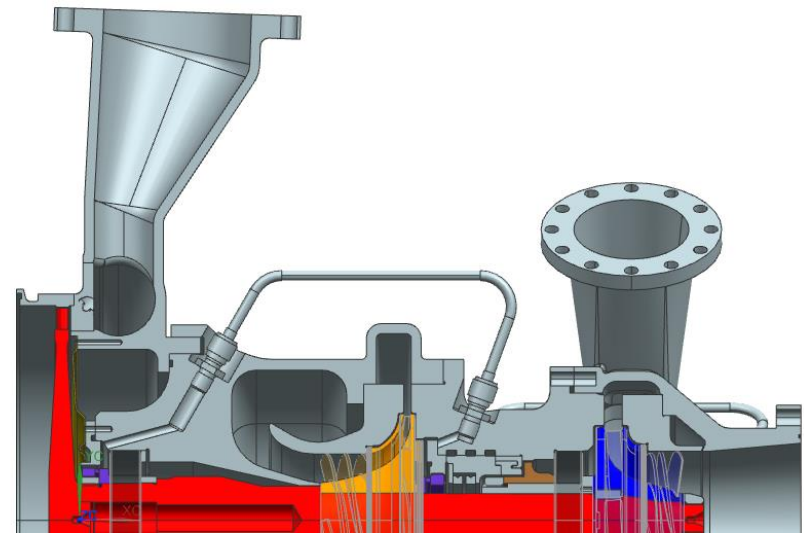
- Modeling of leakages on backface of impellers
  - Determination of pressure in balance piston cavity
  - Amount of leakage determines overall machine efficiency
- Modeling of ported shroud
  - Estimation of recirculating flow temperature
- Modeling of system piping
  - Calculation of bleed mass flow



# Turbopump Example

RP1 internal flow path

LOx

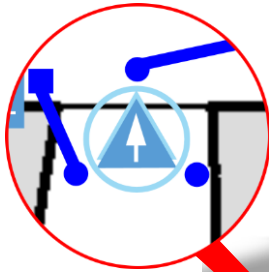


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# Turbopump Example

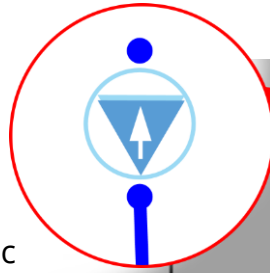
## **Inlet**

Allows specifying inlet boundary conditions, such as total pressure and total temperature



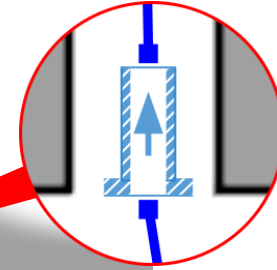
## **Outlet**

Allows determining outlet boundary conditions, such as static pressure



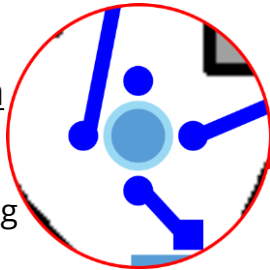
## **Entrance**

One of the predefined flow resistances, such as orifices, bends, seals, etc.



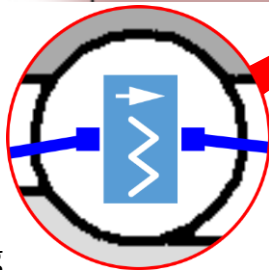
## **Junction**

Allows modeling flow mixing and separation



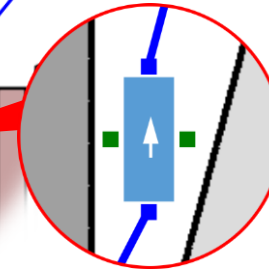
## **Resistance**

Allows specifying custom flow resistance using scripting



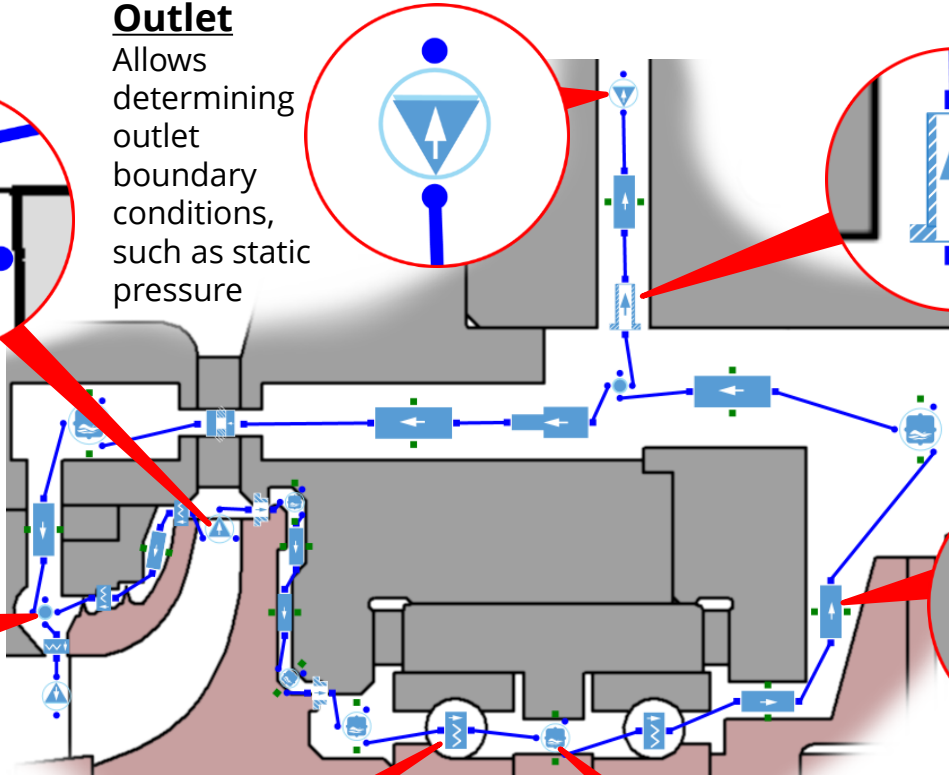
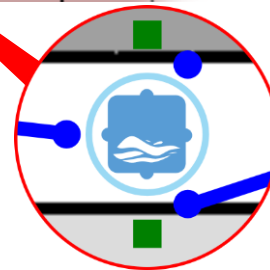
## **Pipe**

Allows simulating flow in pipes and channels taking into account friction and centrifugal forces

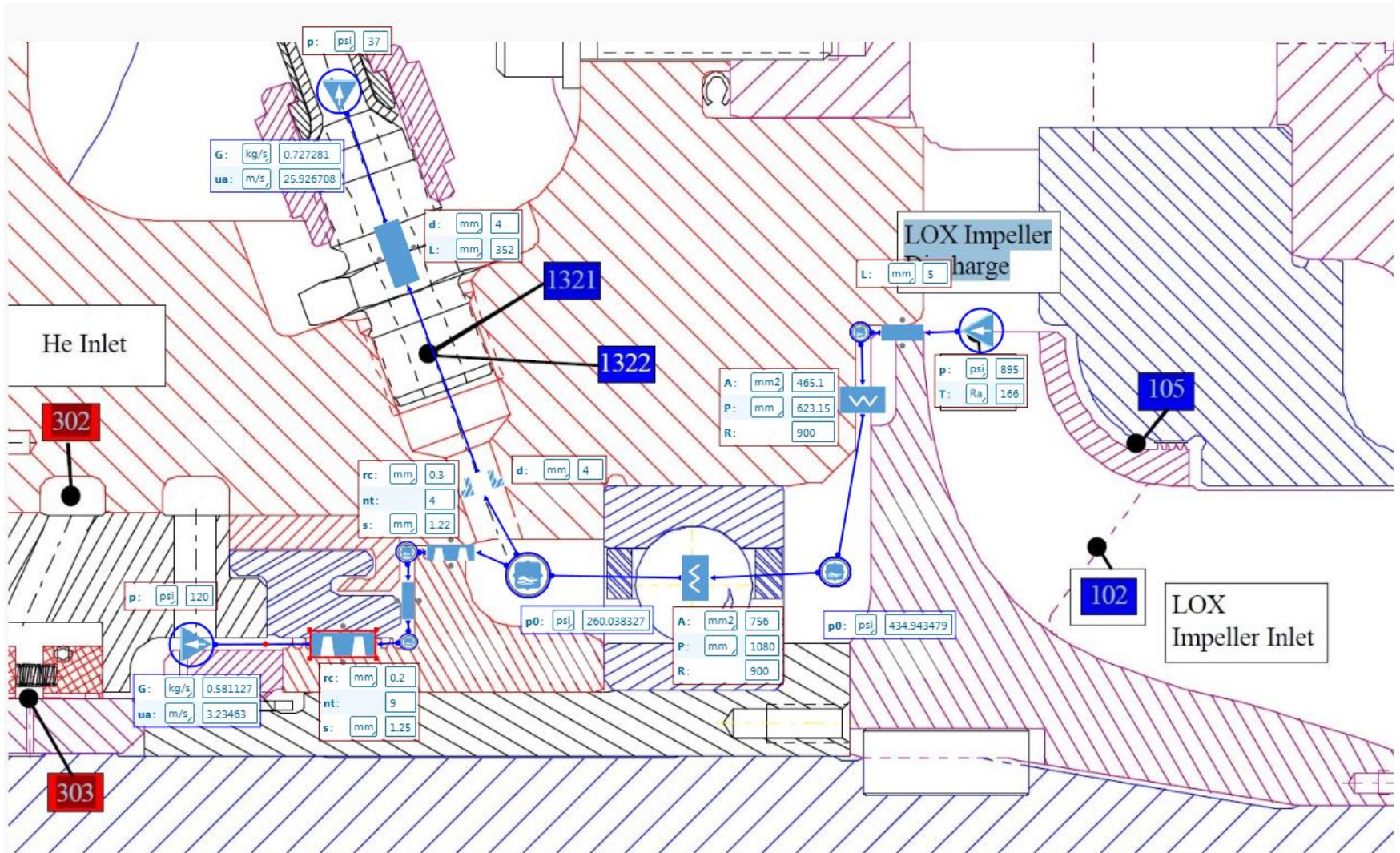


## **Chamber**

Allows representing chambers and cavities with rotated walls and specify heat source

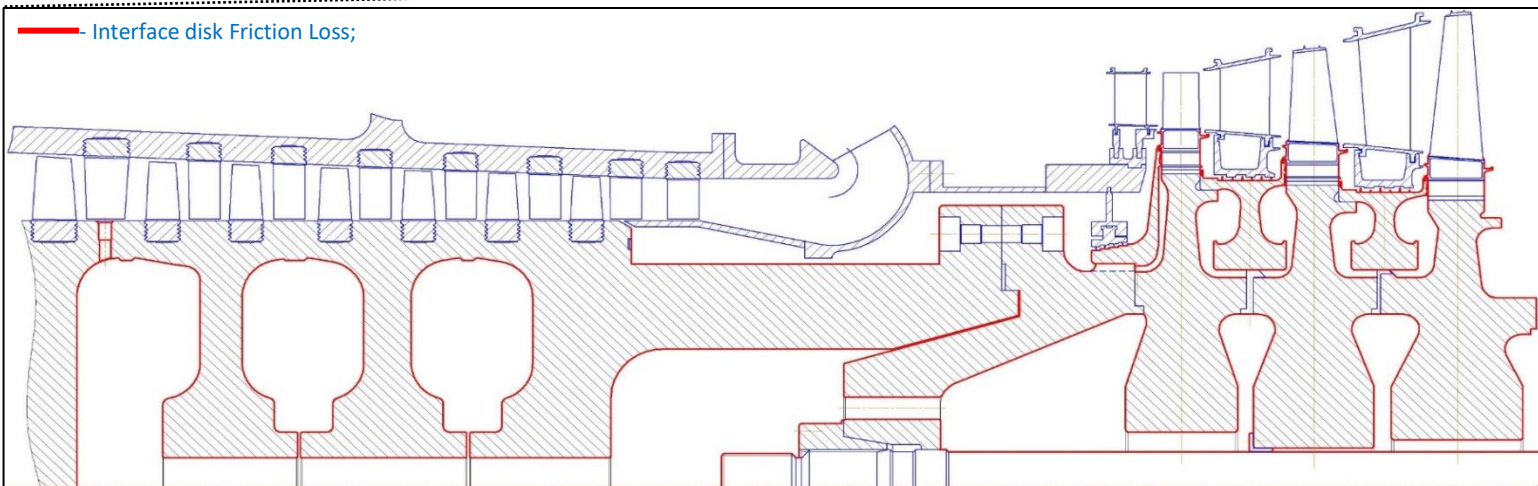
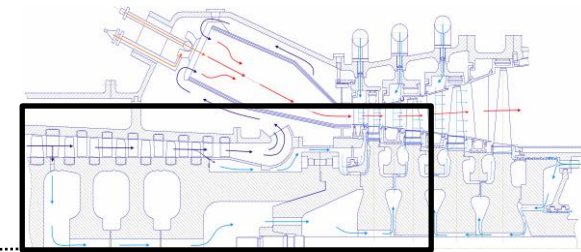


# Turbopump Example



# Disk Friction Loss

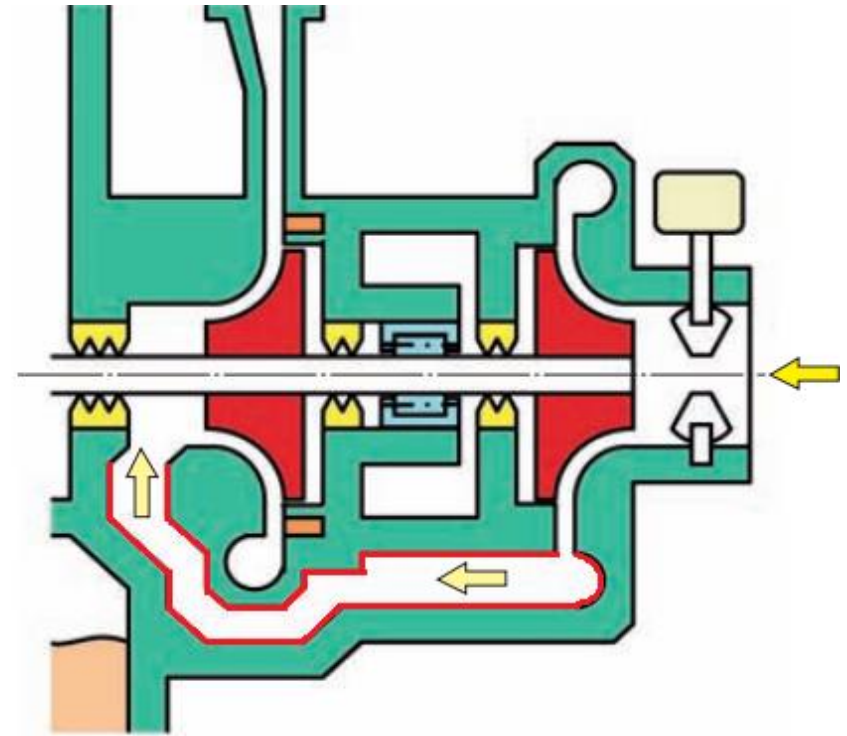
- Thermal loss due to friction between a rotating disk surface and a gas that should be overcome by consuming additional power.
- $W = \beta_0 \cdot \rho \cdot \omega^3 \cdot D_2^5 \cdot (1 + 5 \frac{b_2}{D_2})$  where  $\omega$  – angular velocity rotor,  $\rho$  – fluid density,  $b$  – blade height,  $D$  - diameter;



# Case Study - Centrifugal Compressor with Inter-Stage Piping

# Problem Formulation

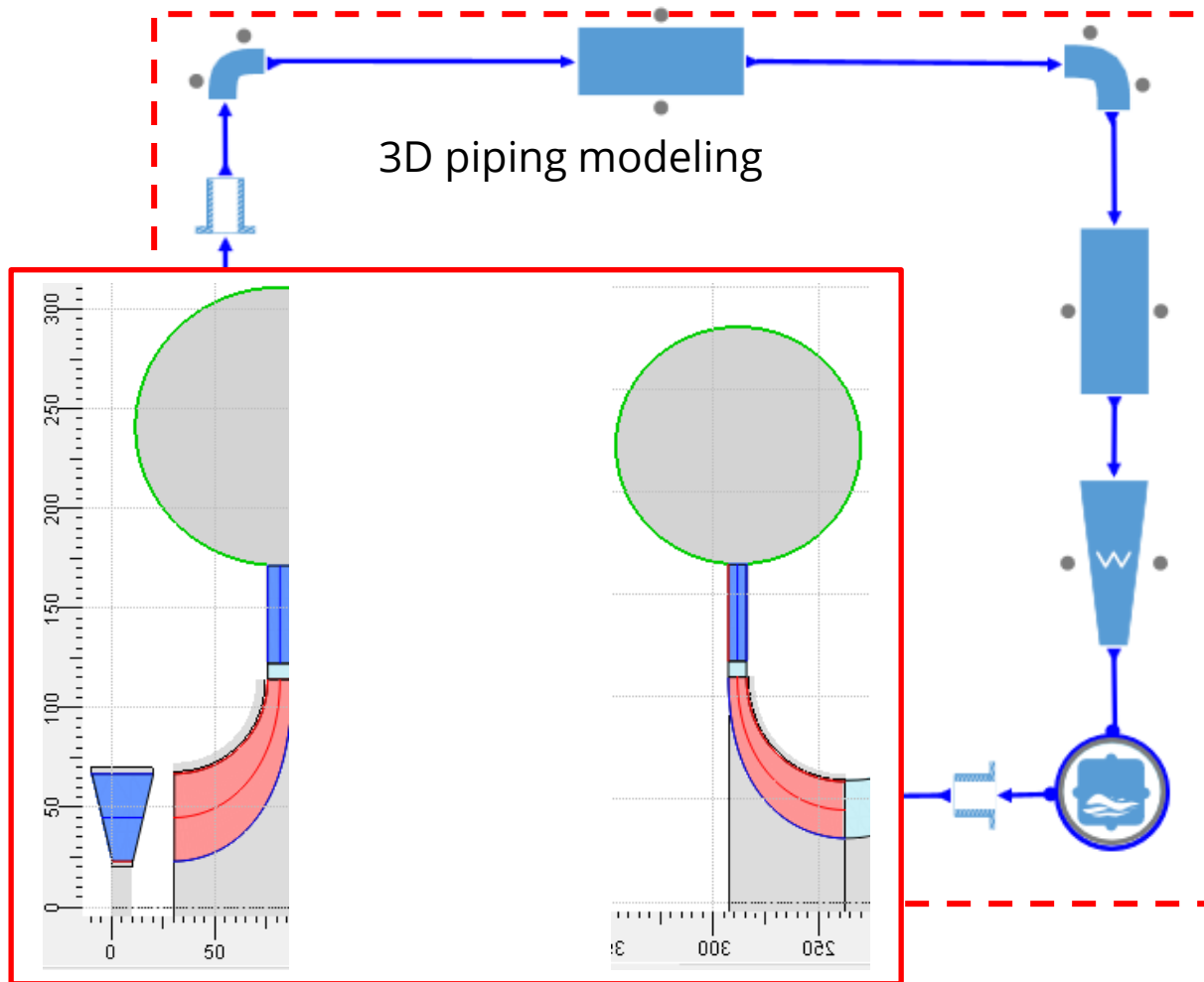
- Task
  - Model a 2-st centrifugal compressor with inter-stage 3-D pipe.
- Objective
  - Total pressure loss through the 3-D piping is properly accounted for during calculation.
  - Obtain performance map of the overall stage.
- Boundary Conditions (at Design Point)
  - Inlet Total Temperature: 20 °C
  - Inlet Total Pressure: 100 kPa
  - MFR: 84 kg/min
  - Rotational Speed: 29600 rpm



*Example of Inter-stage pipe configuration  
[Hideaki et al., 2009]*

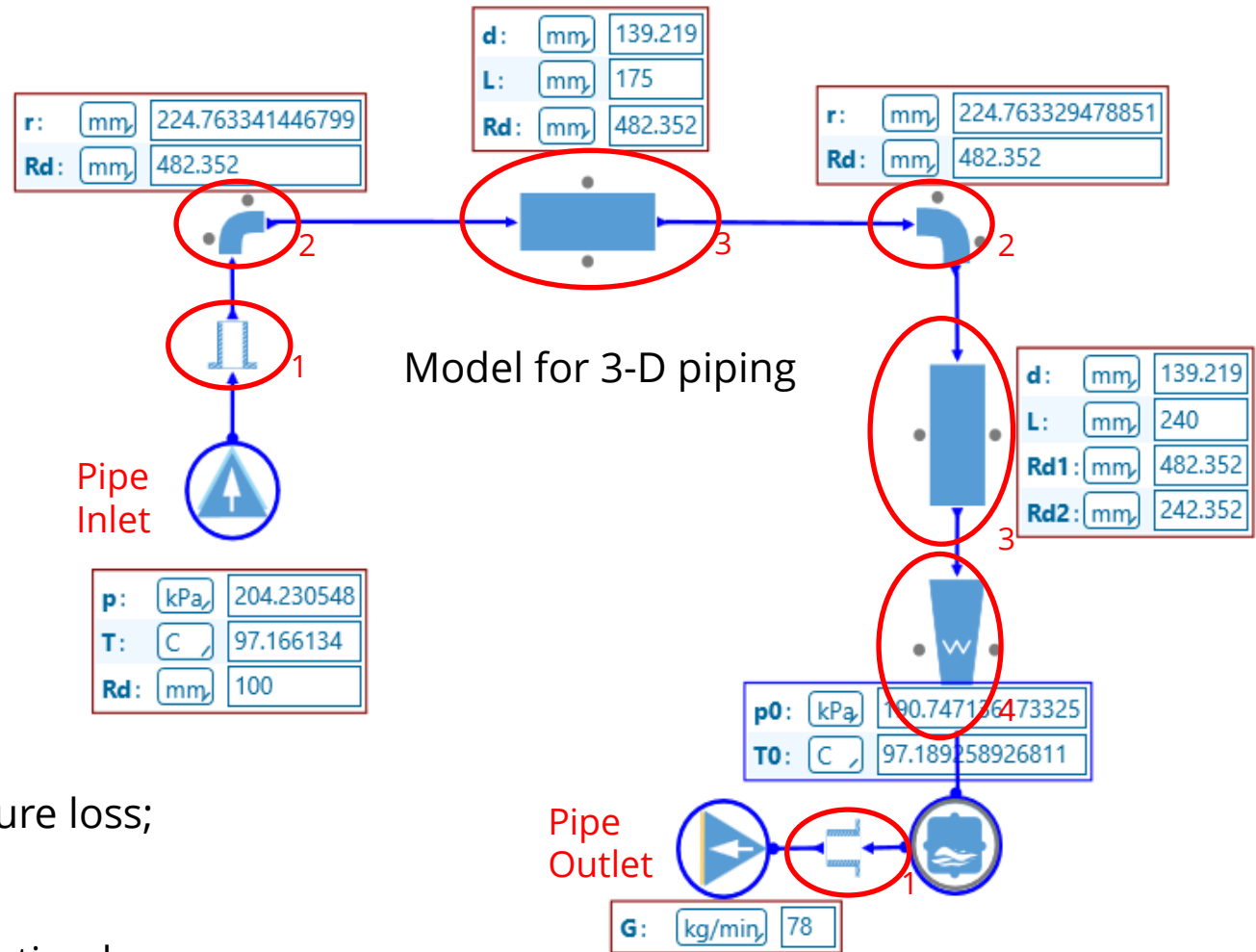
Note: the 3-D pipe connects the outlet of 1<sup>st</sup> stage volute and 2<sup>nd</sup> stage inlet.

# Schematics



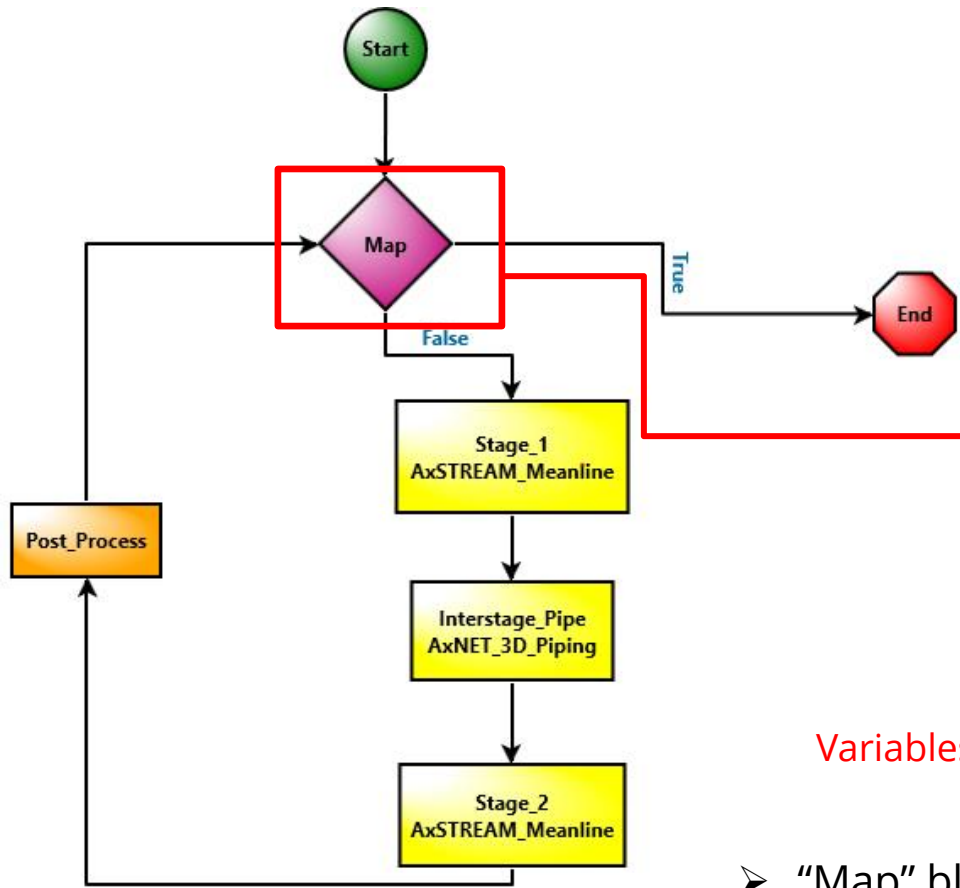
Compressors flow path

# 3D Piping Modeling



1. inlet/outlet loss;
2. flow path curvature loss;
3. duct flow loss;
4. flow path contraction loss.

# Integration of Multiple Modules



Flow Chart of Integrated Platform

Design of Experiment | Adjust parameters | Calculation results | Map

Name:

| Name              | Description          |   |
|-------------------|----------------------|---|
| Pstat_loss_pipe   | Static Pressure Loss | ✗ |
| Prstat_stage      | Pstat ratio          | ✗ |
| Prt_stage         | New Description      | ✗ |
| Efficiency_stage  | New Description      | ✗ |
| Pt_loss_pipe      | New Description      | ✗ |
| Prstat_origin     | New Description      | ✗ |
| Prt_origin        | New Description      | ✗ |
| Efficiency_origin | New Description      | ✗ |

Performance data

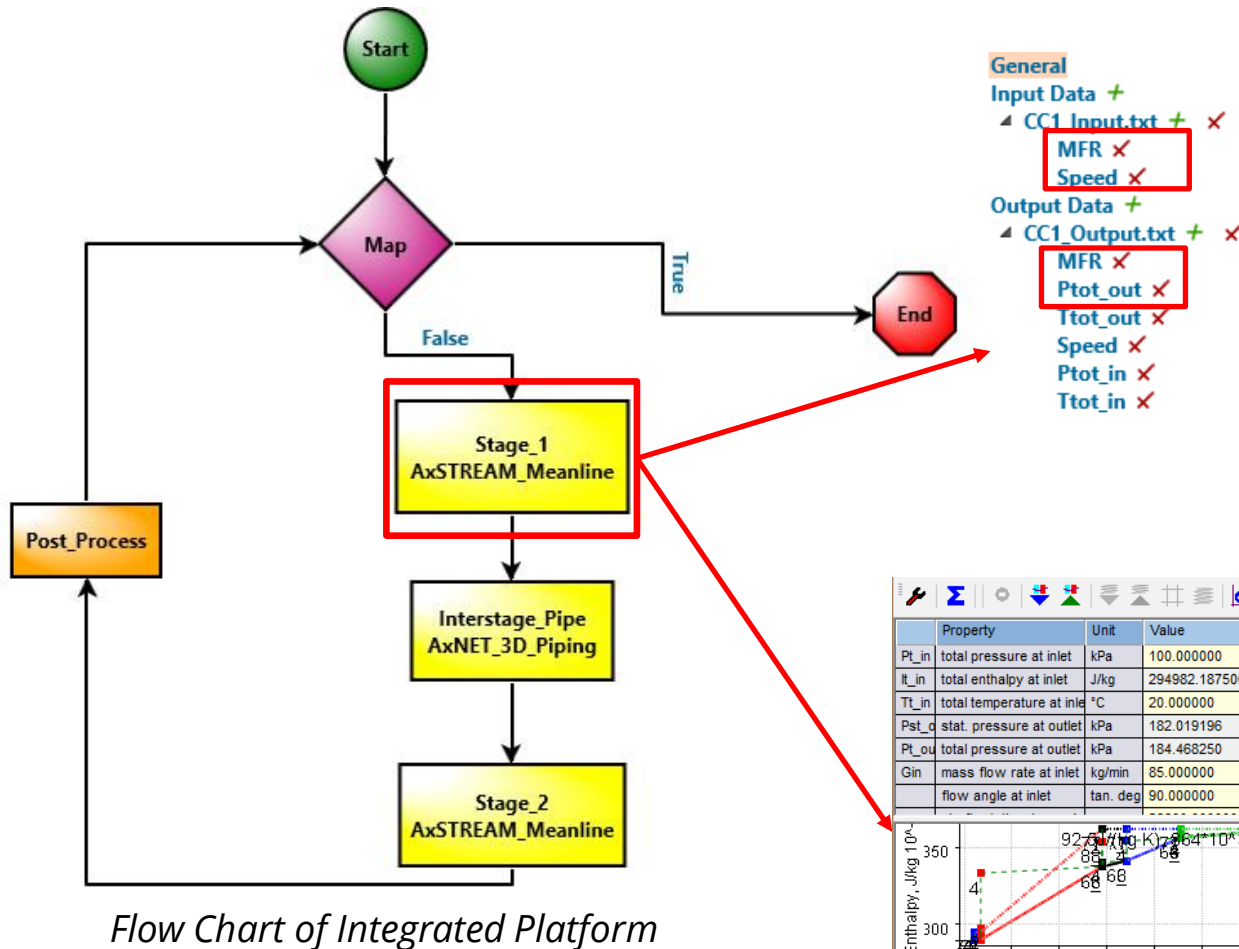
+

| Name         | Description     | Points Count | Major                               | Min                                 |
|--------------|-----------------|--------------|-------------------------------------|-------------------------------------|
| MFR (kg/min) | New Description | 11           | <input type="checkbox"/>            | <input checked="" type="checkbox"/> |
| speed        | New Description | 3            | <input checked="" type="checkbox"/> | <input type="checkbox"/>            |

Variables

- “Map” block assigns variables as inputs to the Process Blocks.
- Performance data will be returned to “Map” after calculation.

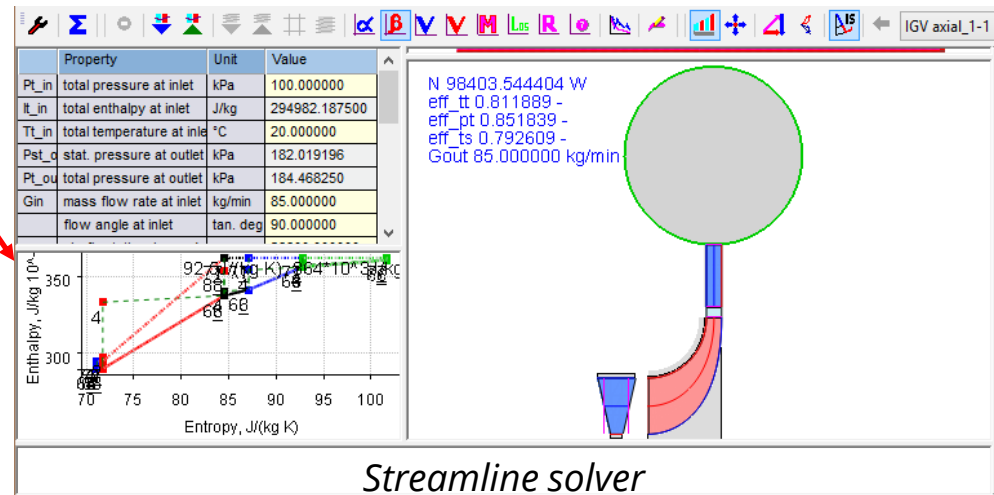
# Integration of Multiple Modules



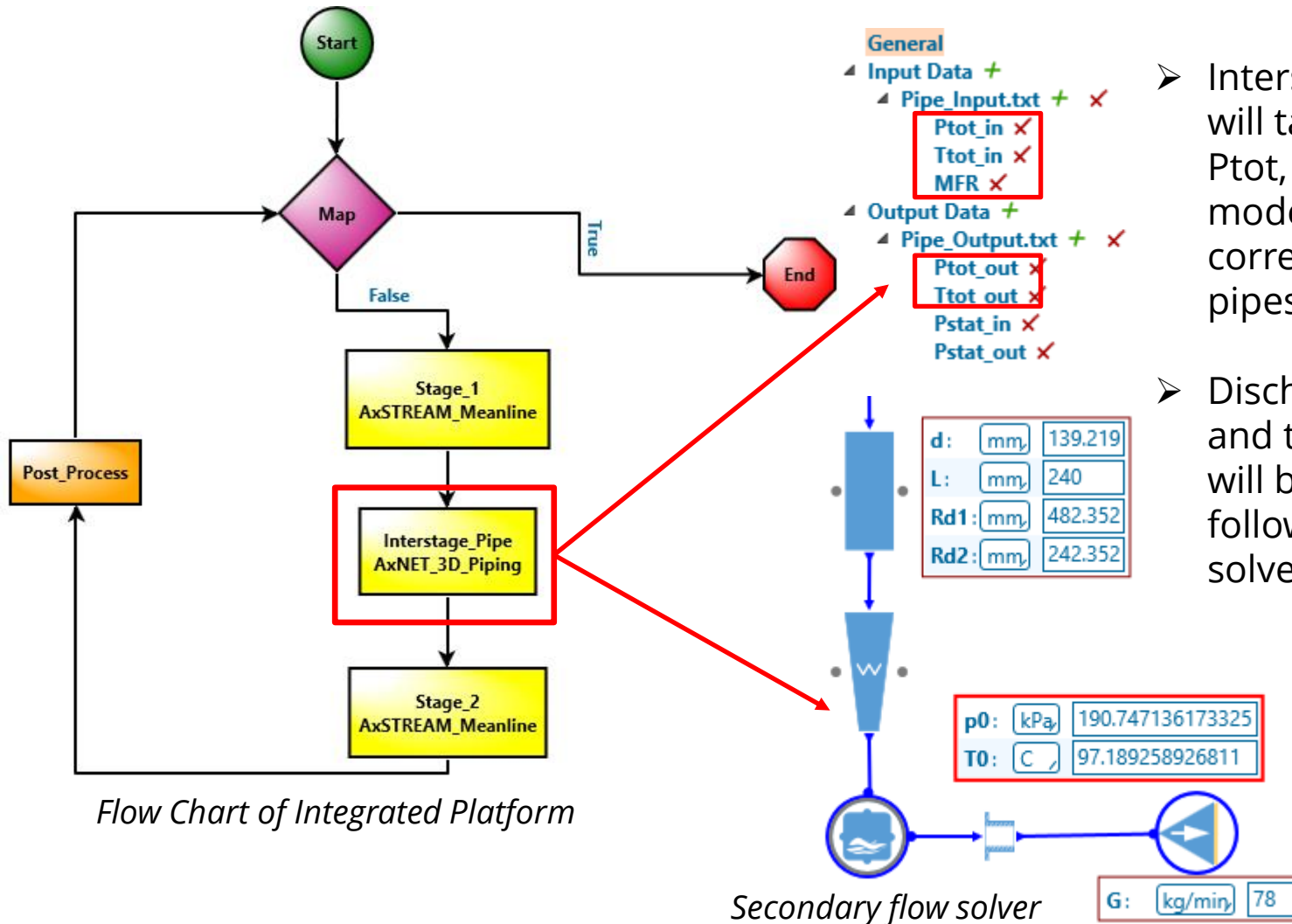
General  
 Input Data +  
 CC1 Input.txt +  
 MFR ✗  
 Speed ✗  
 Output Data +  
 CC1 Output.txt +  
 MFR ✗  
 Ptot\_out ✗  
 Ttot\_out ✗  
 Speed ✗  
 Ptot\_in ✗  
 Ttot\_in ✗

➤ “Compressor\_1” block will take in assigned MFR and speed with ambient inlet to run analysis in streamline solver for the 1<sup>st</sup> stage.

➤ Discharge total pressure and total temperature will be passed to secondary flow solver.



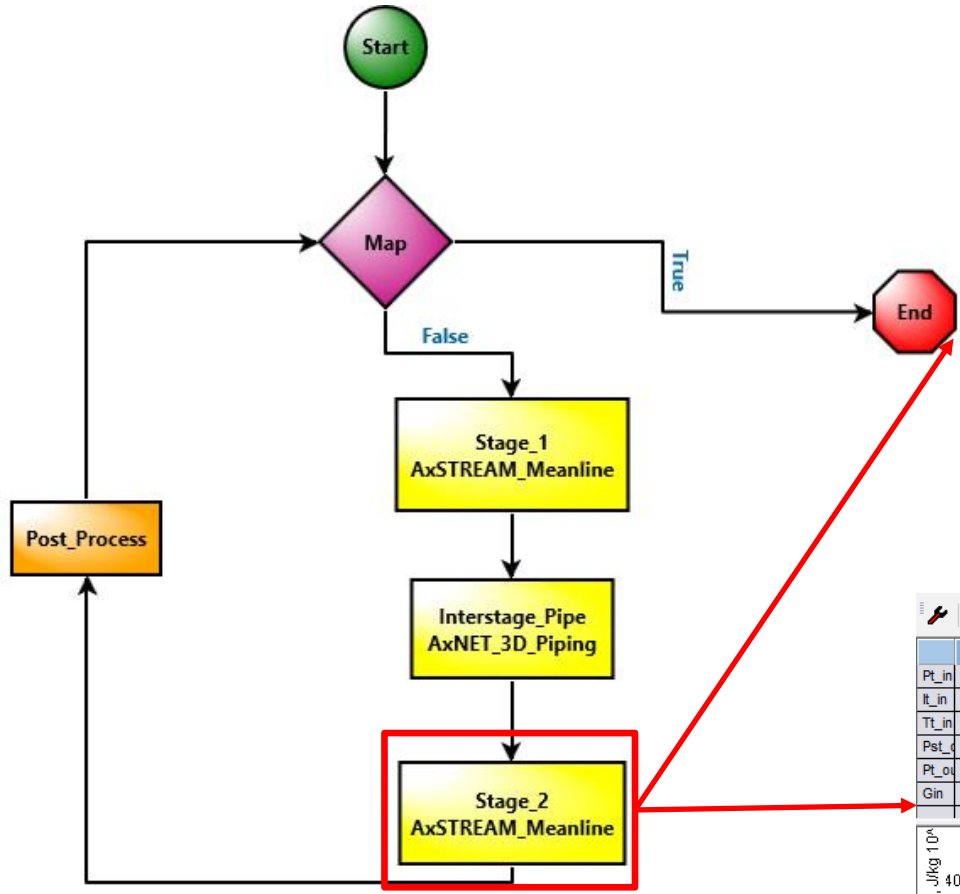
# Integration of Multiple Modules



- Interstage\_Pipe block will take in upstream Ptot, Ttot and MFR to model the corresponding 3-D pipes.
- Discharge total pressure and total temperature will be passed to following streamline solver.

# Integration of Multiple Modules

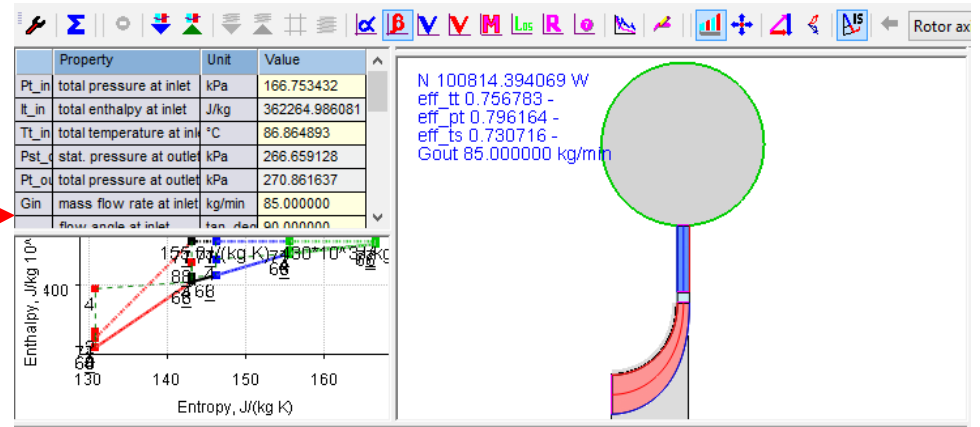
- Compressor\_2 block will take assigned MFR and speed with  $P_{tot}$  and  $T_{tot}$  from secondary flow tool to run analysis in streamline solver for the 2<sup>nd</sup> stage.
- Discharge total pressure and total temperature will be passed Post-Process to calculate performance.



General

- Input Data
  - CC2\_Input.txt
    - MFR ✗
    - Speed ✗
    - Ptot\_in ✗
    - Ttot\_in ✗
- Output Data
  - CC2\_Output.txt
    - Ptot\_out ✗
    - Ttot\_out ✗
    - Pstat\_out ✗
    - MFR ✗
    - Speed ✗

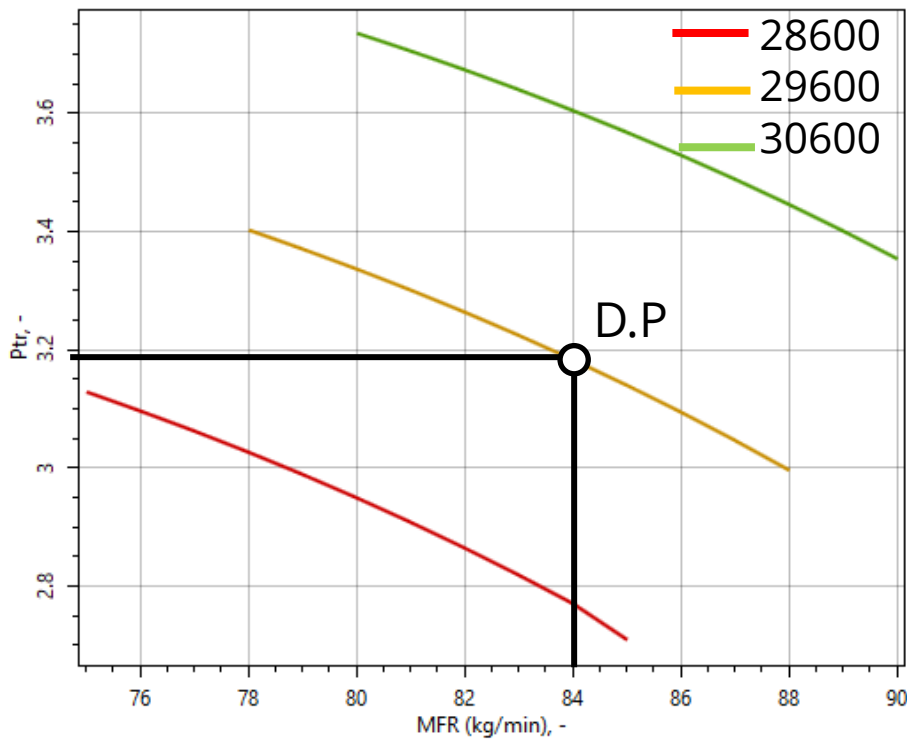
Flow Chart of Integrated Platform



Streamline solver

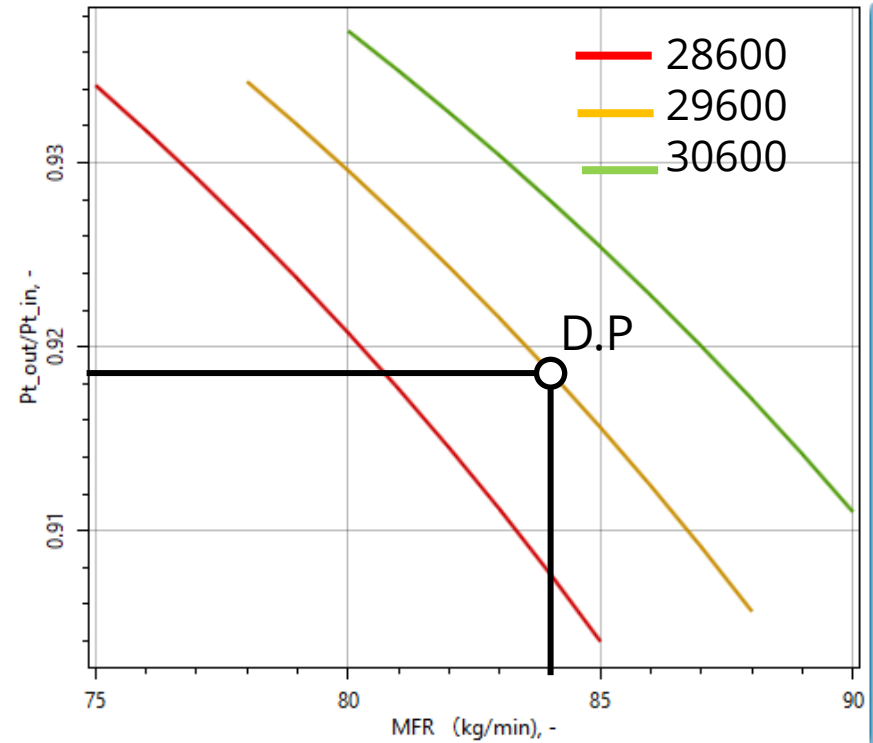
# Simulation Results

Design of Experiment | Adjust parameters | Calculation results | Map



Total Pressure Ratio vs. MFR

Design of Experiment | Adjust parameters | Calculation results | Map



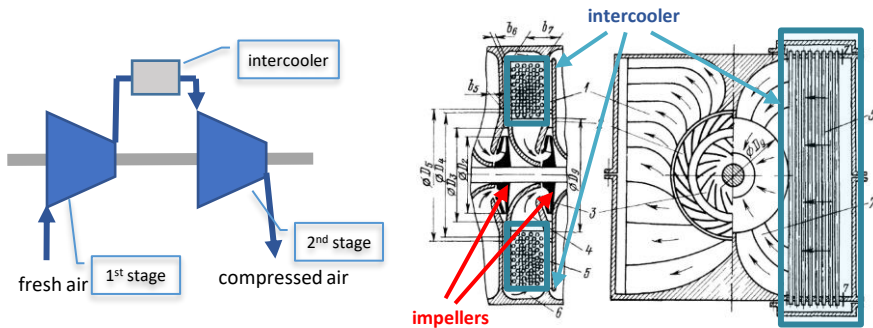
Pipe Pressure Loss vs. MFR

# Air-cooled Heat Exchanger

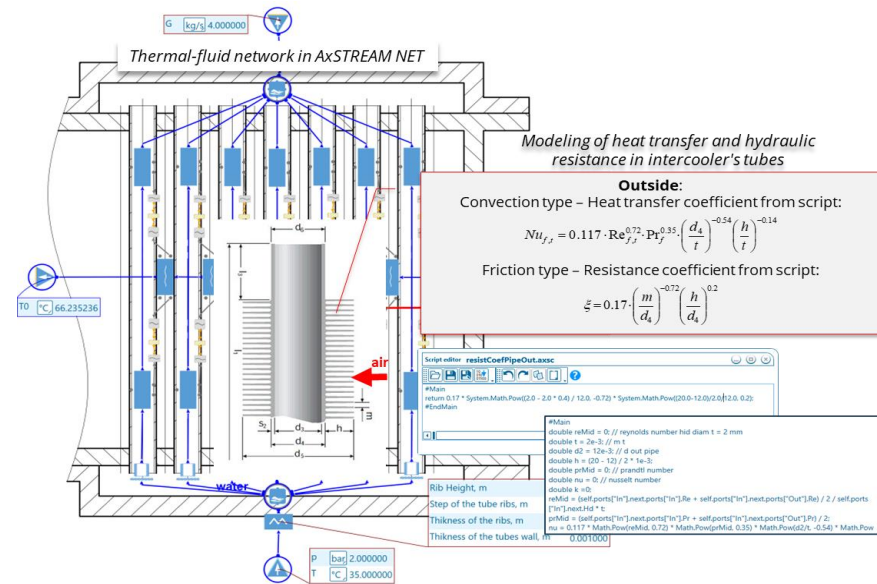
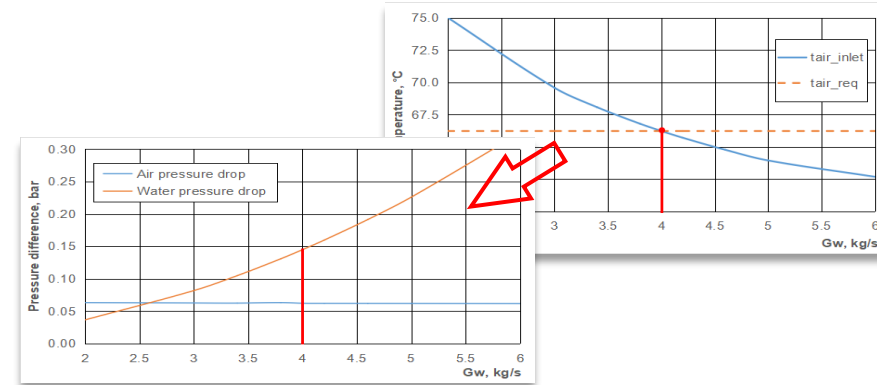
**Goal:** Creation of simulation model for compressor intercooler from known geometric data and analyze it at different modes.

## Tasks:

- Modeling heat transfer and hydraulic resistance of ribbed tubes
- Calculating pressure losses of air flow in intercooler
- Determining cooling water flow rate to provide desired air outlet temperature
- Predicting cooling water pressure drop to provide required mass flow rate



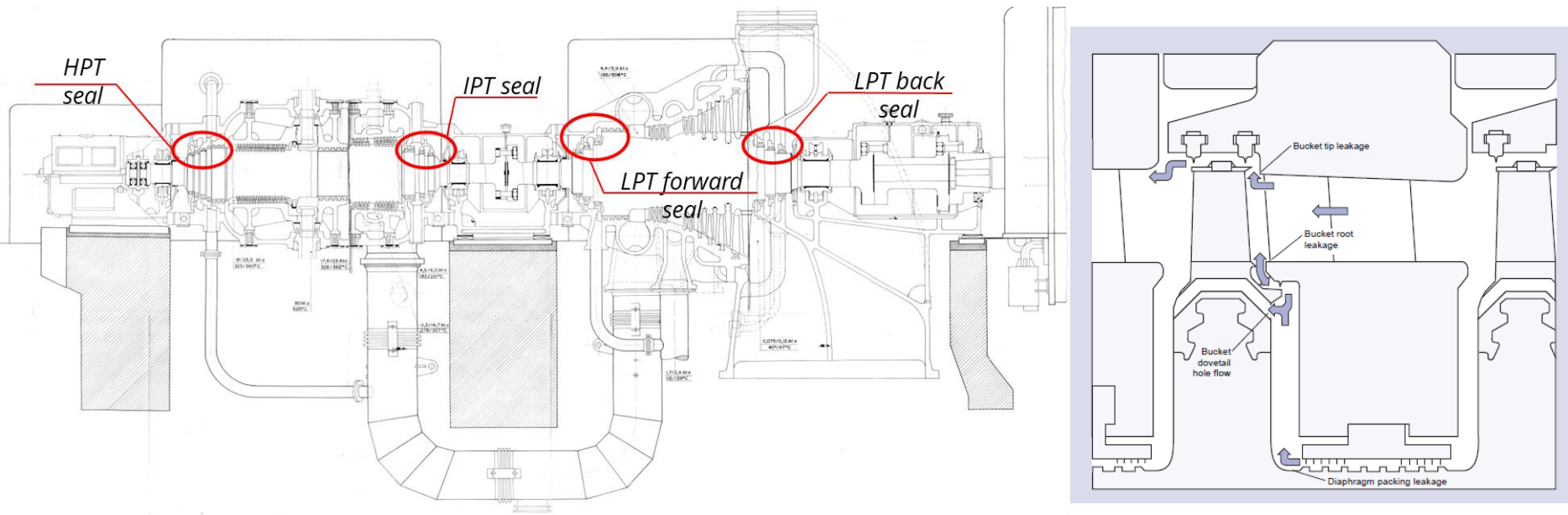
Multi-stage compressor showing location of the intercooler



# Seals & Bearings Modeling

# Seals

- Seals help improve machine and system performance
  - Reduce leakages at shaft ends and for each blade row
- Less leakage = more flow in blades to produce power/pressure ratio
- Higher upstream vs. downstream pressure ratio = more leakages
  - Higher rotor leakages in impulse vs. reaction turbine

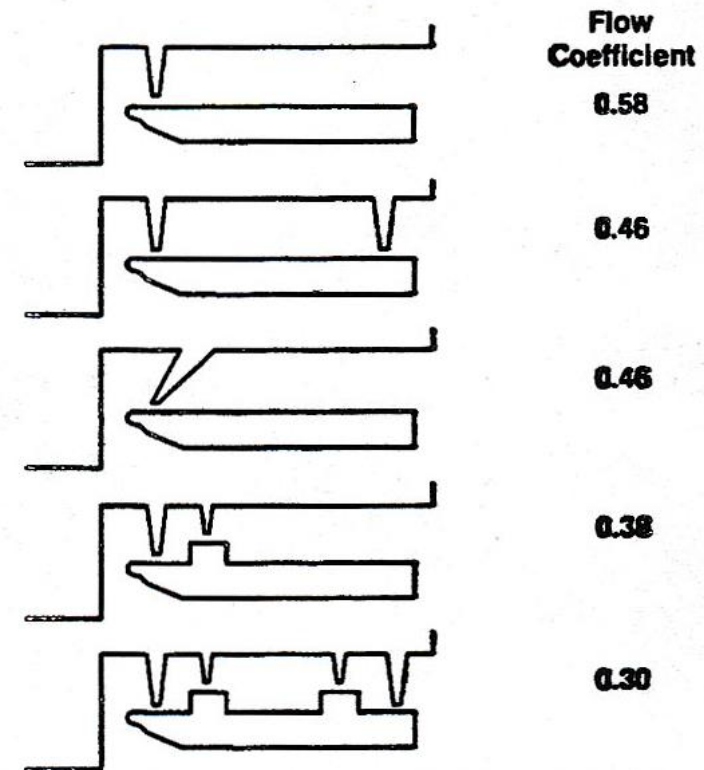


# Seals

- Seal geometry determines flow coefficient

$$\Delta G = k\mu F \sqrt{\frac{P_0}{v_0} \sqrt{\frac{1 - \varepsilon^2}{z}}}$$

- $k$  - seal factor
- $\mu$  - seal mass flow coefficient
- $F$  - clearance circumferential area
- $P_0, v_0$  - pressure and volume before clearance
- $\varepsilon = P_1/P_0$  - steam pressure relation
- $z$  - number of seal teeth

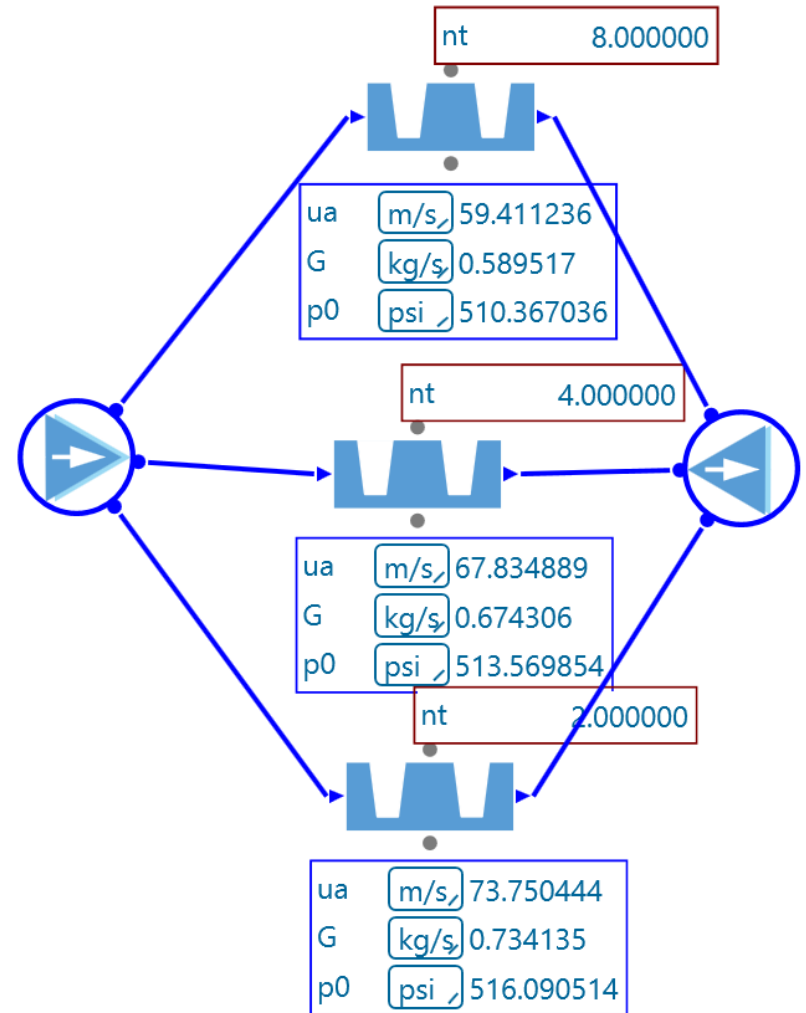


# Straight-Smooth vs. Stepped Seals

- 2 main configurations:
  - Stepped vs. straight seals

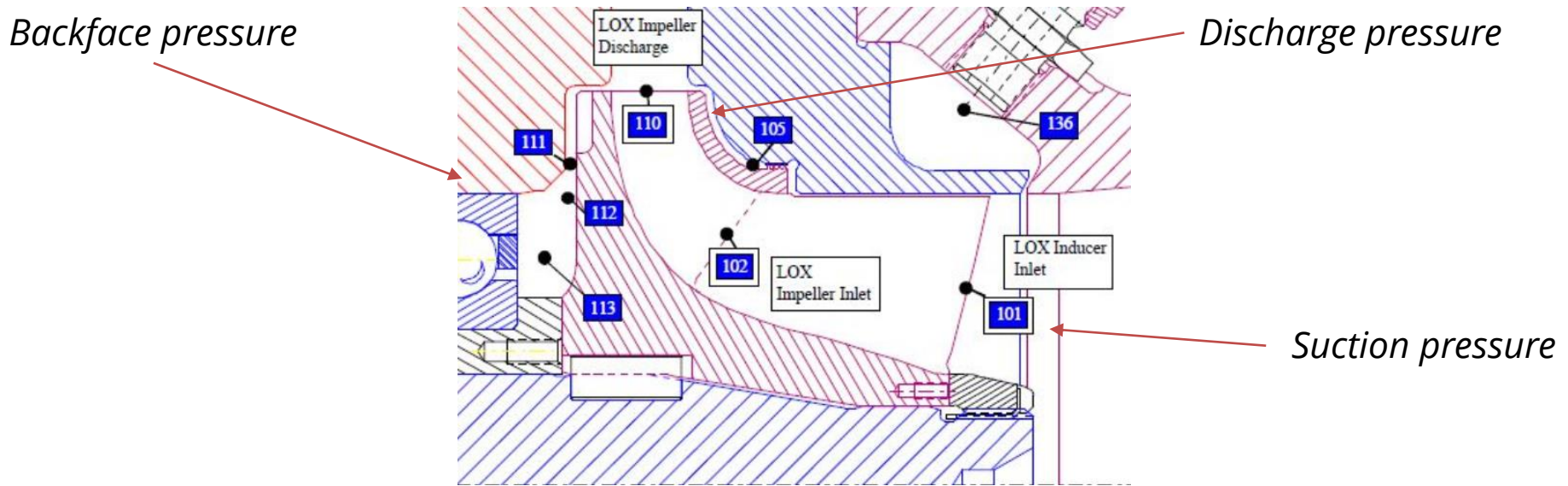


- Stepped seals provide higher flow resistance
  - Less leakages
  - Harder to manufacture and operate



# Axial Load Balancing – Impellers

- Axial load depends on upstream and downstream pressures
  - Therefore component pressure ratio, quality of seals, etc.
- $F_1$  = Axial impeller force = Discharge-side – Suction-side axial force
- $F_j$  = Momentum force =  $\rho Q \Delta V_{ax}$
- $F_1 + F_j$  = Axial Thrust =  $f(\rho, g, Head, Mean outlet impeller diameter, k)$ 
  - $k$  - coefficient based on specific diameter

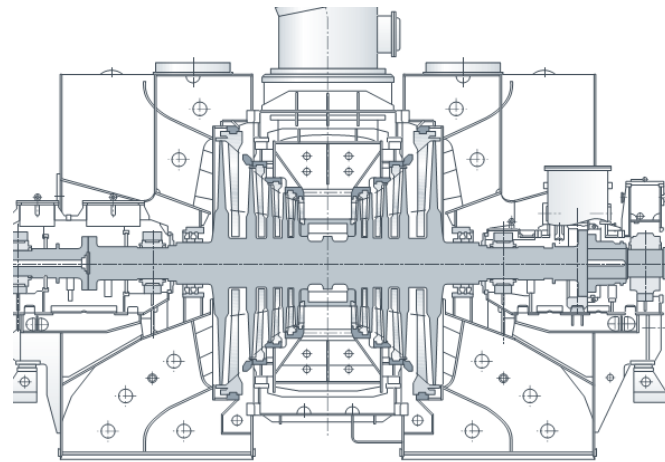
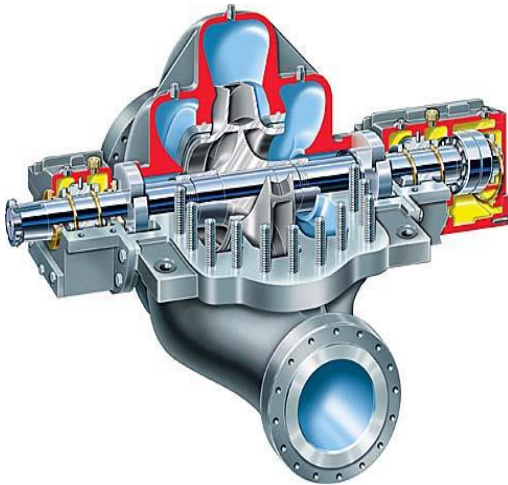


# Ways to Achieve Axial Thrust Balancing

- Mechanically
  - Use of thrust bearing

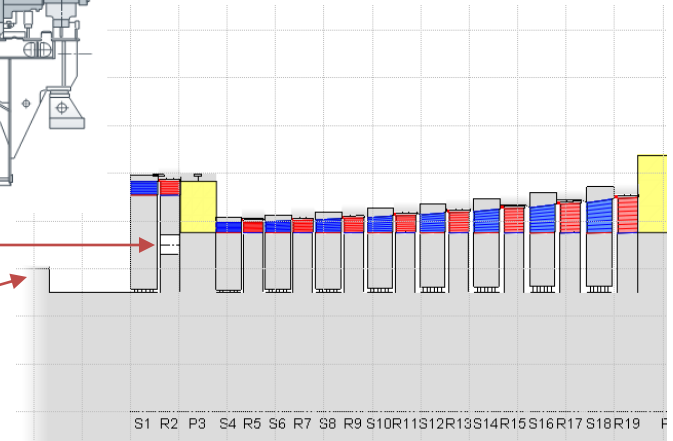


- Design-based
  - Back-to-back arrangement of impellers or stages



- Balance holes

- Balance pistons/reaction drum



# Leakage Calculations for Bearing Analysis

## Bearing cooling requirements and leakage amount determination

### Friction power of bearings:

$$N_{fr} = f(F, \omega)$$

$F = \sqrt{Fa^2 + Fr^2}$  - total bearing load;

$Fa$  - axial load on bearing (from axial loading calculation);

$Fr$  - radial load (from rotordynamics calculation);

$\omega$  - shaft rotational speed (maximal available ).

### Leakage value of MFR is clarified using condition:

$$\frac{2(G_i - G_{i-1})}{G_i + G_{i-1}} < 0.001$$

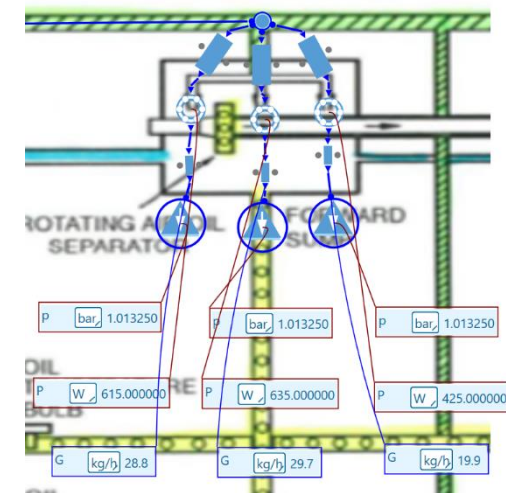
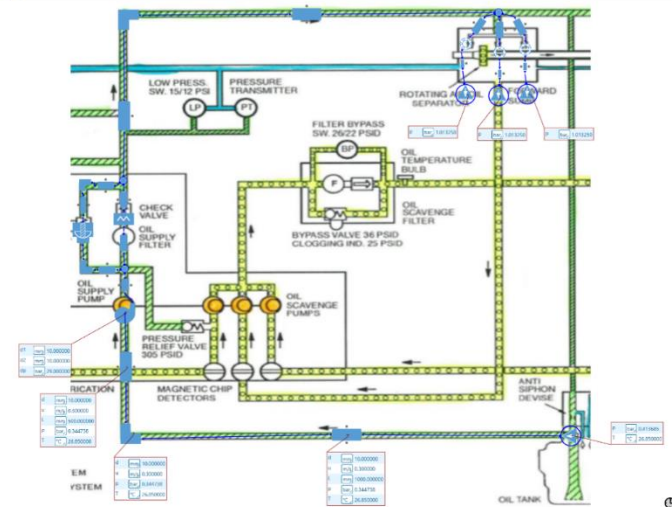
$G_i$  - leakage MFR value at current iteration;

$G_{i-1}$  - leakage MFR value at previous iteration;

For the first iteration the leakage MFR is assigned arbitrarily. The second iteration and its subsequent is determined by secondary flow system calculation until convergence.

# Bearing cooling

- **Goals**
  - Create an oil supply system to provide oil from a tank for bearings
  - Estimate, knowing the power friction of each bearing, the needed mass flow rate to obtain acceptable bearing cooling
- **Fluid** considered – Incompressible B3-V oil
- **Results** – System accurately modeled using series of inlet and outlet with pressure components, pipes, elbows, lossless junctions, bearings by friction power, a pump, a valve and a local fluid resistance
- Cooling flow rate determined for all 3 bearings to provide acceptable operating conditions

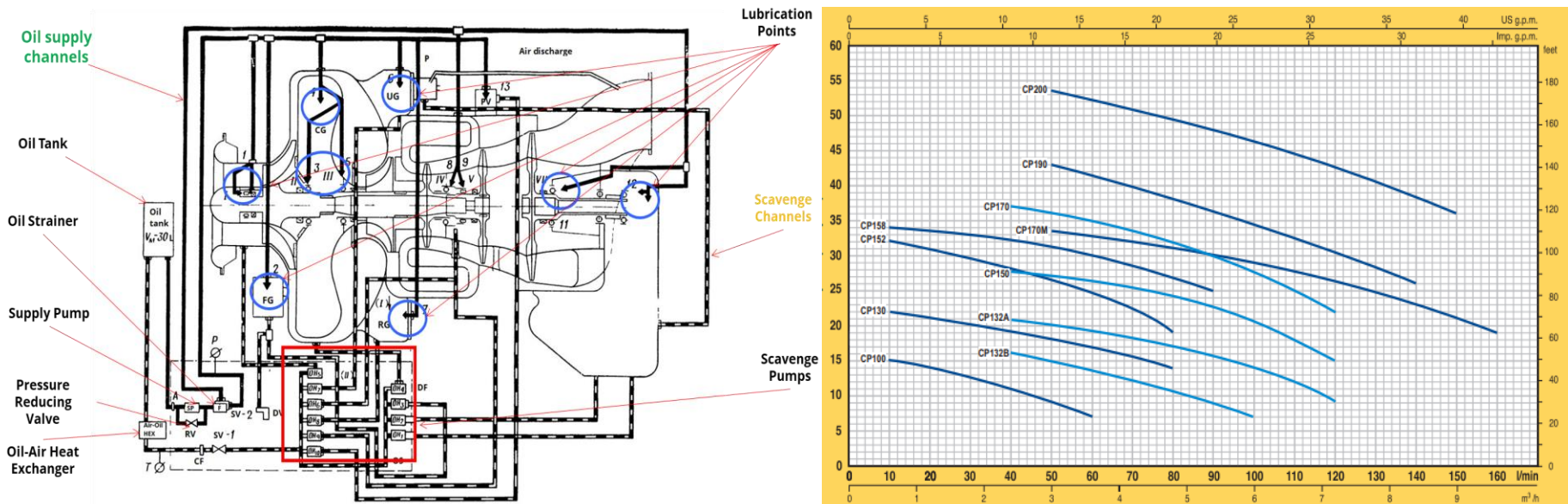


# Gas Turbine Lubrication System

**Goal:** Creation of digital twin of pressure relief valve GTE oil (B-3V) system for steady-state modelling.

## Challenges:

- Determination of pressure levels in the supply pipes
- Selection of proper equipment to ensure required mass flow rate given pump characteristics



# Gas Turbine Lubrication System

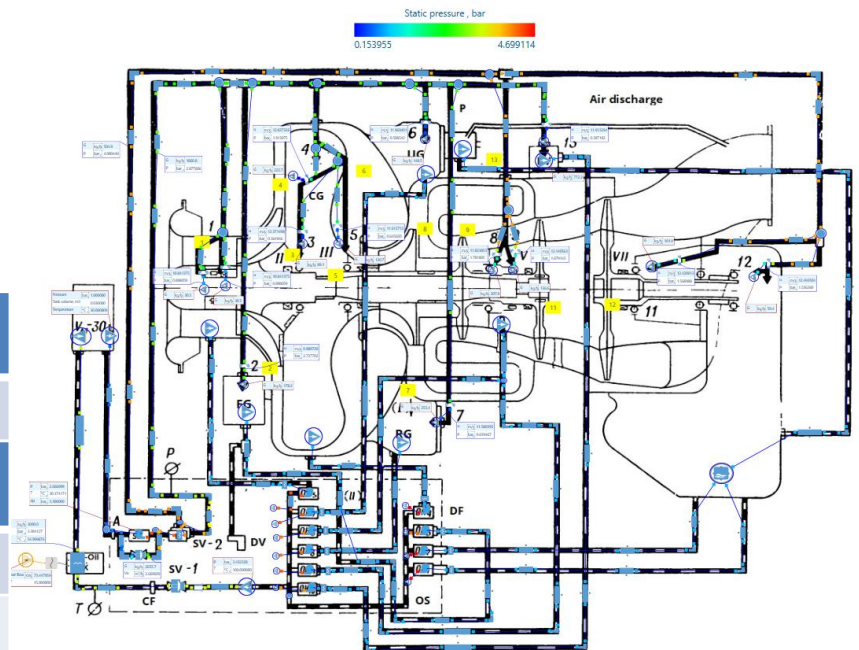
- Pressure levels determined based on known oil mass flow rate for each bearing for cooling and lubrication
- Scavenge system used to clean up the air-oil mixture for oil recycling

Table editor PumpCharacteristics(Pedrollo 170).axtb

| 2.5    | 3      | 3.5    | 4      | 4.5    |
|--------|--------|--------|--------|--------|
| 370000 | 360000 | 350000 | 340000 | 330000 |

Interpolation:       Extrapolation:

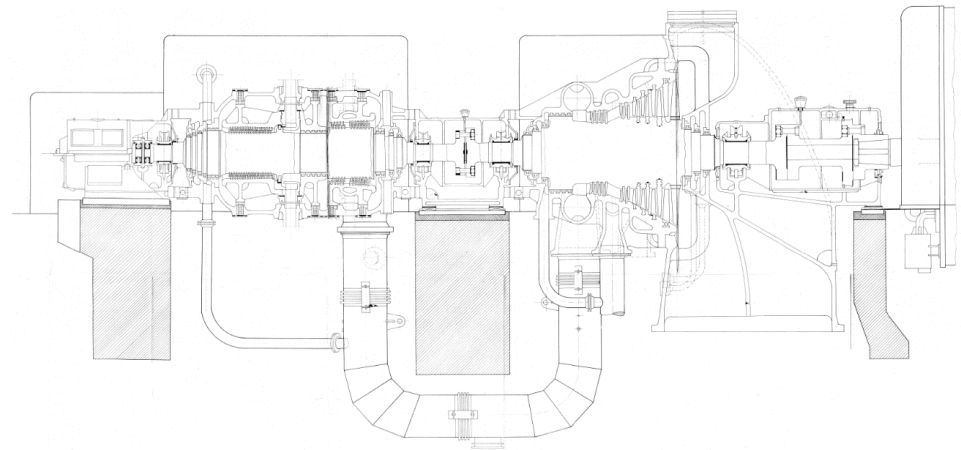
| Pump              | SP    | OH1   | OH2  | OH3    | OH4    | OH4   | OH4    | OH4   | OH4    | OH4    | OH4   | OH10 |       |
|-------------------|-------|-------|------|--------|--------|-------|--------|-------|--------|--------|-------|------|-------|
| l/min             | 59    | 60    | 60   | 40     | 60     | 16    | 36     | 13    | 29     | 16     | 16    |      |       |
| Lubrication point | 1     | 2     | 3    | 4      | 5      | 6     | 7      | 8     | 9      | 10     | 11    | 12   | 13    |
| Supply, l/min     | 1.2   | 2.5   | 1.5  | 3.8    | 2.2    | 2.2   | 2.5    | 4.3   | 3.5    | 2.3    | 1.7   | 13.0 | 1     |
| Supply, kg/h      | 71.28 | 148.5 | 89.1 | 225.72 | 130.68 | 148.5 | 255.42 | 207.9 | 136.62 | 100.98 | 772.2 | 59.4 | 71.28 |



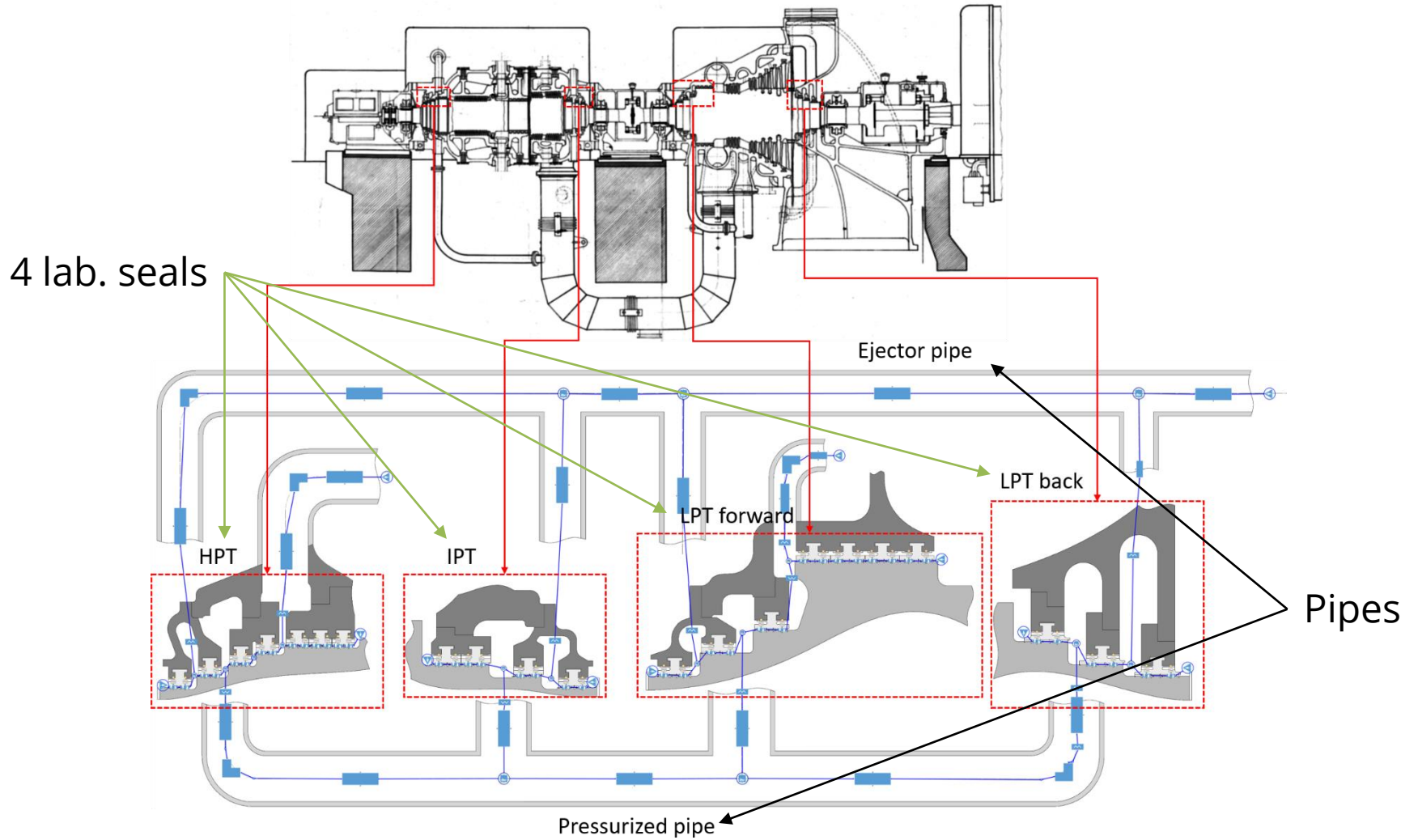
# Case Study – Seal Modeling in Steam Turbines

# Examples of Seal Modeling in Steam Turbines

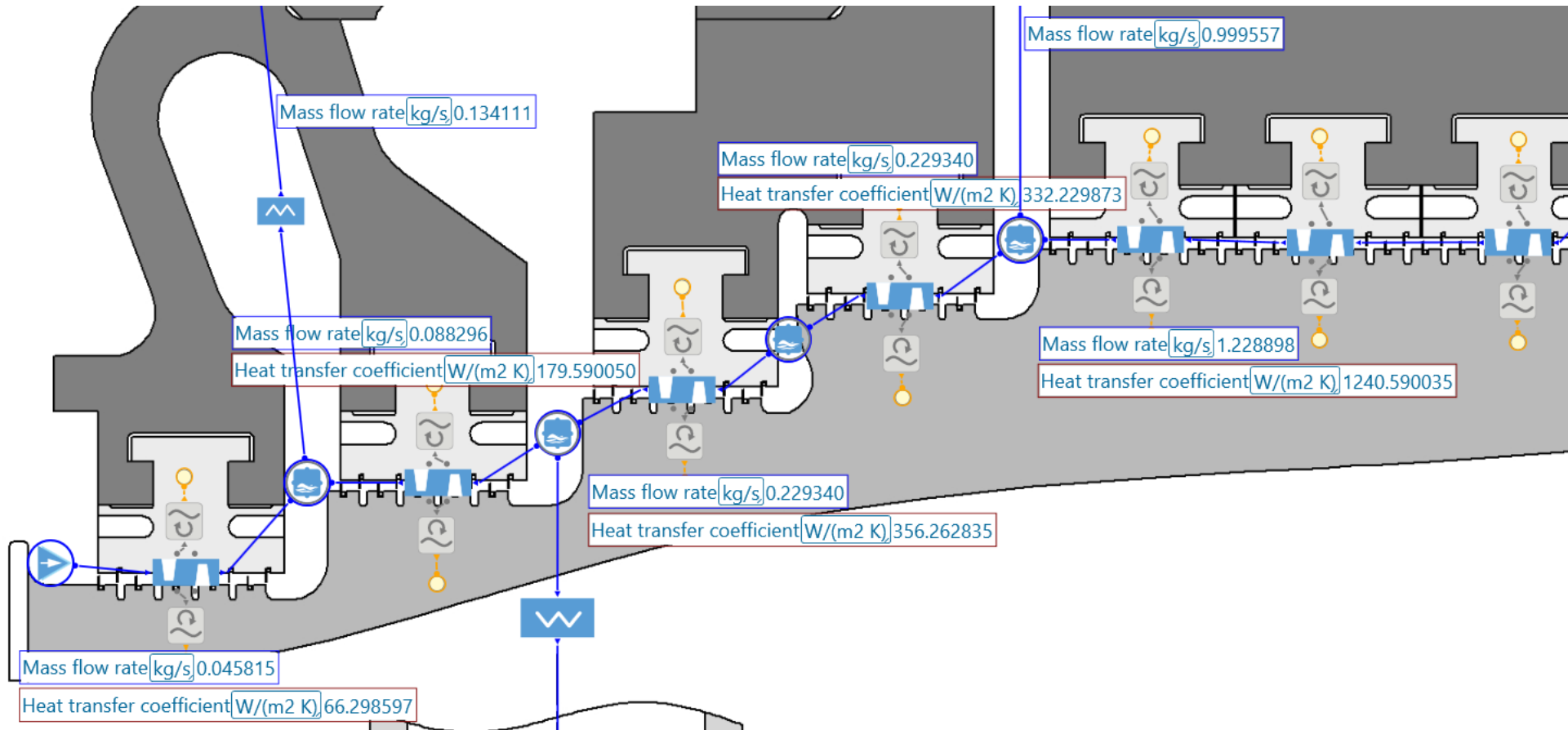
- Modeling of leakages and heat transfer in seals
  - End seals
  - Stage seals
- Goal 1: Determination of minimum mass flow through lab. seals & selection of seal design
  - Why? Amount of leakage determines overall machine efficiency
- Goal 2: Determination of heat transfer coefficient for each seal case
  - Used to account for thermal expansion of rotor and stator parts (start-up)
  - Calculation of thermal boundary conditions for thermo-stress estimation
- Working fluids: Air & steam
- Modeling of flows in
  - Cross-over pipes
  - Exhaust hoods
  - Etc.



# Steam Turbine Example



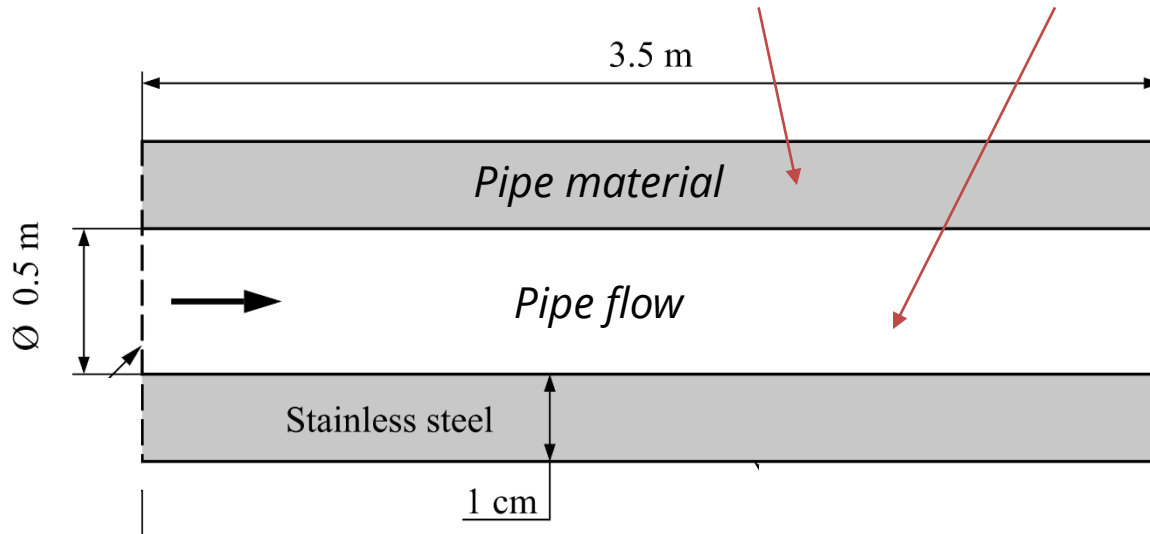
# Detailed Steam Turbine Seal Modeling



# Heat Transfer

# Conduction

- Definition – transfer of heat through collisions of particles and movement of electrons WITHIN a body
  - Responsible for heat transfer inside a material or inside a fluid

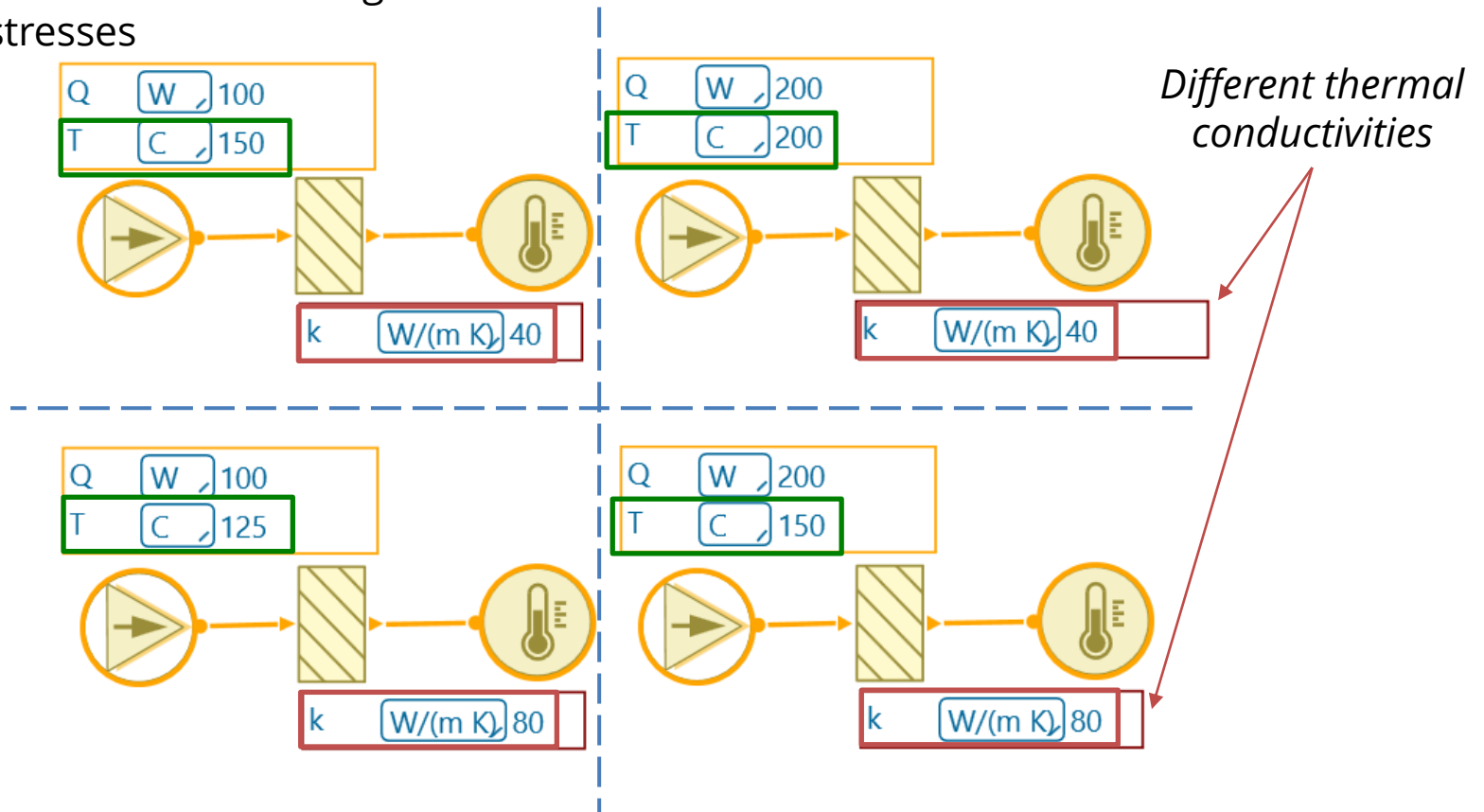


$$\frac{\partial Q}{\partial t} = -k \oint_S \nabla T \cdot dS$$

- The rate, in time, of heat transfer through a material is proportional to the negative gradient in the temperature and to the area at right angles, to that gradient, through which the heat is flowing

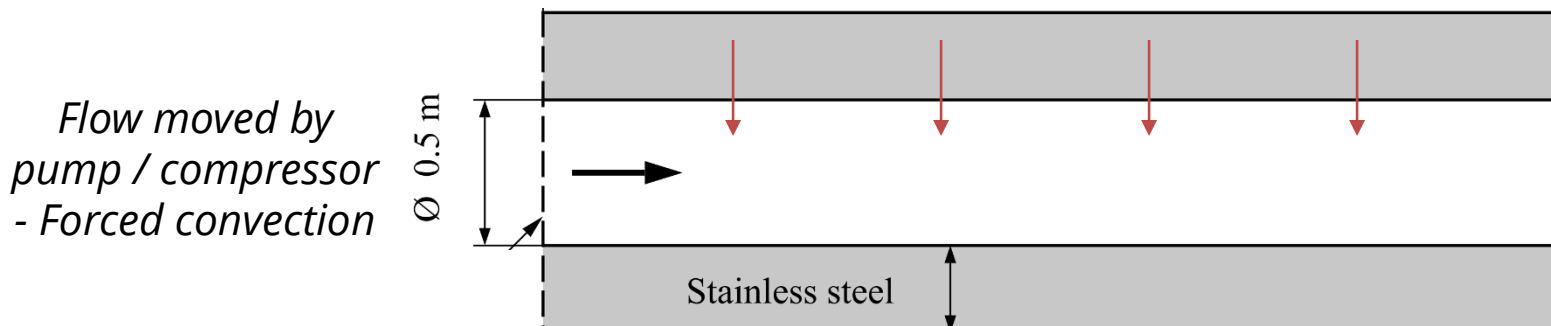
# Thermal Conductivity

- In previous equation,  $k$
- Ability of material to conduct heat
  - Good in metals and gasses – better heat distribution vs. local thermal stresses

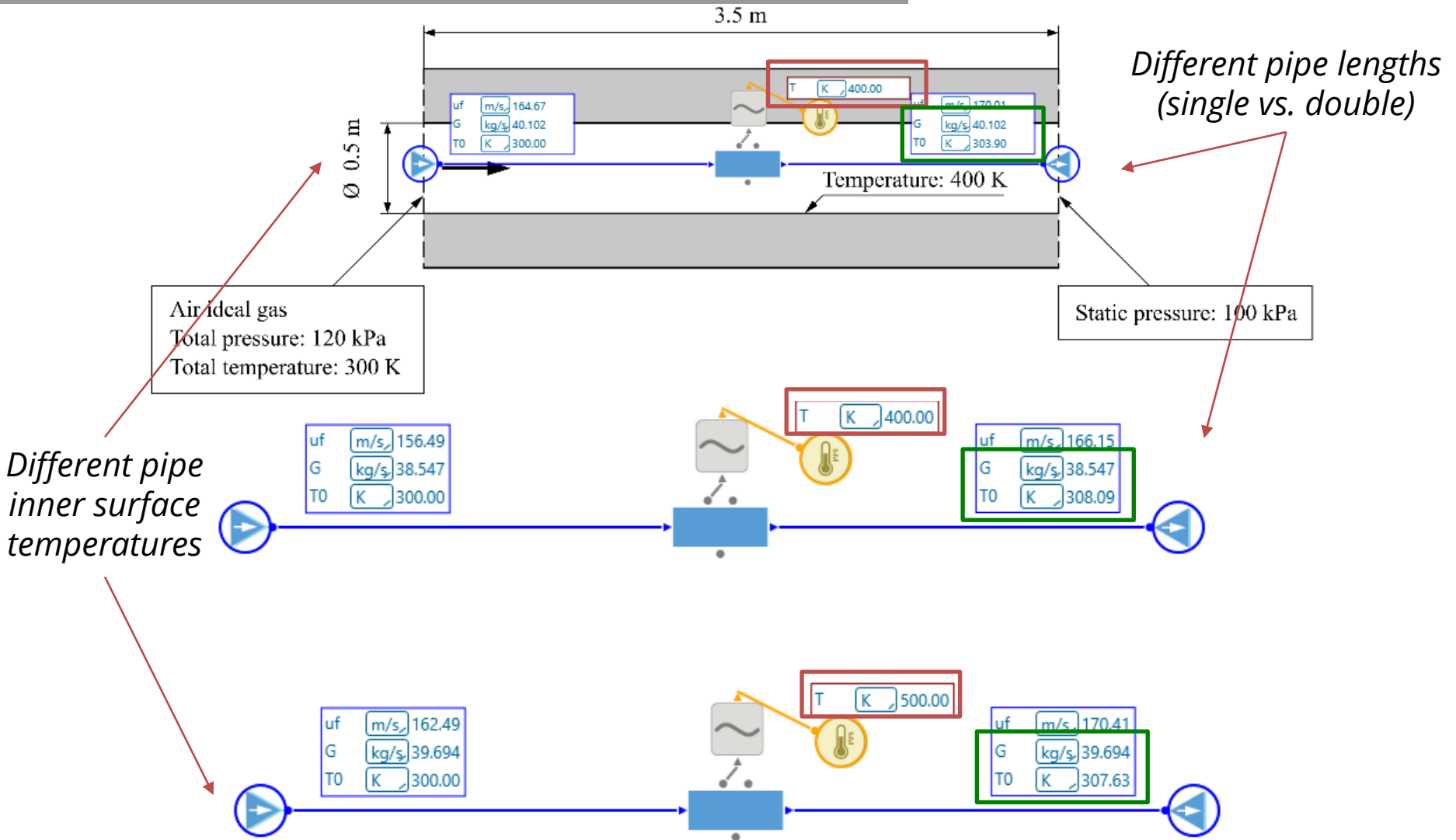


# Convection

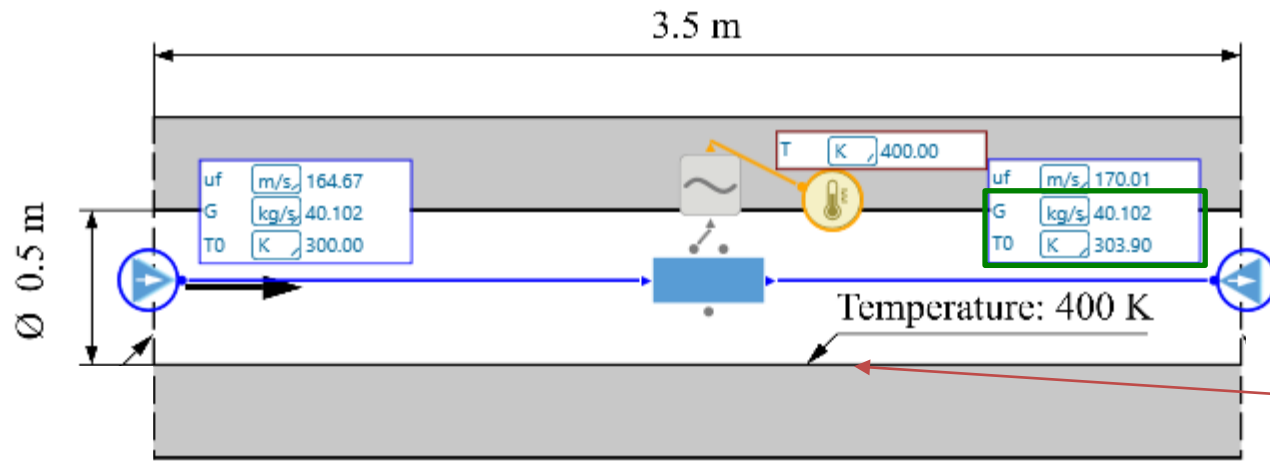
- Definition – transfer of heat through movement of matter in fluids
  - Advection (directional transfer of energy)
  - Diffusion (non-directional)
- Not in solids due to bulk current flows or diffusion being too low
- Convection also responsible for mass transfer
- 2 types
  - Free heat convection (natural) – fluid bulk flow due to temperature-induced differences in buoyancy
  - Forced heat convection – fluid is moved by forces other than buoyancy
    - Type of interest in most turbomachinery applications



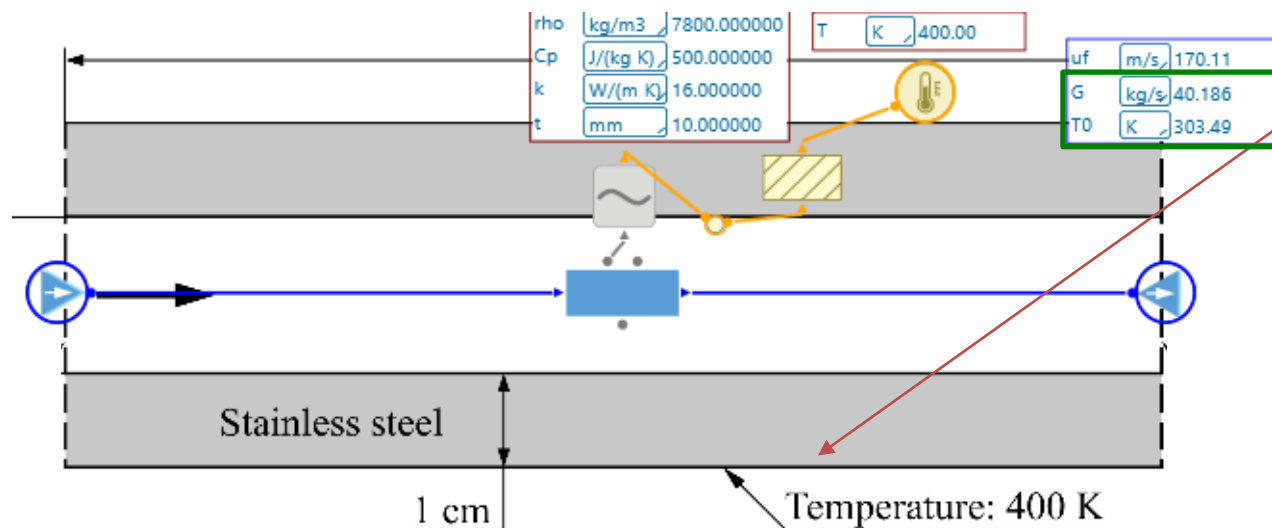
# Convection in Pipe Flow Example



# Conduction & Convection in Pipe Example

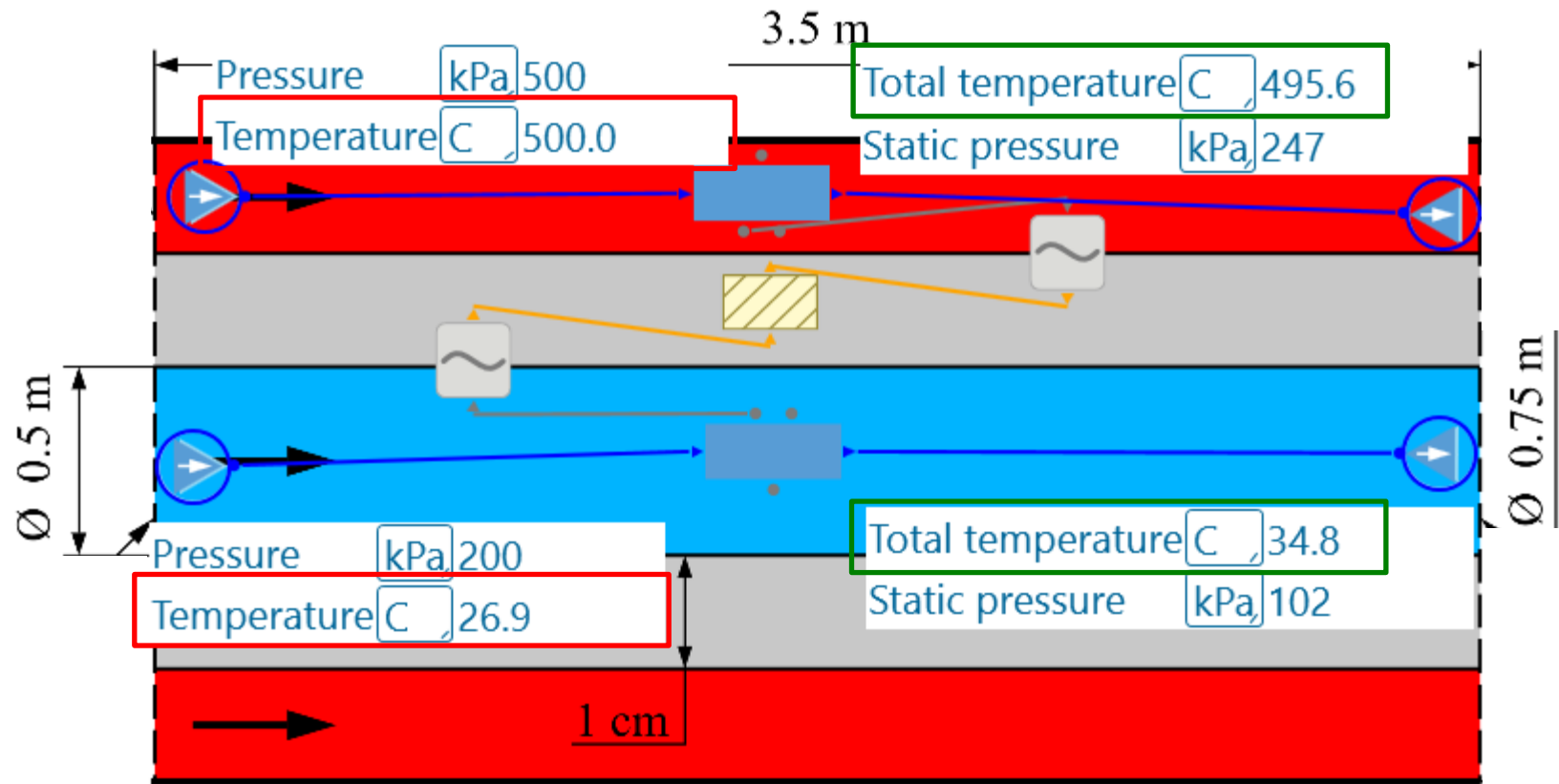


*Known temperature at inner (top) vs. outer (bottom) pipe surface*



# In- & Outside Pipe Flows Heat Exchange

- Series of heat exchanges possible – Hot fluid -> pipe surface -> cold fluid



# Convection in Cooling

- Convection can be used to determine cooling/heating system effectiveness
  - Based on heat flow rates and solid & fluid temperatures
- Main heat transfer in cooled gas turbine blades
  - Flow in serpentine removes heat from blade material
  - Then conduction tries equalizing the material temperature
- Example – Cooling channels should be sized properly in order to maintain blade material temperature below melting point
  - At fixed inlet (bleed) and outlet (turbine flow path) pressures, increasing channel cross-section area increases flow through cooled blade which decreases metal temperature through convection
- *Blade acts as heat exchanger*

# Heat Transfer Coefficient

- Definition – Proportionality constant between heat flux and thermodynamic driving force for the flow of heat (i.e. temperature difference)
- Also called film coefficient or film effectiveness 
$$h = \frac{q}{\Delta T}$$
- From general heat exchange equation  $\dot{Q} = hA(T_2 - T_1)$ 
  - $A$  - surface area where heat transfer takes place |  $T_2$  - temperature of surrounding fluid |  $T_1$  - temperature of solid surface
- Different equations are used depending on the considered element, their orientation, etc.

# Equations for Convection - Examples

- Different equations exist to model convection under various circumstances

- Dittus & Boelter equation

$$Nu = 0.023 \cdot Re^{0.8} \cdot Pr^{0.4}$$

- Kays equation

$$Nu = 0.022 \cdot Re^{0.8} \cdot Pr^{0.6}$$

- Kulikova equation for binary mixtures of inert gasses

$$Nu = 0.044 \cdot Re^{0.73} \cdot Pr^{0.58}$$

- Water and light liquids equation

$$Nu = 0.0155 \cdot Re^{0.83} \cdot Pr^{0.5}$$

- Heavy liquids and oils

$$Nu = 0.0118 \cdot Re^{0.9} \cdot Pr^{0.3}$$

- Mikheyev equation

$$Nu = 0.021 \cdot Re^{0.8} \cdot Pr^{0.43}$$

- Petukhov equation

$$Nu = \frac{\frac{\lambda}{8} \cdot Re \cdot Pr}{k + 12.7 \cdot \sqrt{\frac{\lambda}{8}} \cdot \left( Pr^{\frac{2}{3}} - 1 \right)}$$

$$\lambda = (1.8 \cdot \lg Re - 1.64)^{-2}$$

$$k = 1.07 + \frac{900}{Re}$$

$$Nu = 5 + 0.015 \cdot Re^a \cdot Pr^b$$

- Sleicher & Rouse equation

$$a = 0.88 - \frac{0.24}{4 + Pr}$$

$$b = \frac{1}{3} + 0.5 \cdot e^{-0.6 \cdot Pr}$$

# Equations for Convection - Application

- Application of different equations to model according to
  - Reynolds number

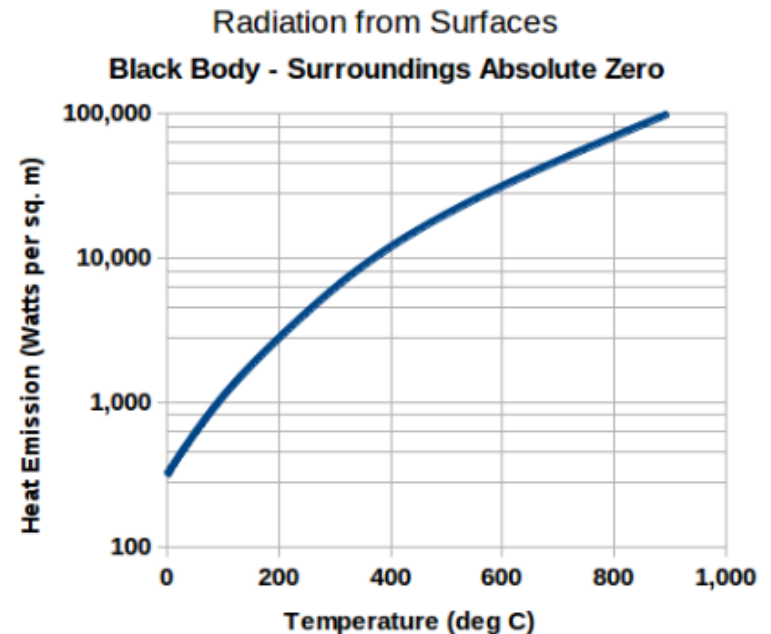
| Reynolds Number Applicability of Different Convection Equations  |       |       |       |       |       |       |       |       |       |       |       |       |       |       |       |       |       |       |       |       |       |       |      |
|--|-------|-------|-------|-------|-------|-------|-------|-------|-------|-------|-------|-------|-------|-------|-------|-------|-------|-------|-------|-------|-------|-------|------|
| 1E+04  | 2E+04 | 3E+04 | 4E+04 | 5E+04 | 6E+04 | 7E+04 | 8E+04 | 9E+04 | 1E+05 | 2E+05 | 3E+05 | 4E+05 | 5E+05 | 6E+05 | 7E+05 | 8E+05 | 9E+05 | 1E+06 | 2E+06 | 3E+06 | 4E+06 | 5E+06 | 5E6+ |
| Dittus and Boelter equation  |       |       |       |       |       |       |       |       |       |       |       |       |       |       |       |       |       |       |       |       |       |       |      |
| Kays equation - Kulikova - Simulation of heat transfer to the gas coolant with low Prandtl number value (2015) |       |       |       |       |       |       |       |       |       |       |       |       |       |       |       |       |       |       |       |       |       |       |      |
| Kulikova equation - Kulikova - Sim   |       |       |       |       |       |       |       |       |       |       |       |       |       |       |       |       |       |       |       |       |       |       |      |
| Petukhov equation  |       |       |       |       |       |       |       |       |       |       |       |       |       |       |       |       |       |       |       |       |       |       |      |
| Sleicher and Rouse equation  |       |       |       |       |       |       |       |       |       |       |       |       |       |       |       |       |       |       |       |       |       |       |      |

- Prandtl Number

| Prandtl Number Applicability of Different Convection Equations |       |       |     |     |   |   |   |   |   |   |   |   |   |    |    |    |    |    |    |    |    |    |     |     |     |     |     |     |     |     |     |     |      |      |      |      |      |      |      |      |      |       |       |       |       |       |       |       |       |       |        |      |
|--|-------|-------|-----|-----|---|---|---|---|---|---|---|---|---|----|----|----|----|----|----|----|----|----|-----|-----|-----|-----|-----|-----|-----|-----|-----|-----|------|------|------|------|------|------|------|------|------|-------|-------|-------|-------|-------|-------|-------|-------|-------|--------|------|
| 0.1  | 0.239 | 0.322 | 0.5 | 0.7 | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 | 20 | 30 | 40 | 50 | 60 | 70 | 80 | 90 | 100 | 120 | 200 | 300 | 400 | 500 | 600 | 700 | 800 | 900 | 1000 | 2000 | 3000 | 4000 | 5000 | 6000 | 7000 | 8000 | 9000 | 10000 | 20000 | 30000 | 40000 | 50000 | 60000 | 70000 | 80000 | 90000 | 100000 | 1E5+ |
| Dittus and Boelter equation                                    |       |       |     |     |   |   |   |   |   |   |   |   |   |    |    |    |    |    |    |    |    |    |     |     |     |     |     |     |     |     |     |     |      |      |      |      |      |      |      |      |      |       |       |       |       |       |       |       |       |       |        |      |
| Kays equation  |       |       |     |     |   |   |   |   |   |   |   |   |   |    |    |    |    |    |    |    |    |    |     |     |     |     |     |     |     |     |     |     |      |      |      |      |      |      |      |      |      |       |       |       |       |       |       |       |       |       |        |      |
| Kulikova equation  |       |       |     |     |   |   |   |   |   |   |   |   |   |    |    |    |    |    |    |    |    |    |     |     |     |     |     |     |     |     |     |     |      |      |      |      |      |      |      |      |      |       |       |       |       |       |       |       |       |       |        |      |
| Mikheyev equation  |       |       |     |     |   |   |   |   |   |   |   |   |   |    |    |    |    |    |    |    |    |    |     |     |     |     |     |     |     |     |     |     |      |      |      |      |      |      |      |      |      |       |       |       |       |       |       |       |       |       |        |      |
| Petukhov equation  |       |       |     |     |   |   |   |   |   |   |   |   |   |    |    |    |    |    |    |    |    |    |     |     |     |     |     |     |     |     |     |     |      |      |      |      |      |      |      |      |      |       |       |       |       |       |       |       |       |       |        |      |
| Water & light liquids equation                                 |       |       |     |     |   |   |   |   |   |   |   |   |   |    |    |    |    |    |    |    |    |    |     |     |     |     |     |     |     |     |     |     |      |      |      |      |      |      |      |      |      |       |       |       |       |       |       |       |       |       |        |      |
| Sleicher and Rouse equation                                    |       |       |     |     |   |   |   |   |   |   |   |   |   |    |    |    |    |    |    |    |    |    |     |     |     |     |     |     |     |     |     |     |      |      |      |      |      |      |      |      |      |       |       |       |       |       |       |       |       |       |        |      |

# Radiation

- Definition – Heat exchange done through electromagnetic radiation waves or rays
- All matter with a non-absolute zero temperature emits thermal radiation over a wide range of frequencies
- Intensity of thermal radiation proportional to  $T^4$  [K] & Area of the emitting body
- More applicable to *heat exchangers* in the context of turbomachinery systems
  - Not the focus in this presentation

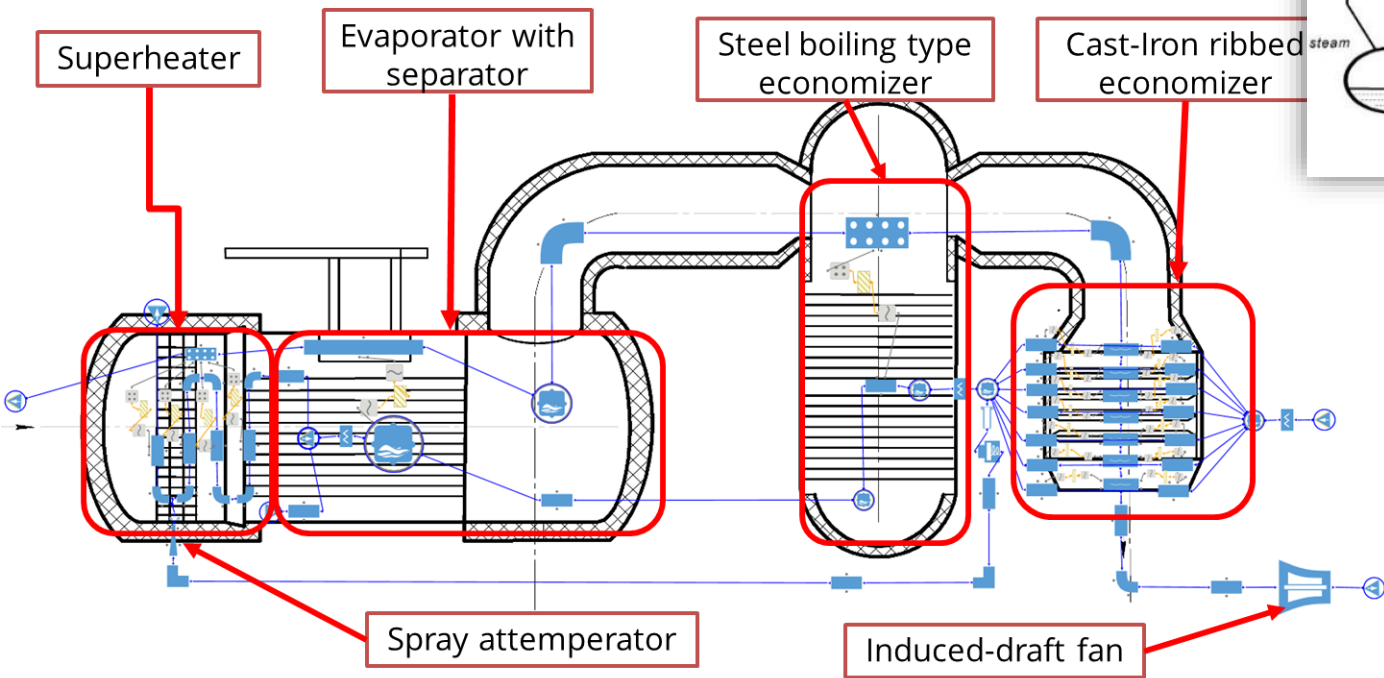
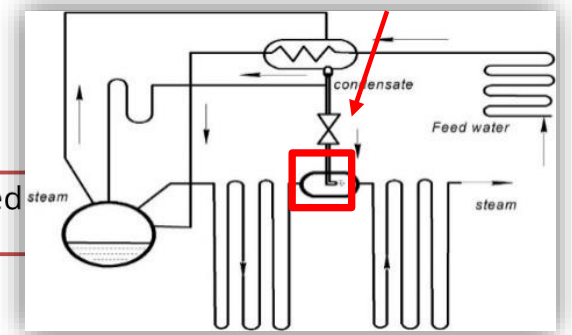


# Heat Recovery Steam Generator Modeling

# HRSG with Spray Attenuator

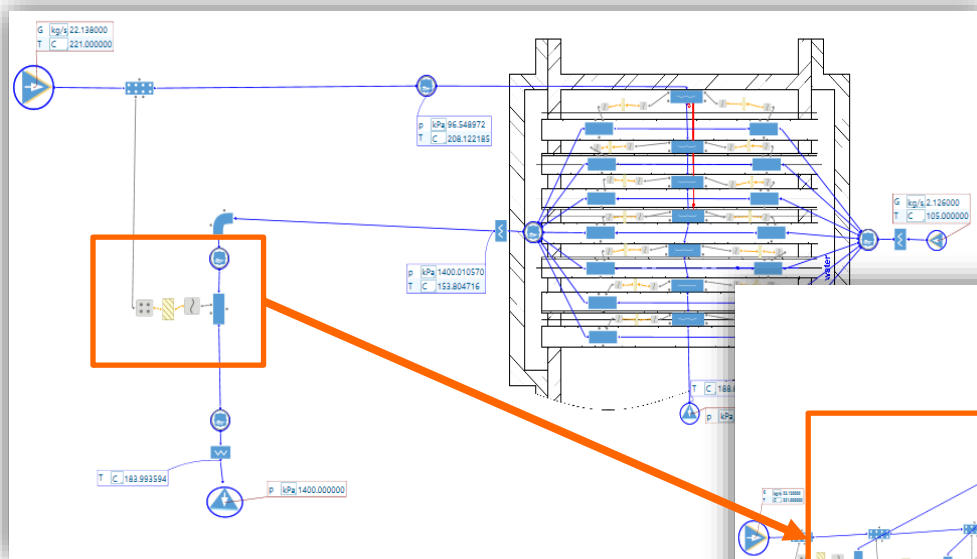
**Objective:** Analyze working HRSG at a few regimes with different ranges of environment temperatures.

Water /injection

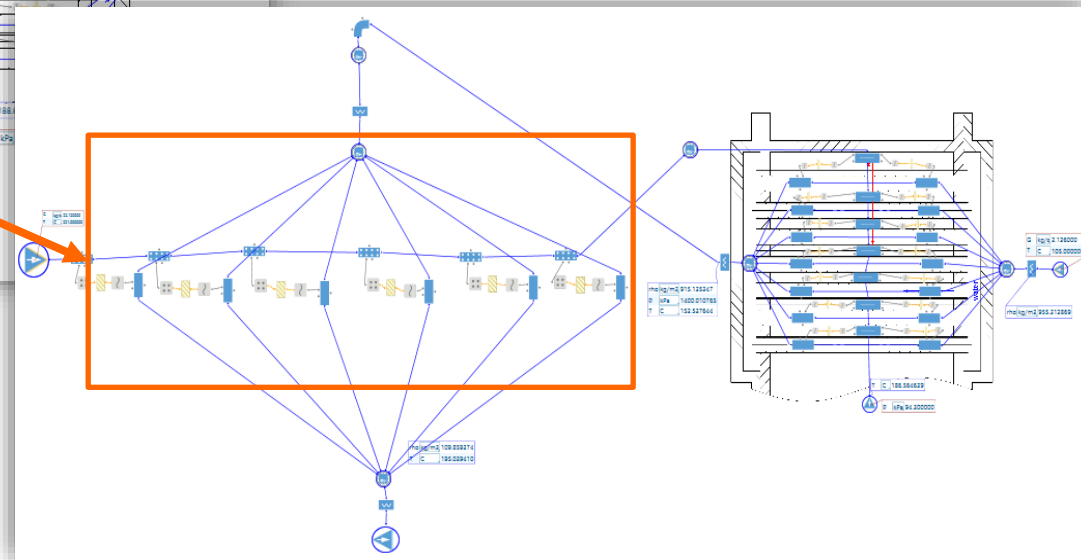


# Simple vs. Detailed Modelling

- Based on desired complexity and fidelity of calculation, detailed systems can be simplified by averaging similar flow conditions
  - Example for economizer



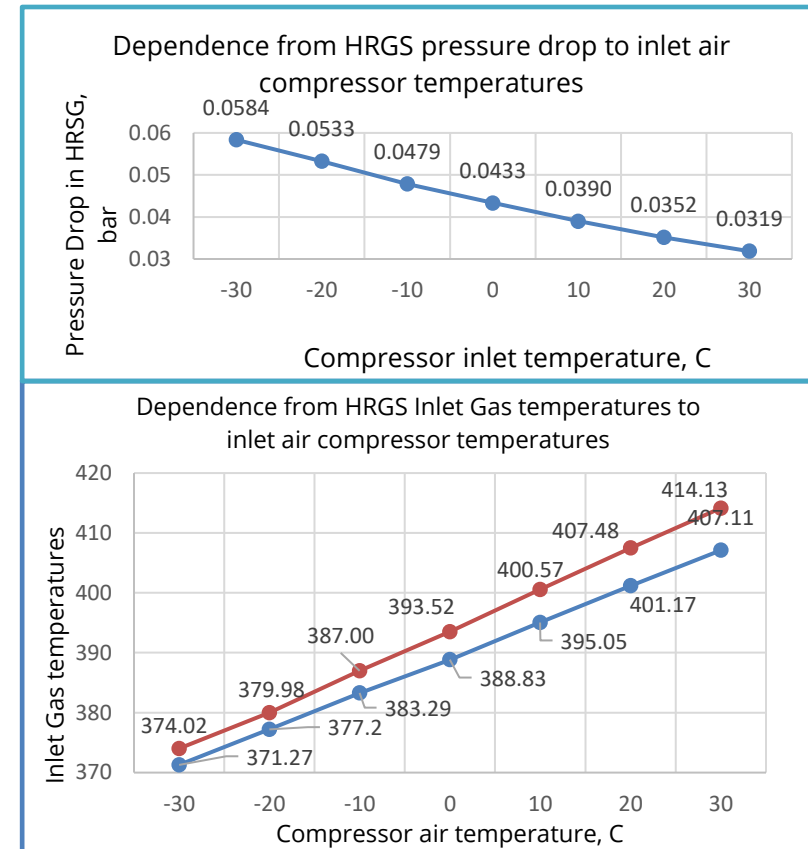
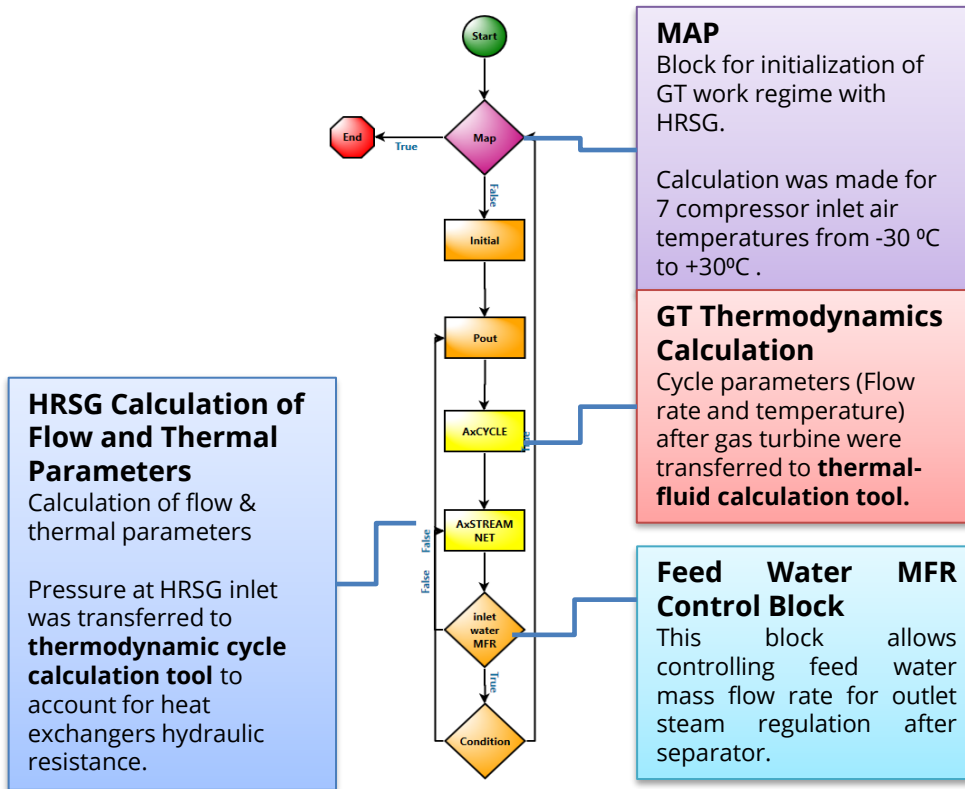
Non-interval economizer model



Interval economizer model

# Automated Process

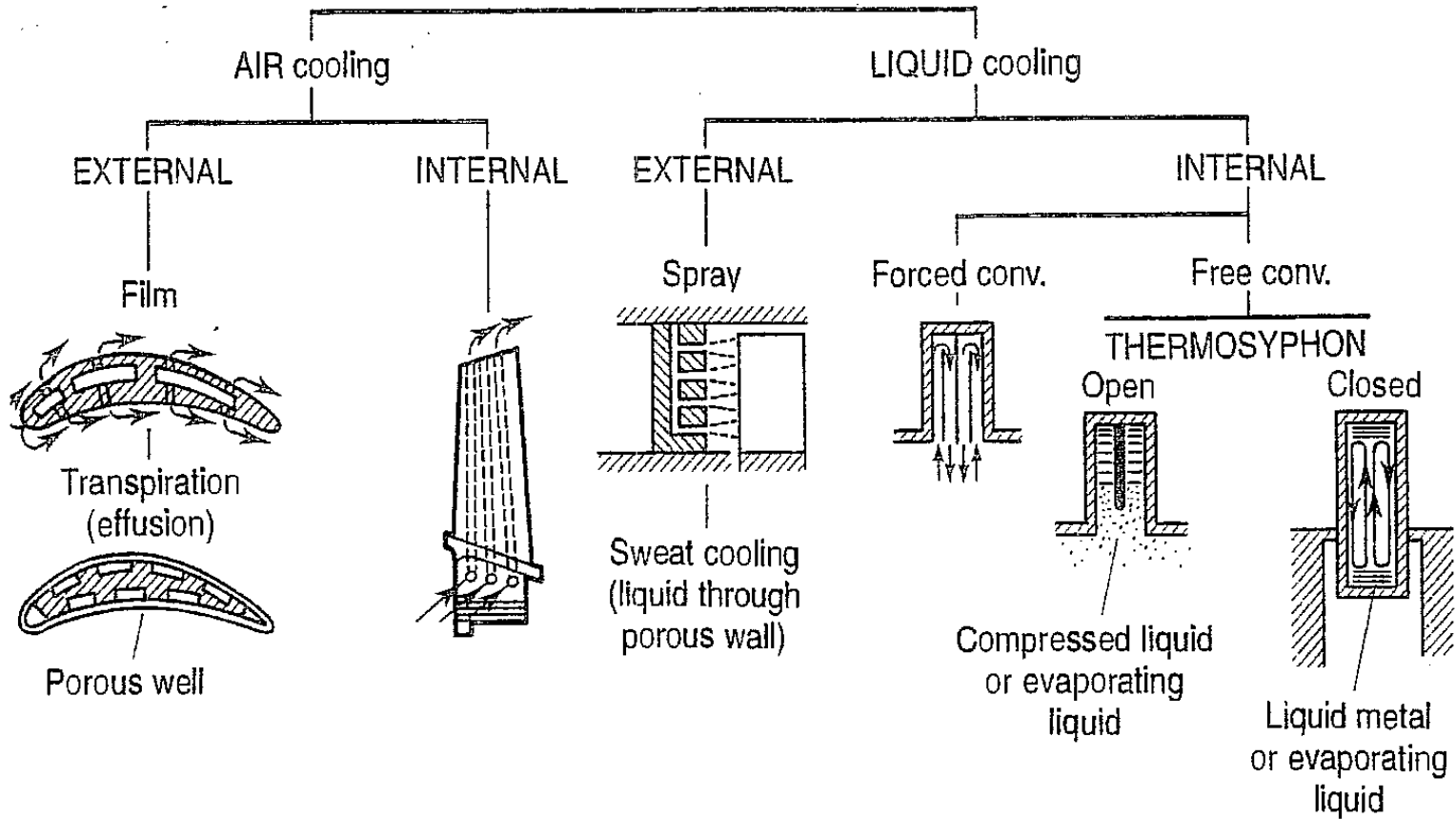
The iteration process allows users to utilize the exact values of the parameters at all key points, automatically combining calculations in different tools



# Cooling

# Internal vs. External Cooling

- Different cooling methods available



# Gas Turbine Cooling Challenges

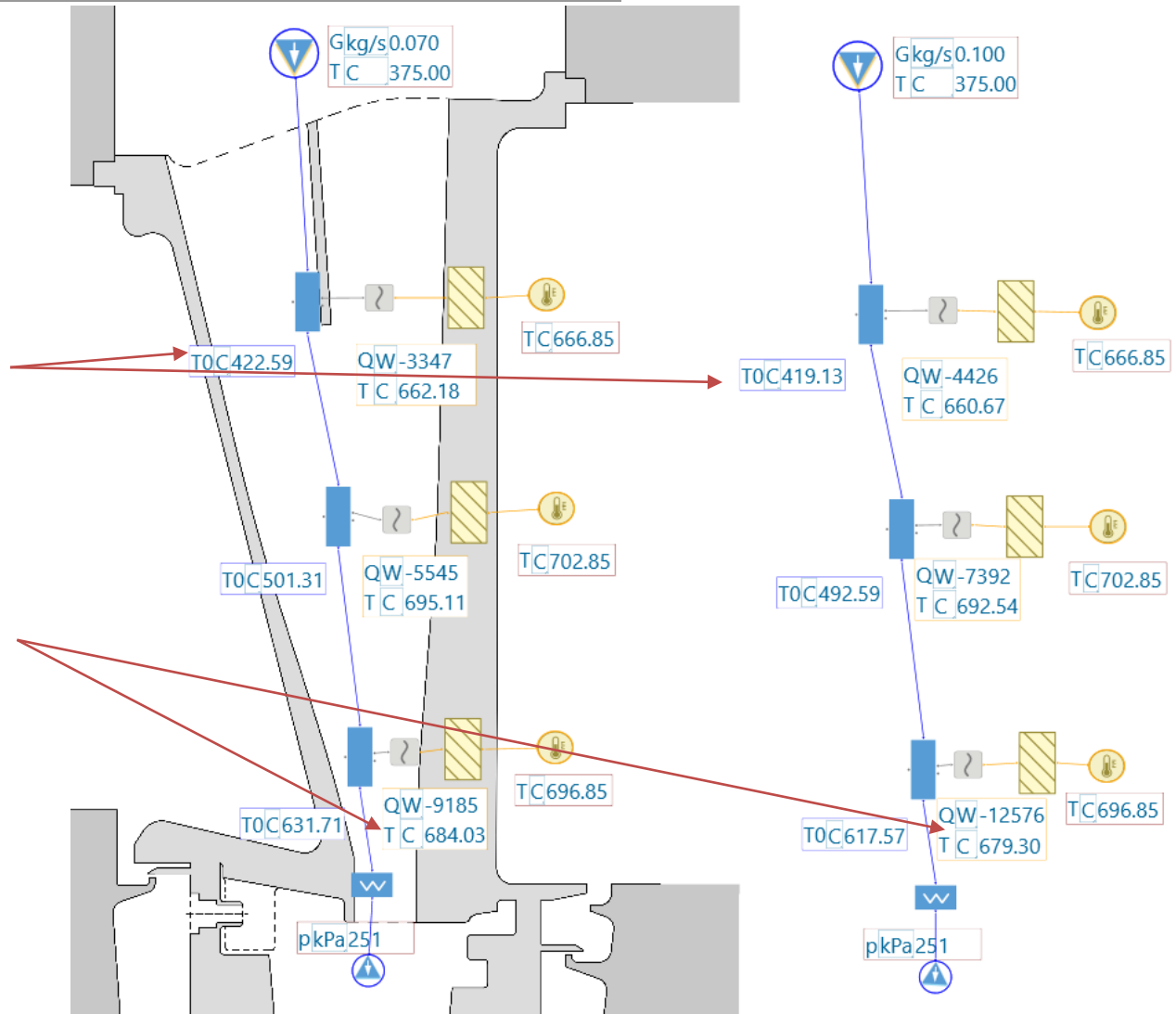
- Gas turbine rotor cooling flow from compressor bleed to turbine side
  - Rotor-stator & rotor-rotor cavities
- Goal: Swirl determination, heat pick up, pressure drop
  
- Cooled blades
  - Serpentine, film cooling, etc.
- Goal: Mass flow rate, pressure and temperature in cooled blades
  
- Working fluid: Air
  
- Objective - Determination of correct cooling flow parameters at mixing location with main gas flow to accurately calculate turbine efficiency

# Cooled Stator Blade Convection Example

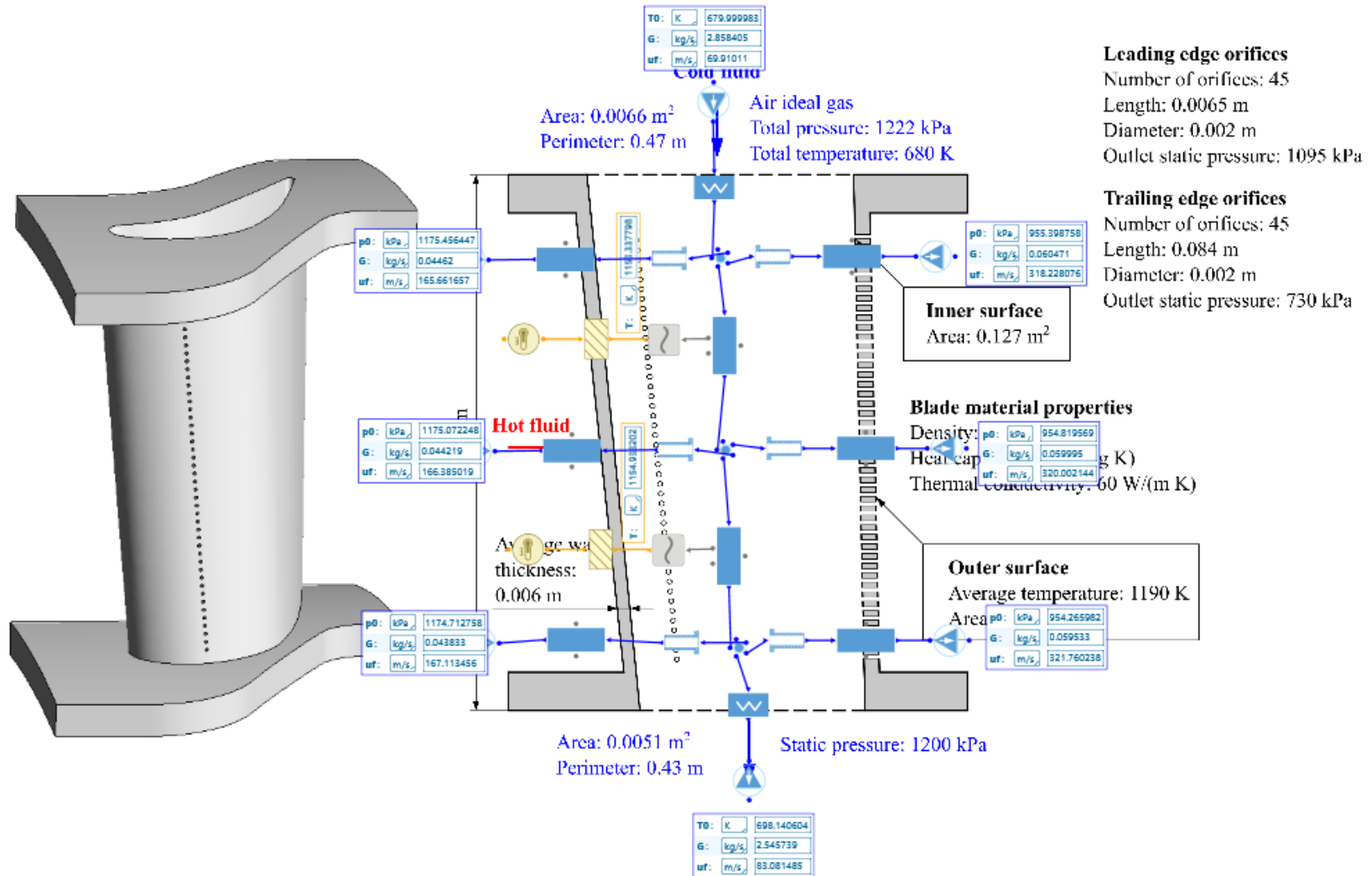
30% less mass flow rate (left) vs. baseline (right)

Different fluid temperatures due to convection

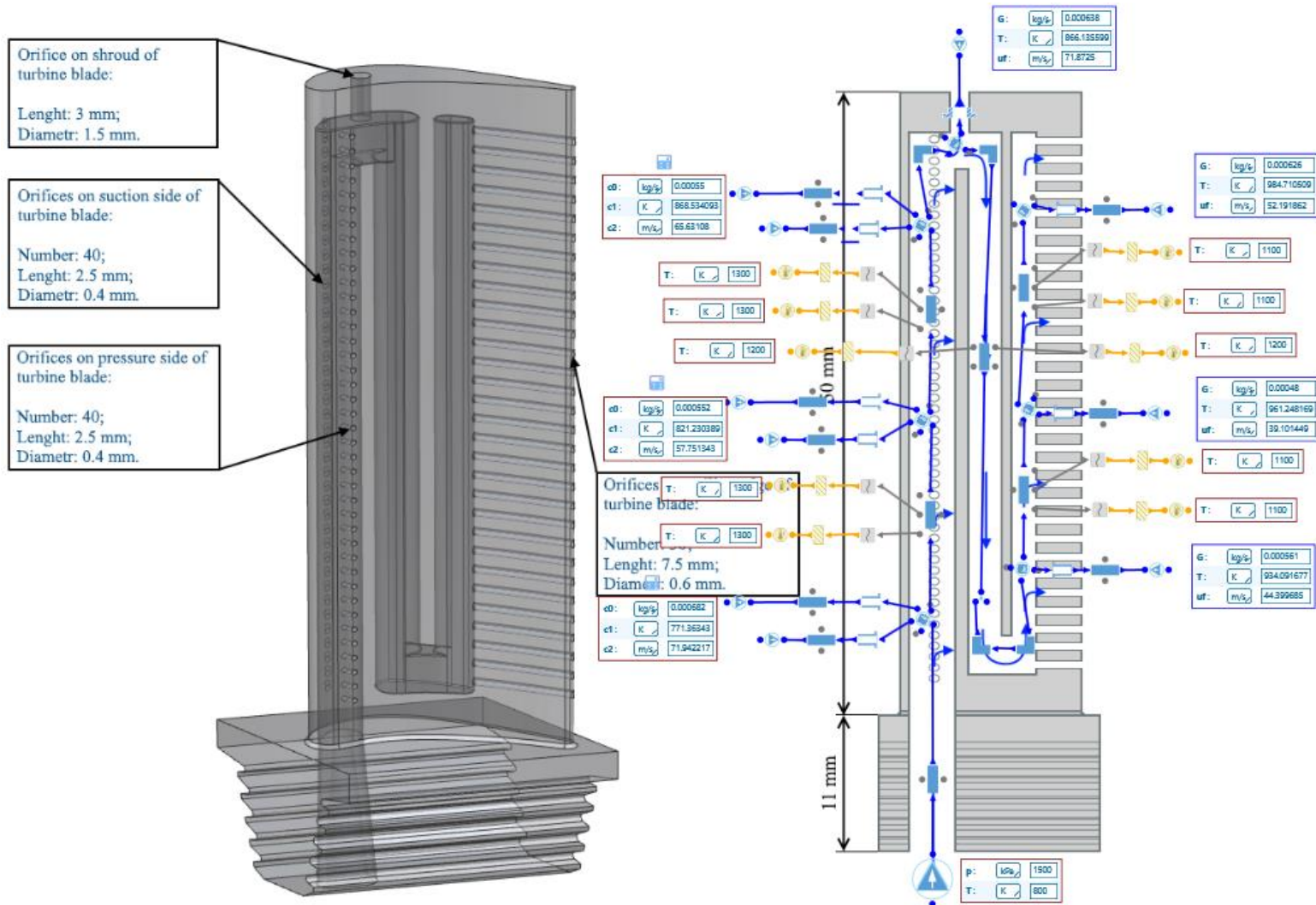
Different solid inner surface temperatures due to conduction



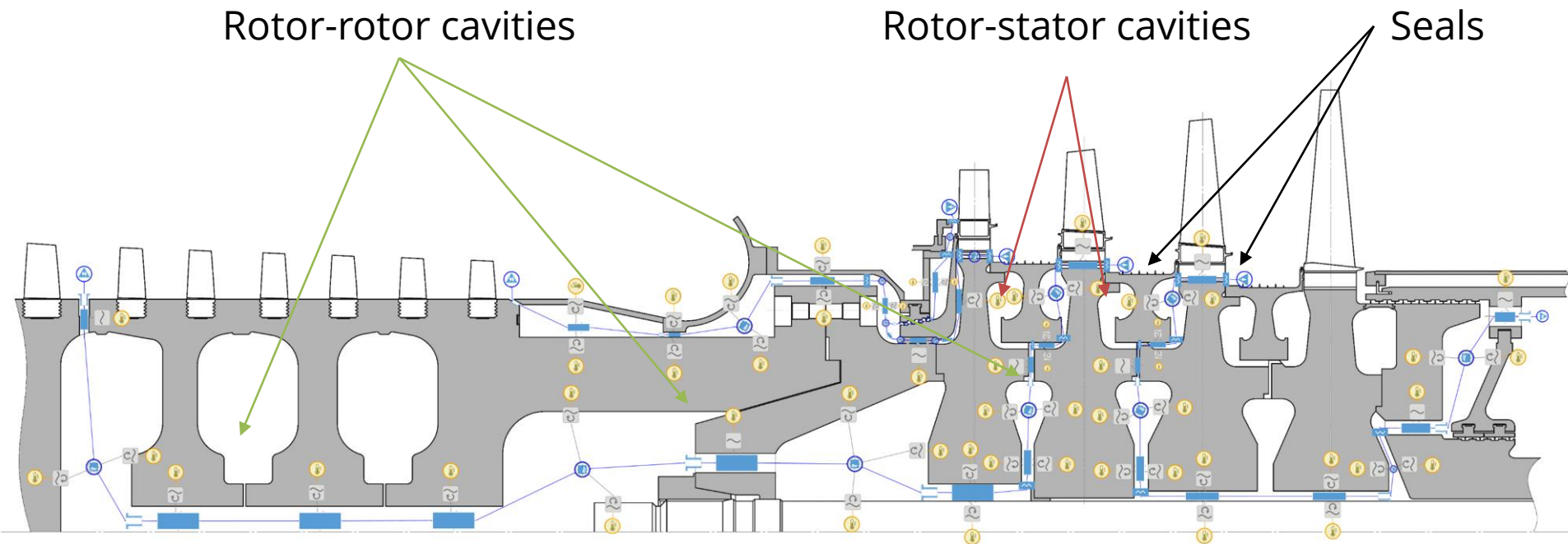
# Gas Turbine Example - 3<sup>rd</sup> Stator Blade



# Gas Turbine Example - Rotor Blade Cooling

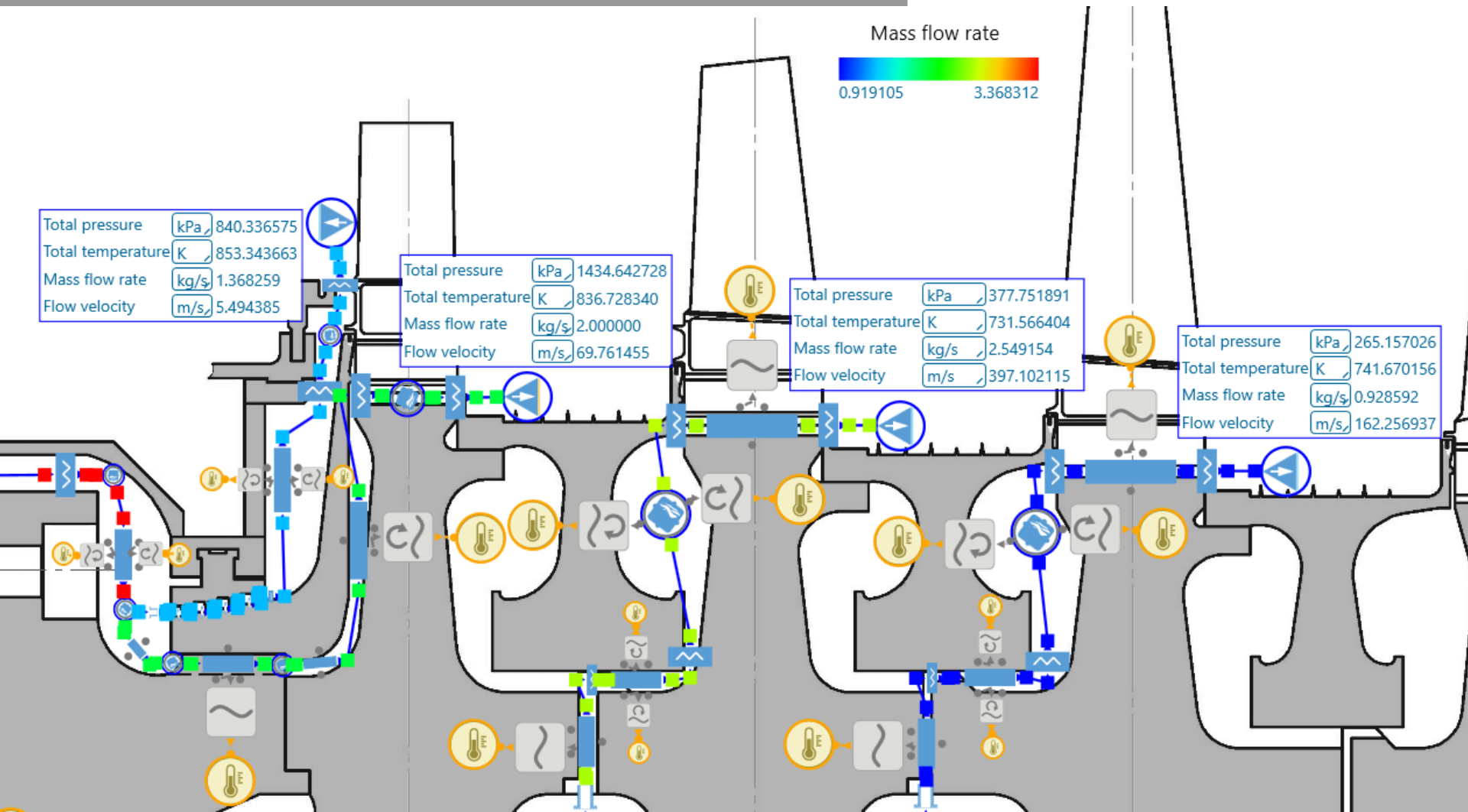


# Gas Turbine Example - Rotor





# Detailed Turbine Cooling Modeling



# Case Study - Optimization of Cooling Flow Channels

# Blade Cooling System Optimization

Goal: Find optimal geometry for one gas turbine cooling row within limit of thermo/structural constraints for the blade for maximal stage efficiency by varying cooling holes diameter using DoE approach and determine dependency of stage efficiency from cooling flow mass flow rate.

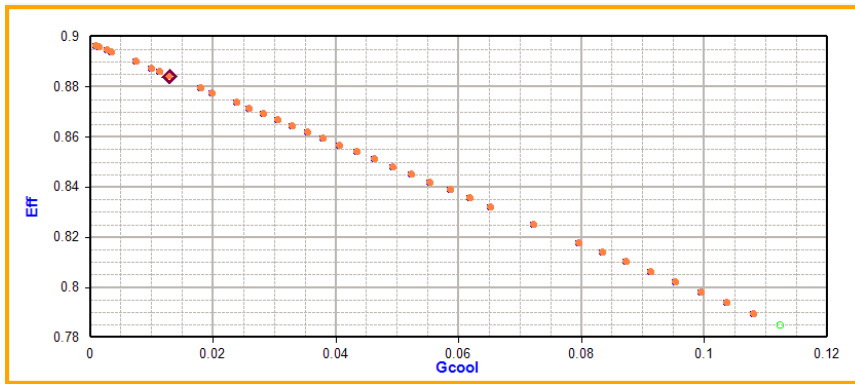
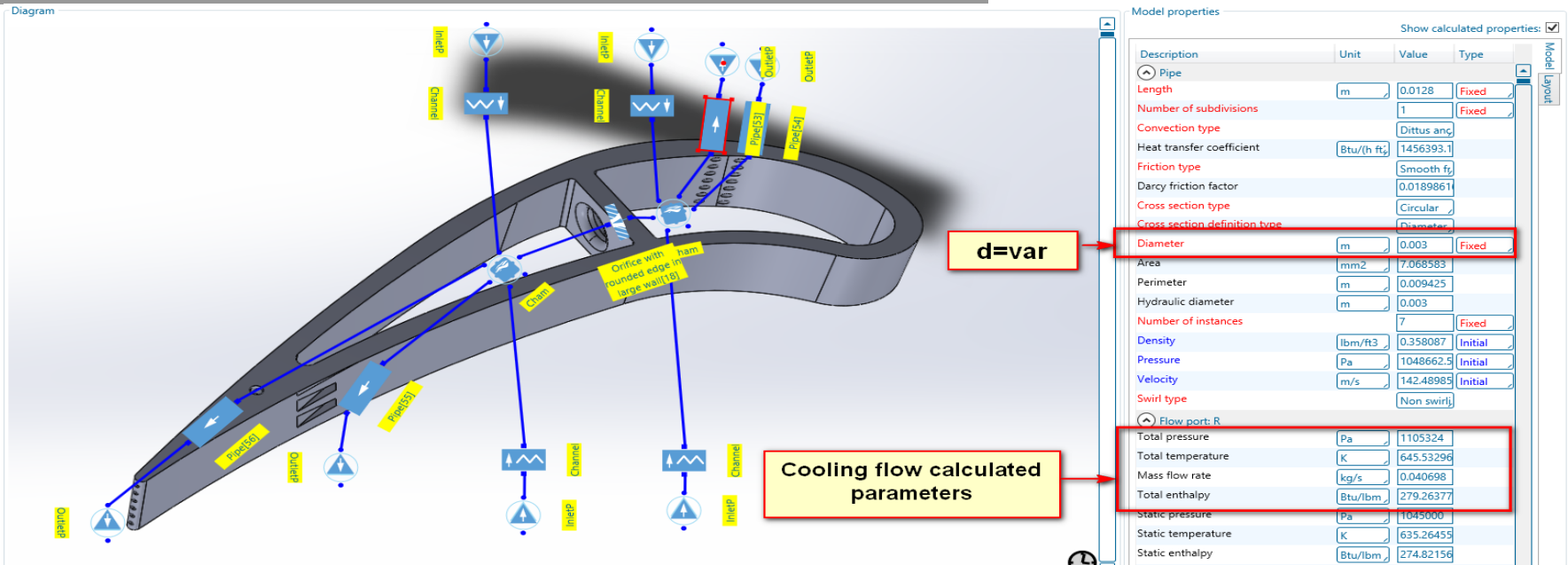
Objective function:

$$F(d_{opt}) = \max[\textit{Efficiency} = f(\textit{MFR}_{cool}; P_{cool}; T_{cool}, \textit{Equivalent Stress}_{max}, T_{max}, \dots)]$$

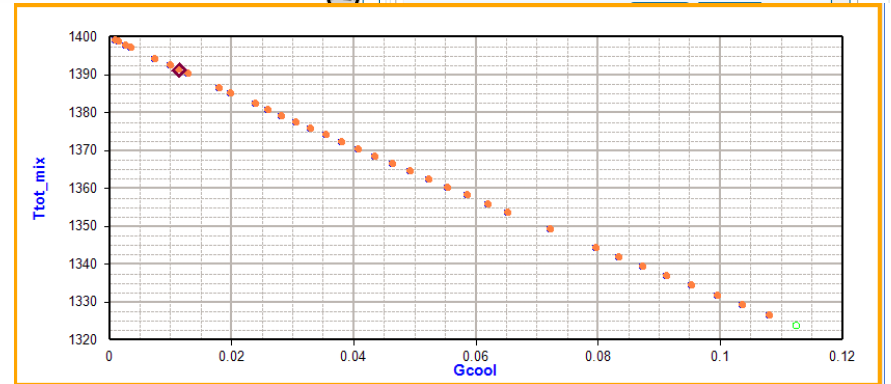
Steps:

- Design of cooled turbine stage
- Stator and blade geometry creation
- Boundary conditions and results of calculation parameterization

# Blade Cooling System Optimization



Stage efficiency vs cooling mass flow rate

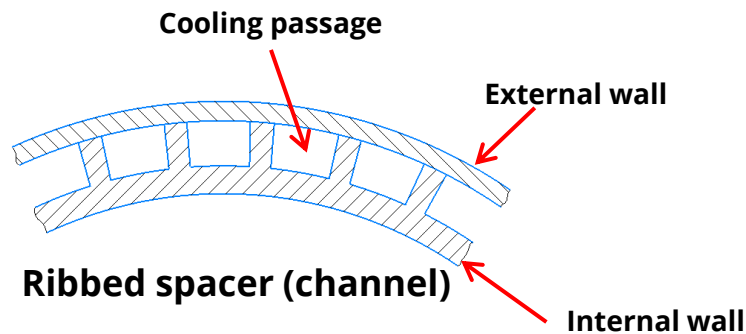


Outlet total temperature vs cooling mass flow rate

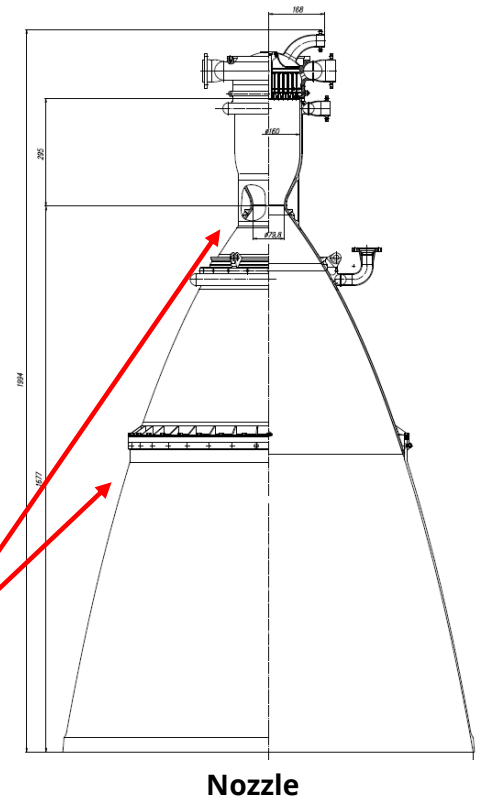
# Rocket Engine Nozzle Cooling

# Goals of Nozzle Cooling System Analysis

- **Objective:** Analyze nozzle cooling system with supercritical hydrogen
- Nozzle cooling system modeling with a phase transition and supercritical hydrogen parameters in the nozzle's cooling jacket
- Calculate wall temperatures and heat transfer coefficients using heat flux - boundary conditions for thermal analysis of wall materials
- Estimate cooling jacket outlet's boundary conditions for subsequent turbine design

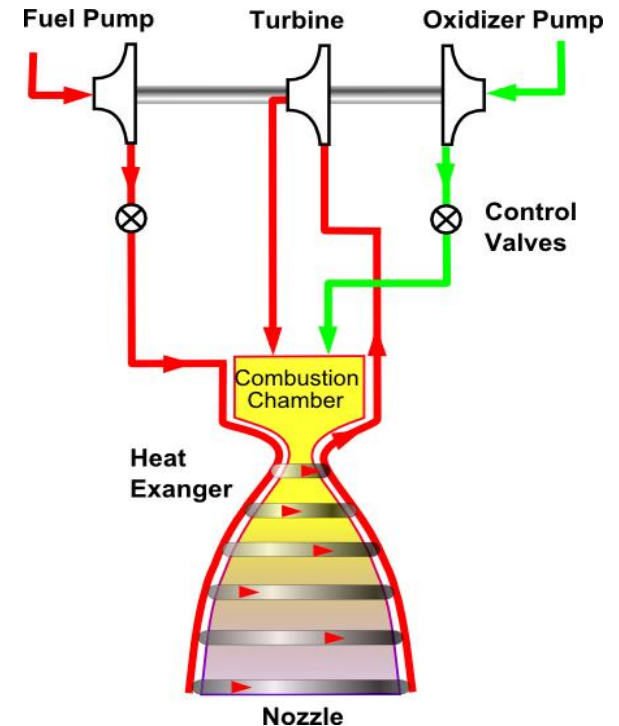


Nozzle jacket cooling with different cross sections



# Nozzle Cooling System Analysis Introduction

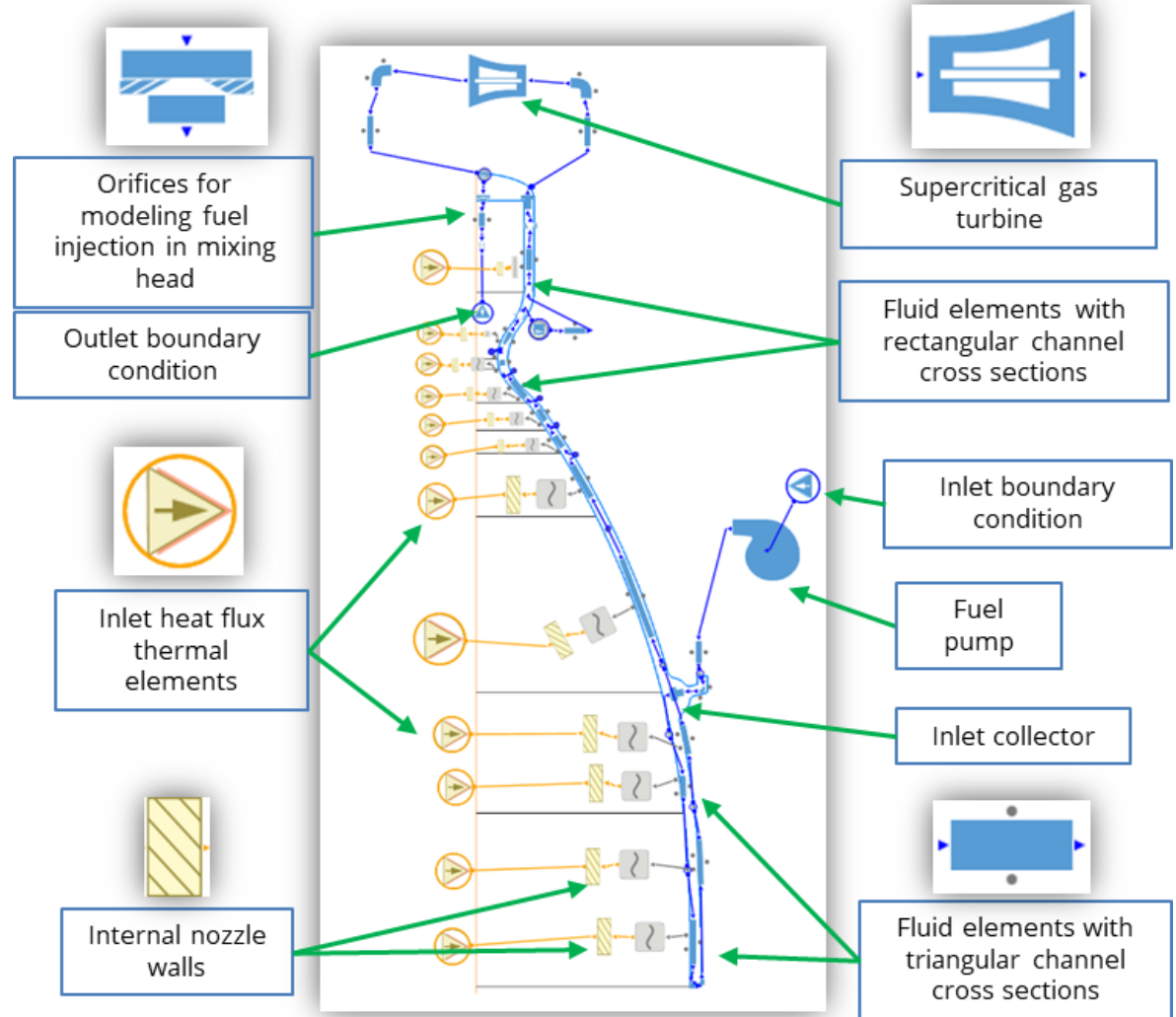
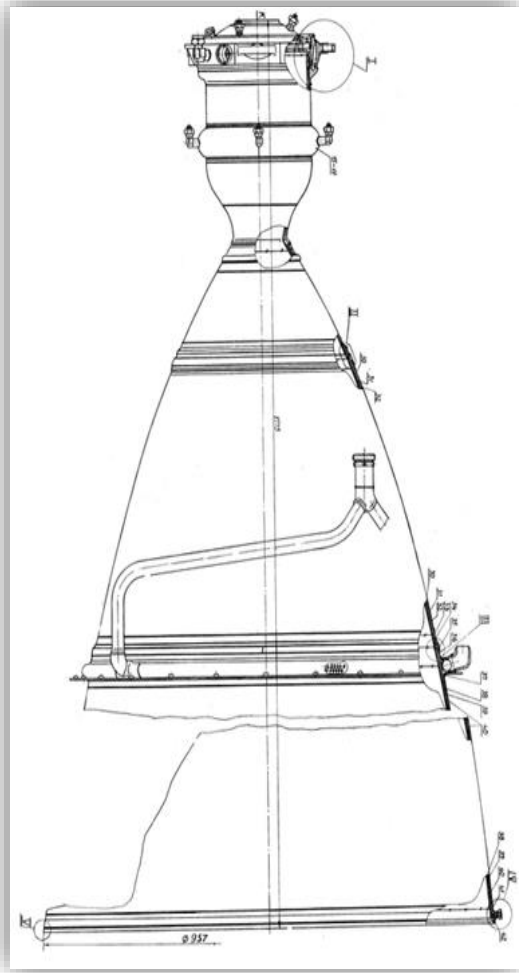
- Model of liquid rocket engine with expander cycle selected
- Liquid hydrogen flows through the fuel pump where it is compressed.
- After being compressed the supercritical hydrogen passes through the nozzle cooling jacket; the liquid hydrogen transitions to a gas state.
- After passing through the nozzle cooling jacket, the gaseous hydrogen flows through to the turbine that drives the liquid hydrogen pump.



**Expander cycle**

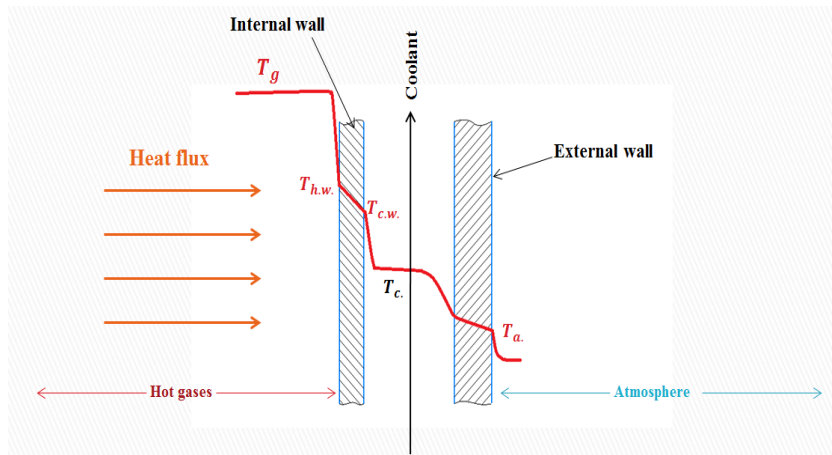
\*Open source - Wikipedia

# Modeled Cooling System

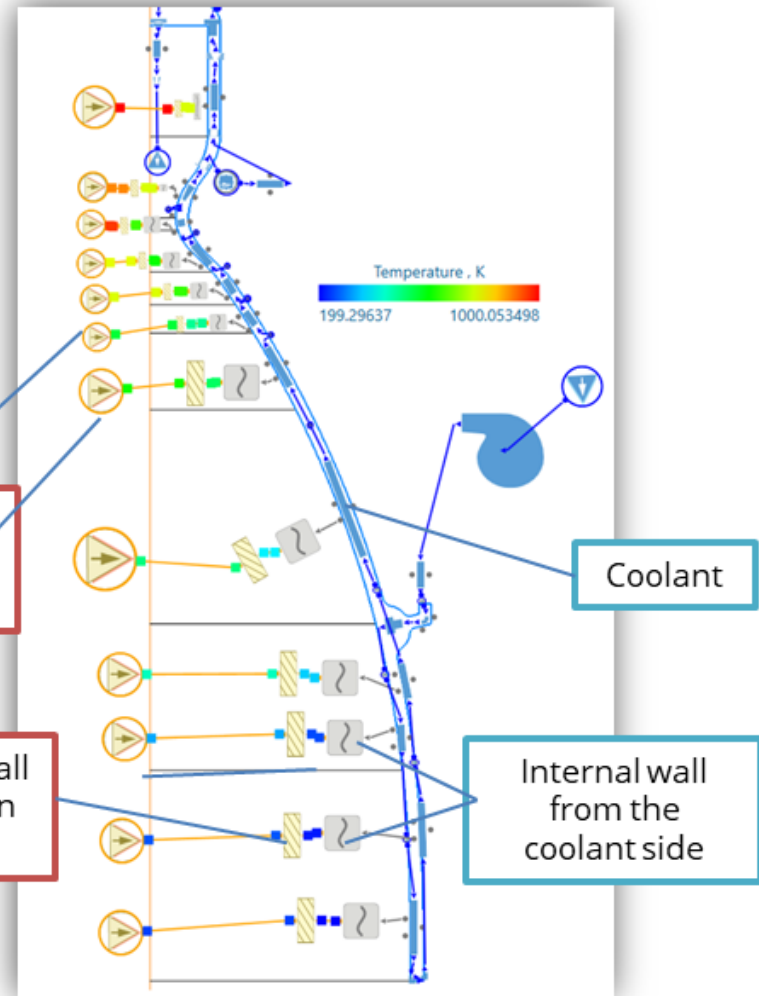


# Cooling System Analysis

## Wall temperature distribution plot



$T_g$  - Temperature of hot gases in combustion chamber  
 $T_{h.w}$  - Temperature of internal hot wall in combustion chamber  
 $T_{c.w}$  - Temperature of internal wall from coolant side  
 $T_c$  - Temperature of the coolant  
 $T_a$  - Temperature of the atmosphere



# Considerations for Integration & Automation

# Role of Integration & Automation

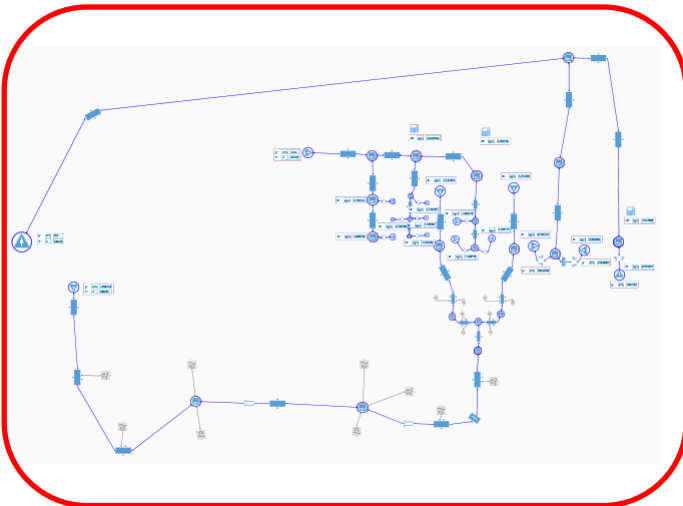
- Repetition of tasks
  - 1 time set up for several projects/applications (reports, performances ...)
  - Smooth workflow upon changes of project specifications, constraints, etc.
- Time-consuming tasks
  - Once task is set up results are 1 button away
  - Particularly useful for highly loaded human resources as they can focus on other projects in the meantime
- Uninterrupted processes
  - Run series of tasks overnight and/or during weekend without having to run sequential tasks manually
- Implementation of guidelines and constraints
  - Accounts for specific (industry standards, company policy, personal experience, etc.) “design” rules without human intervention
  - Fully customizable weighing of mission objectives (different priority criteria for efficiency, space and/or weight, cost of equipment including materials, etc.)
- What-if studies made even easier
  - Studies limited by computing resources instead of valuable engineering time
- Faster validation checks, turn-key overall & better competitive edge
- Improved quality & reliability

# Case Study - Gas Turbine Matching at Design & Off-Design Conditions

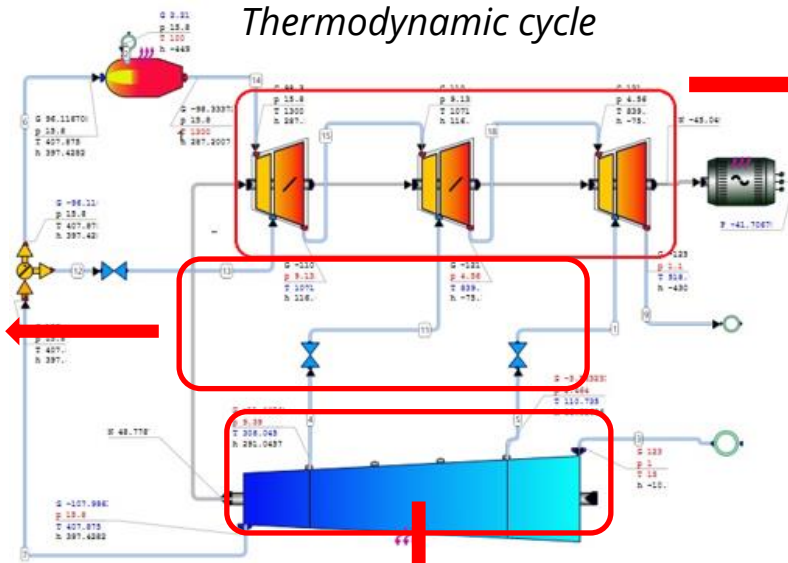
# Matching of Components

- **Task formulation:** Off-design performance estimation of 166.5 MW gas turbine for different values of ambient temperature from 5C to 30C

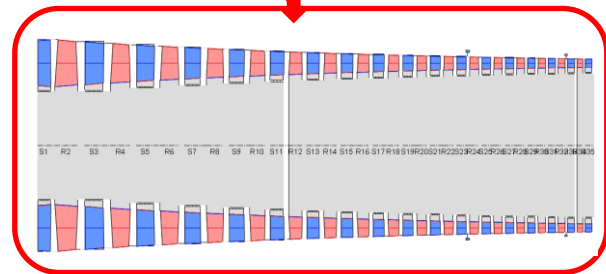
Cooling system



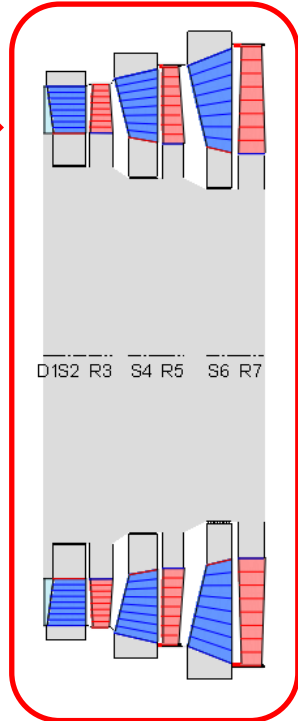
Thermodynamic cycle



Compressor



Turbine



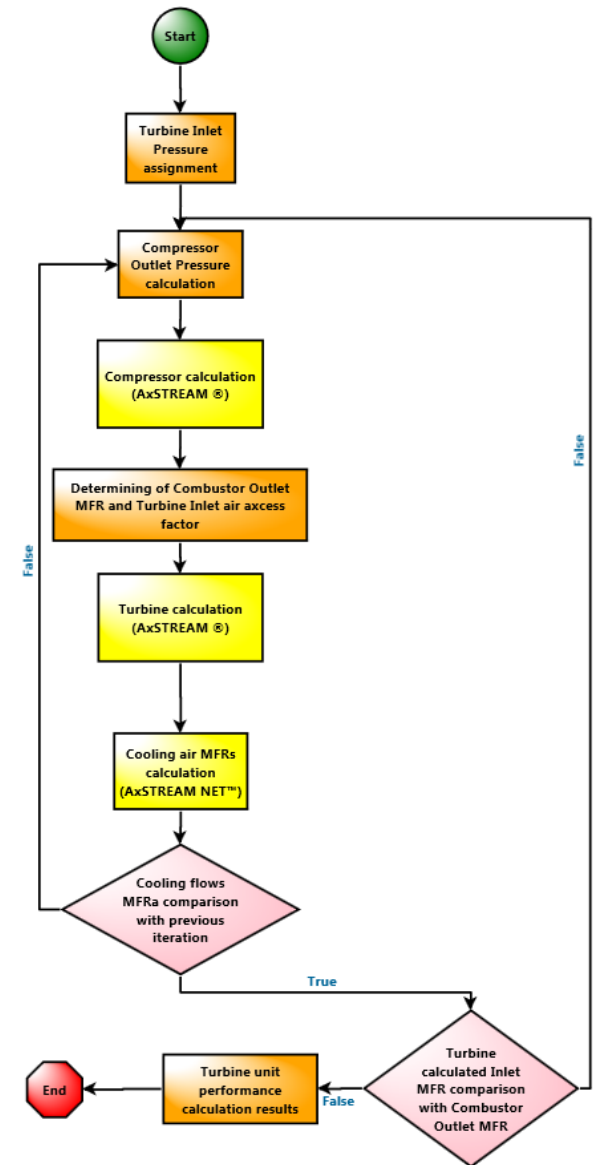
# Matching Task

## ➤ Used blocks and modules:

- Streamline solver for compressor and turbine
- Cooling system calculation
- **Condition** block for logical operation
- **Script** for combustor calculation

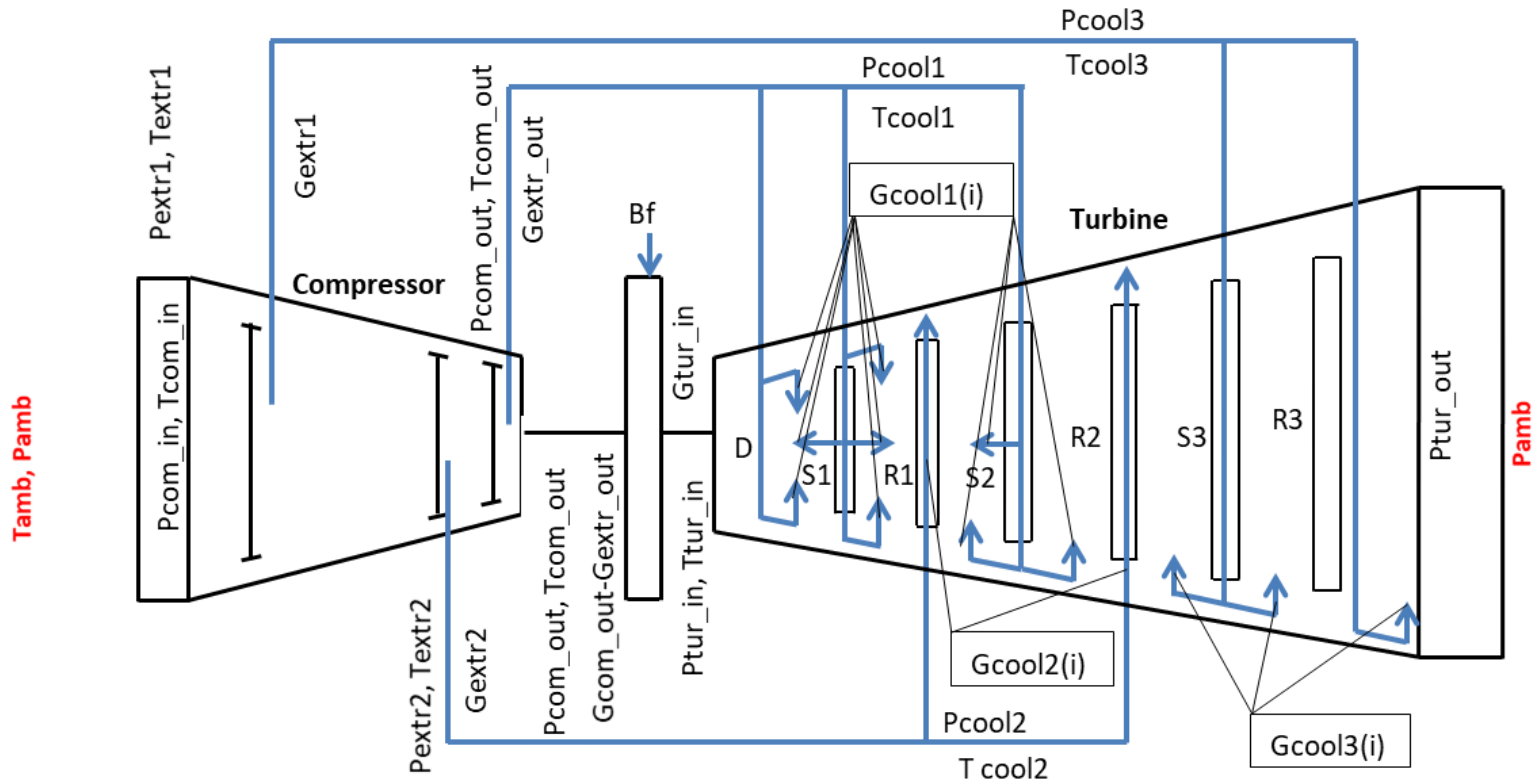
## ➤ Automatic process help:

- Eliminate manual work and human error
- Significantly decrease time to determine characteristic of gas turbine
- Perform optimization of cooling air during different working conditions
- Etc.



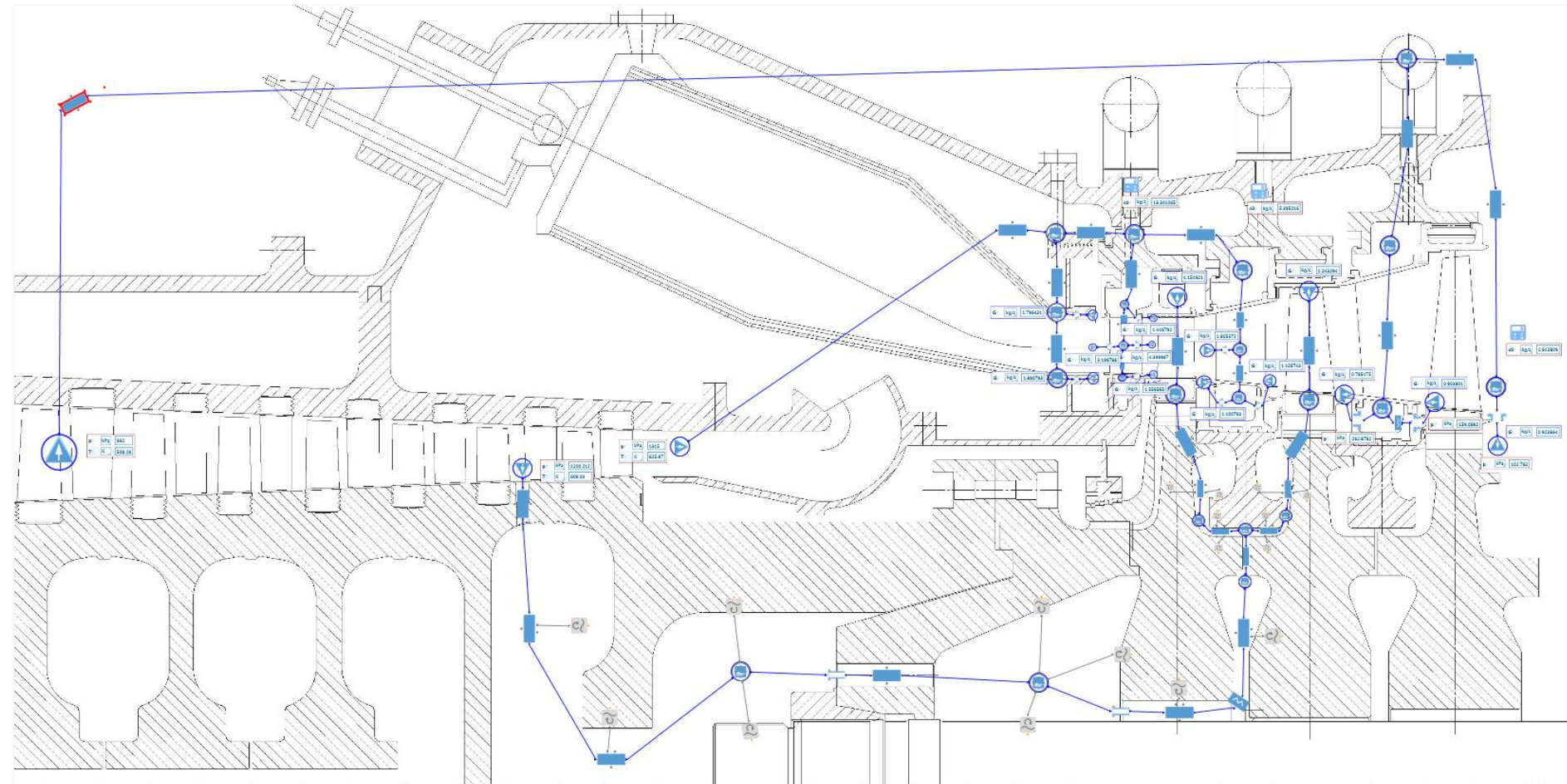
# Schematic Cooling System

- Cooling system is presented by three extraction from compressor that separated on 14 turbine cooling flows. There we have leading and trailing edge cooling, hub and shroud cooling and tip cooling for rotor blades.



# Cooling System

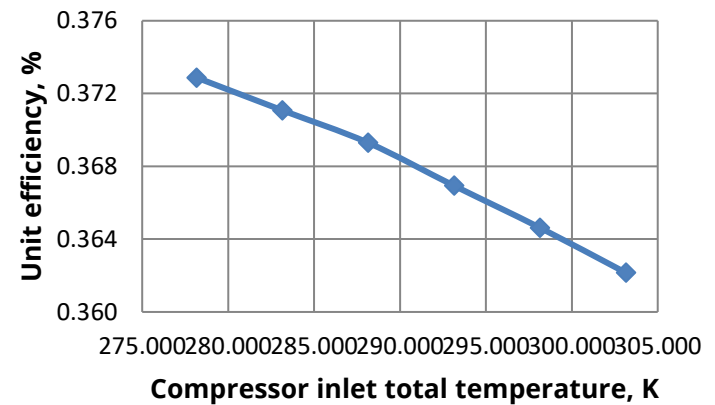
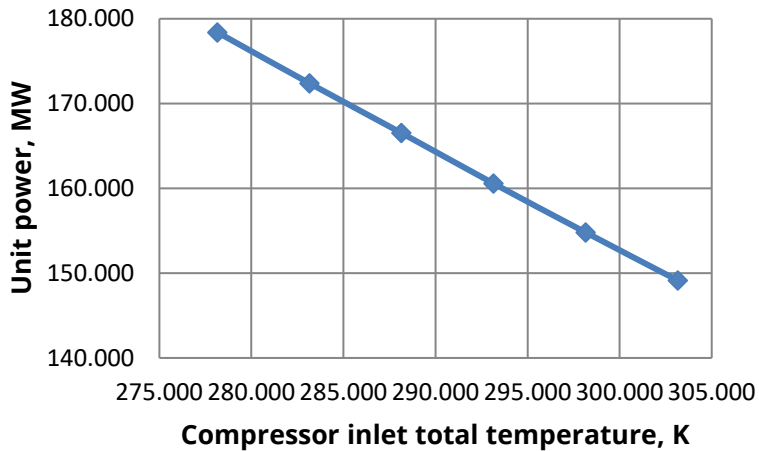
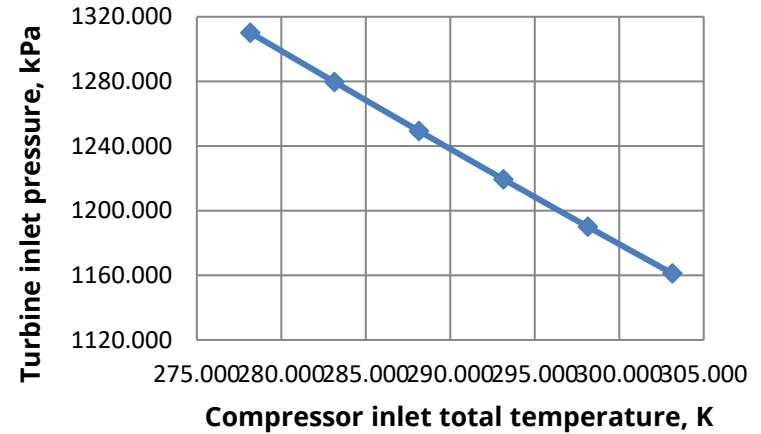
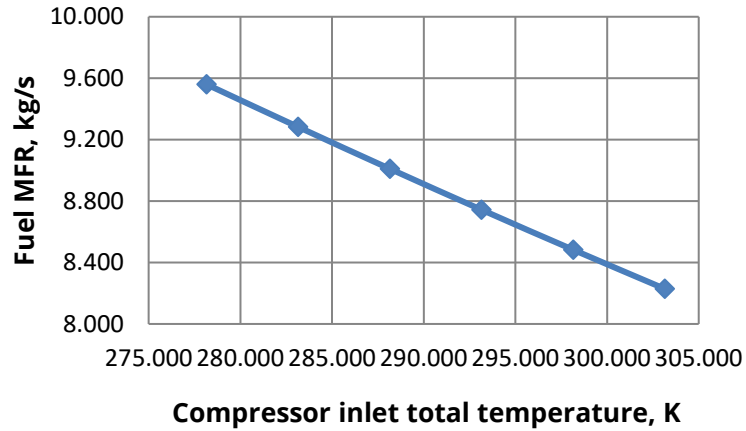
- Cooling flows MFRs calculation at design and off-design.



# Results

| Parameters   | Units | Values |       |              |       |       |       |
|--|-------|--------|-------|--------------|-------|-------|-------|
| <b>Compressor inlet temperature</b>                            | K     | 278    | 283   | <b>288</b>   | 293   | 298   | 303   |
| <b>Turbine inlet pressure</b>                                  | kPa   | 1310   | 1279  | <b>1249</b>  | 1219  | 1190  | 1161  |
| <b>Unit shaft power</b>  | MW    | 178.4  | 172.4 | <b>166.5</b> | 160.5 | 154.8 | 149.1 |
| <b>Fuel mass flow rate</b>                                     | kg/s  | 9.559  | 9.282 | <b>9.009</b> | 8.743 | 8.482 | 8.228 |
| <b>Unit efficiency (w/o bearings and generator efficiency)</b> | %     | 37.3   | 37.1  | <b>36.9</b>  | 36.7  | 36.5  | 36.2  |
| <b>Turbine outlet temperature</b>                              | K     | 898    | 900   | <b>902</b>   | 904   | 906   | 908   |
| <b>Compressor outlet temperature</b>                           | K     | 613    | 620   | <b>626</b>   | 632   | 638   | 644   |
| <b>Compressor outlet pressure</b>                              | kPa   | 1379   | 1347  | <b>1315</b>  | 1284  | 1253  | 1222  |
| <b>Cooling system MFR</b>                                      | kg/s  | 27.72  | 27.01 | <b>26.31</b> | 25.63 | 24.95 | 24.29 |

# Results



# Case Study - Modeling of 50 MW Gas Turbine with Anti-icing System

# Purpose of Design Process

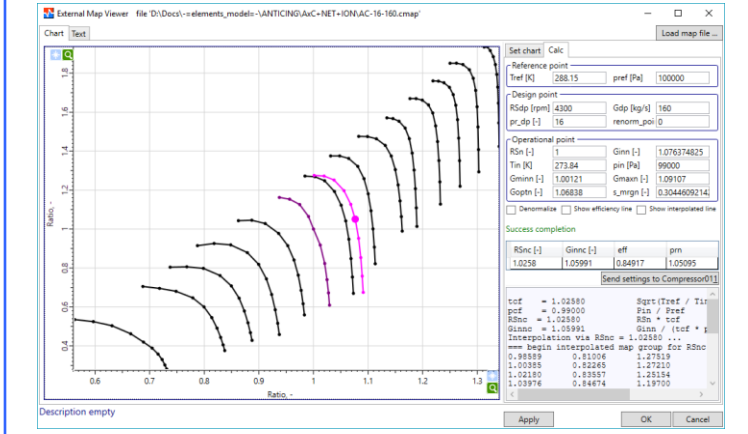
- Design **optimal and reliable anti-icing system (AIS)** for different ambient air parameters and different GT modes of operation.
- Criterion of scheme optimality is **overall efficiency of GT cycle**.
- Criterion of scheme reliable is **absence of intake icing**.
- Different ambient air parameters:
  - ✓ temperature,
  - ✓ humidity,
  - ✓ water content (fog and mist).
- Different GT modes mean that all cycle parameters can be changed if required electrical output power changes.

# Required activity

- So that to ensure maximum GT cycle efficiency and at the same time ensure the reliability of GT unit should realize 3 points:
  1. **Choose AIS inclusion scheme.**
  2. **Determine required heating air mass flow** to reduce intake air humidity to the required value to prevent icing.
  3. **Design heating air supply system** - determinate geometric sizes of bypass pipeline or the damper opening level to ensure conduction of required air flow.
- Iterations between steps 2 & 3 are required to find an optimal solution to the system

# 50 MW GT thermodynamic cycle

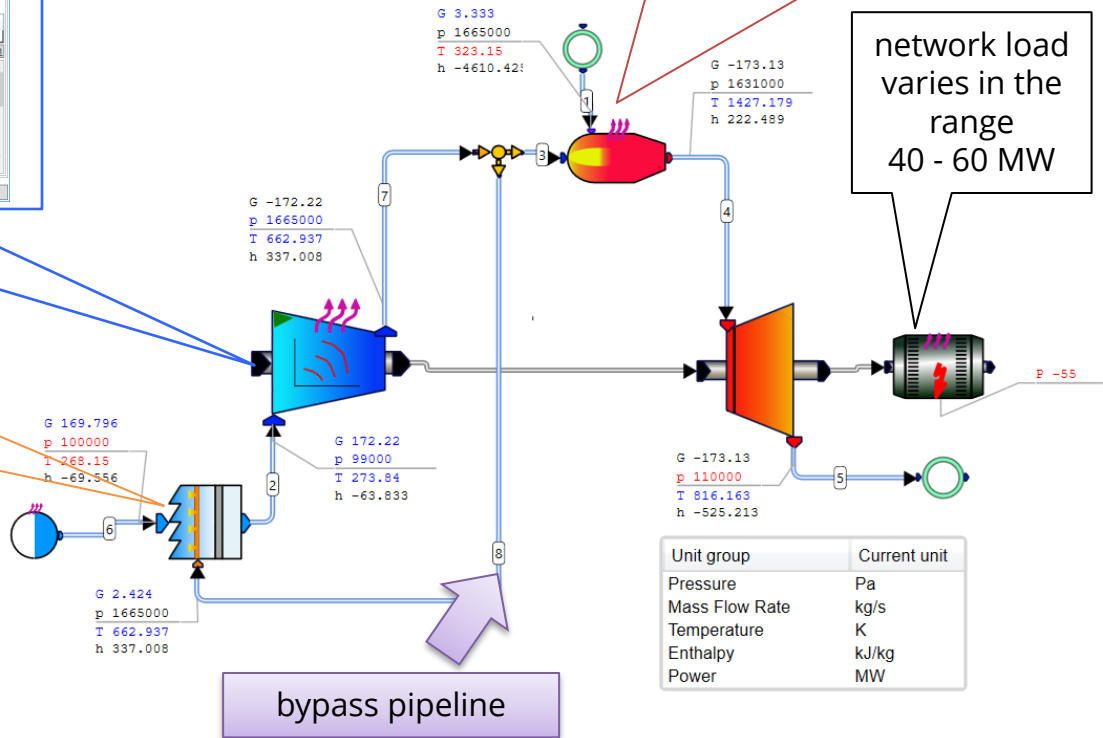
Compressor in off-design modes is specified using External Map



Regulation of the plant's power is implemented by changing fuel consumption in the combustion chamber. At unchanged parameters of the outside air, when the required power of the generator decreases, gases temperature after combustion chamber and air pressure behind the compressor decrease

ANTIICING component calculates the flow of hot air which ensures air overheating above the dew-point temperature (at a static pressure in the narrow section of the channel) 6°C

Outside air inflow to the GT has a humidity 80 - 100% and temperature -14 - +4 °C

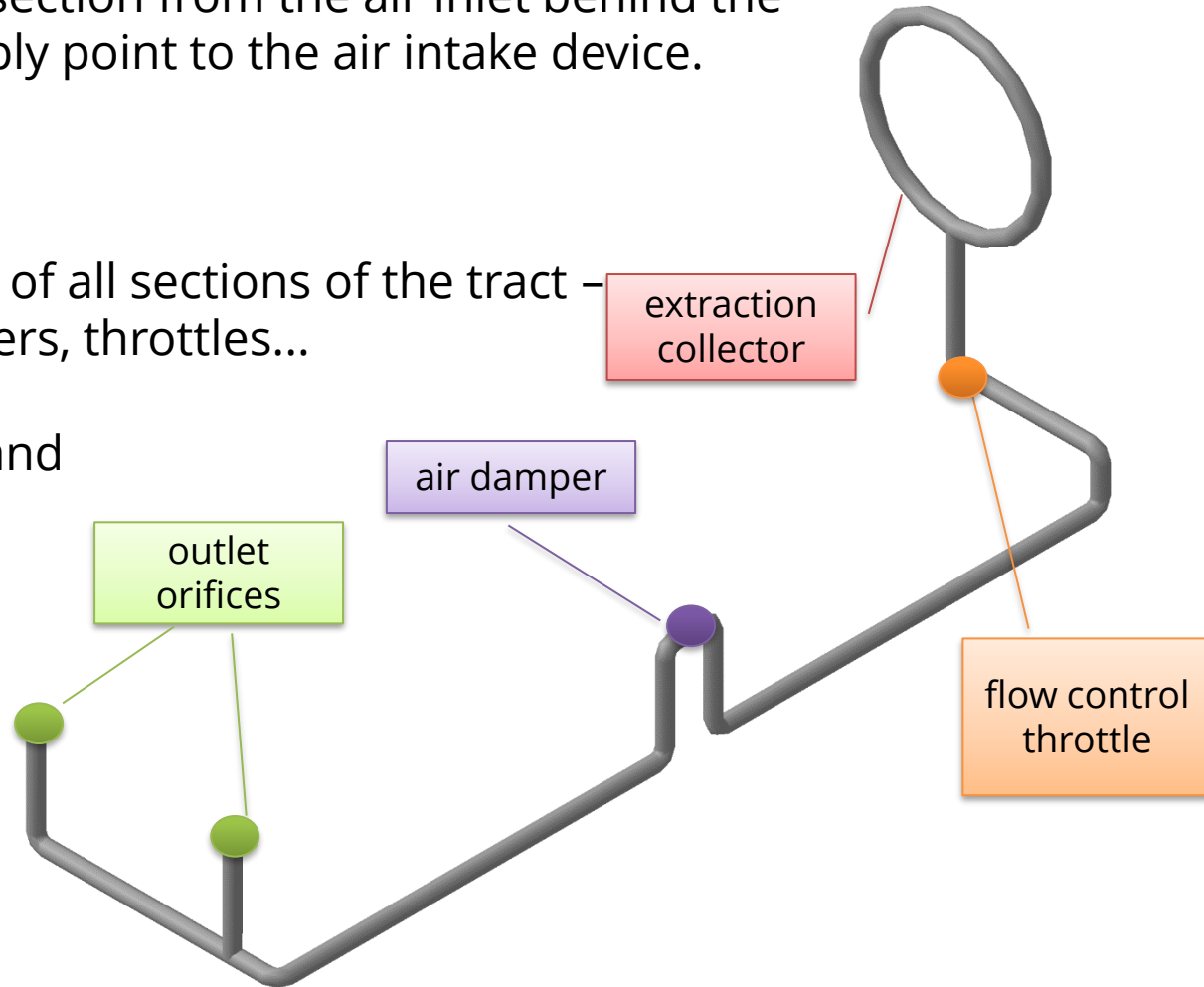


# Bypass pipeline configuration

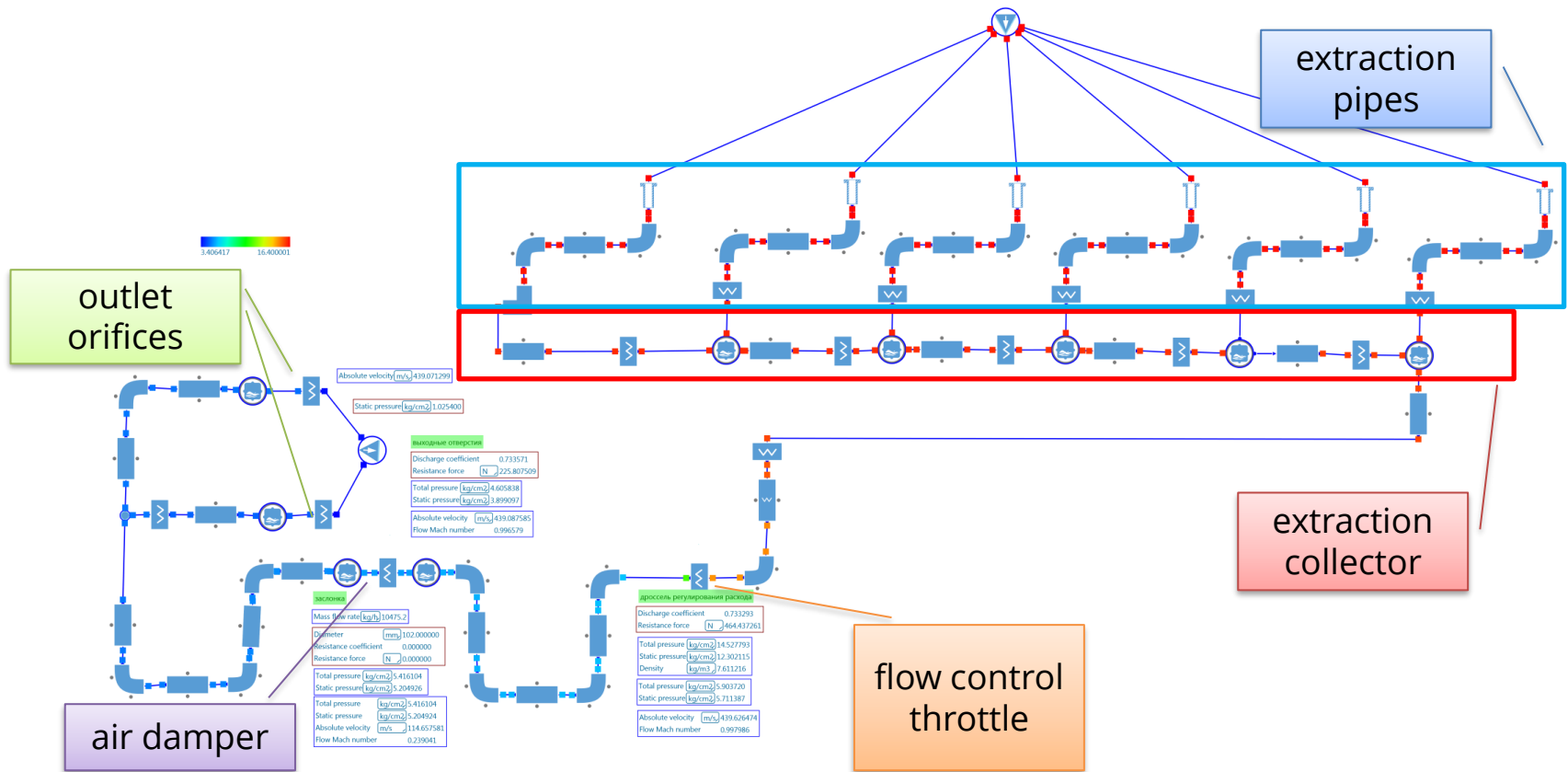
- Simulates the pipeline section from the air inlet behind the compressor to the supply point to the air intake device.
- The initial data are:

1. Geometric parameters of all sections of the tract – diameters, lengths, dumpers, throttles...

2. Thermodynamic inlet and outlet parameters derived from cycle calc.



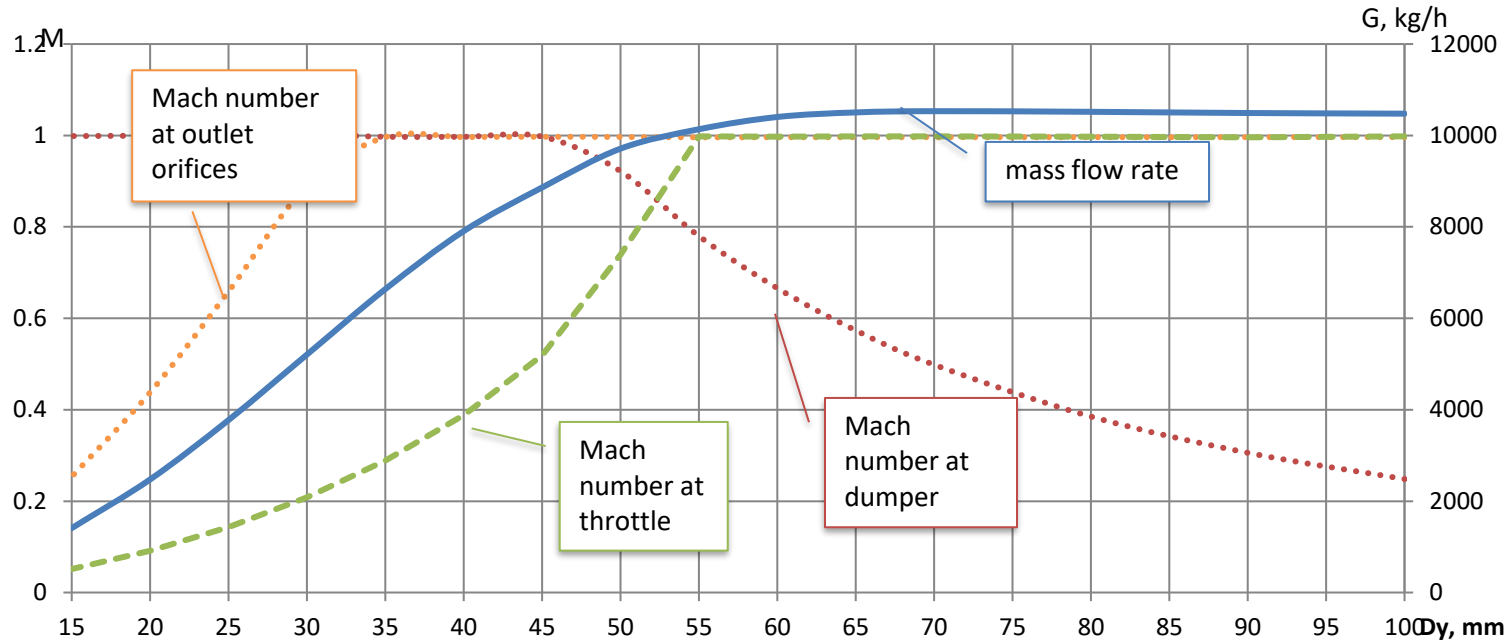
# Pipeline hydraulic network



Bypass pipeline modeling

# Bypass pipeline choking analysis

## Task 1. Analysis of locking and determination of maximum mass flow rate at designing of bypass pipeline

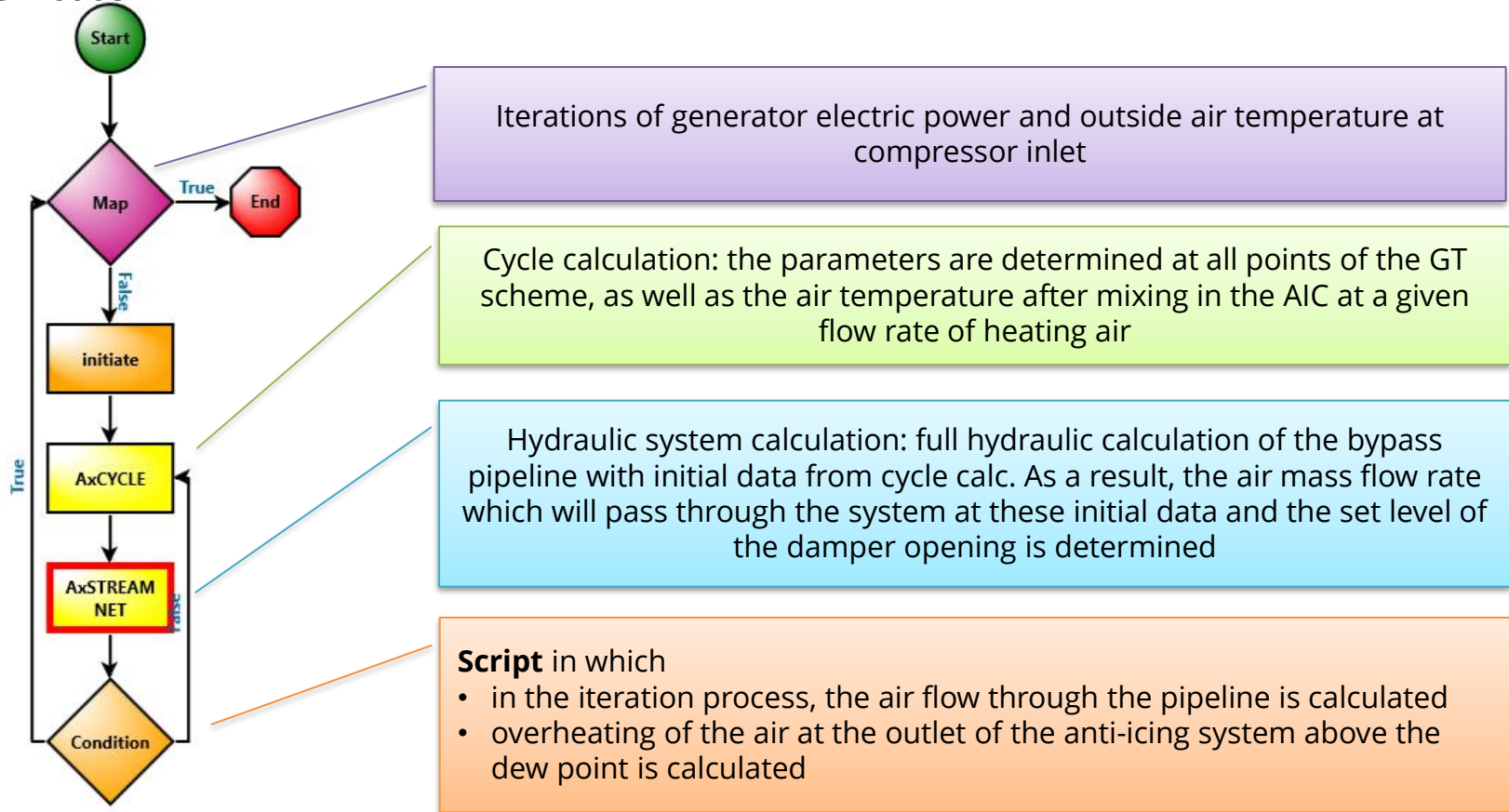


Change in the pipeline parameters at a different level of the damper opening at the nominal GT operating mode (with a fixed throttle cross-section area)

# Investigation of reliability AIS

## Task 1. Evaluation of different operating conditions to find when appearance icing

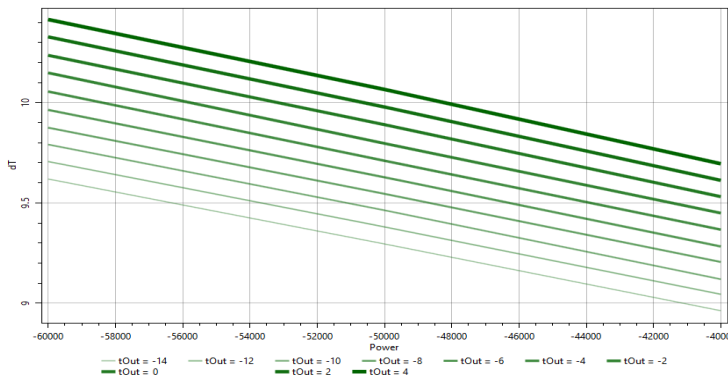
System analysis with a fixed control damper position for various parameters of outside air and GT loads



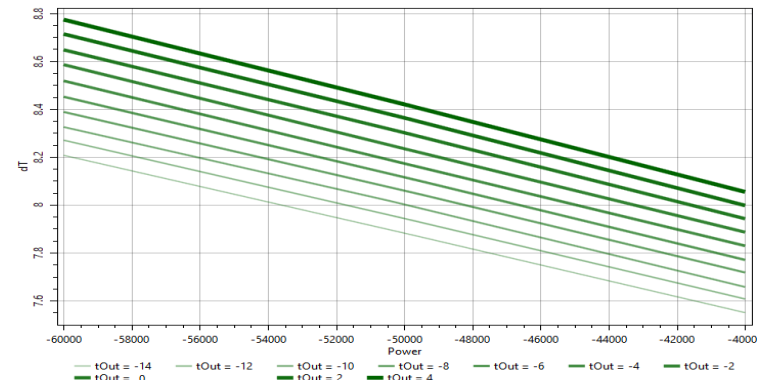
# Results: reliability AIS

## Air overheating above dew point temperature with fully open damper when conditions change

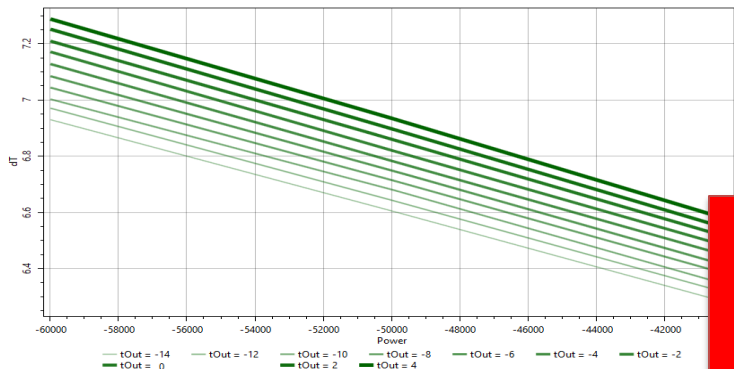
at ambient humidity of 80%



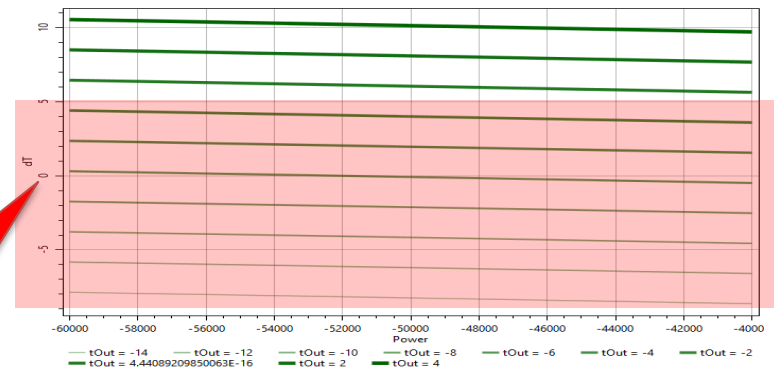
at ambient humidity of 90%



at ambient humidity of 100%



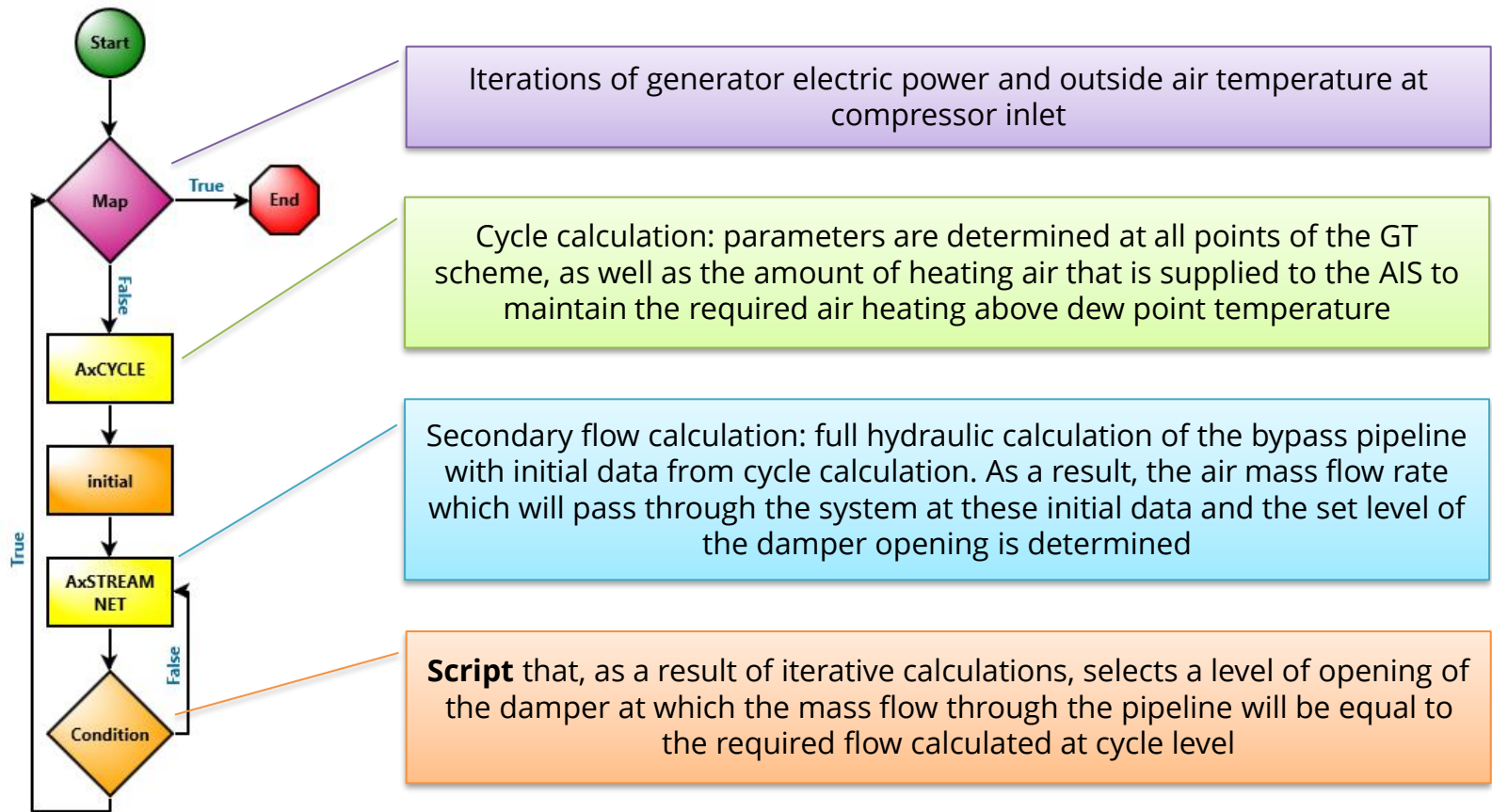
at ambient humidity of 100% and water content 0,1 g/m<sup>3</sup>



area in which icing is possible

# AIS valve regulation law finding

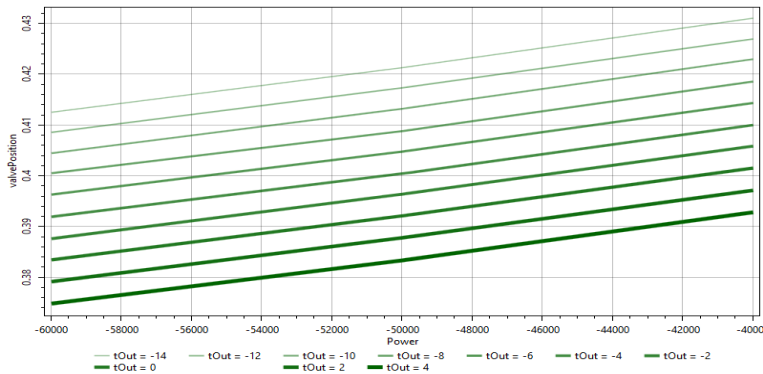
**Task 2. Determine damper opening to satisfied minimum mass flow rate for various parameters of outside air and GT loads**



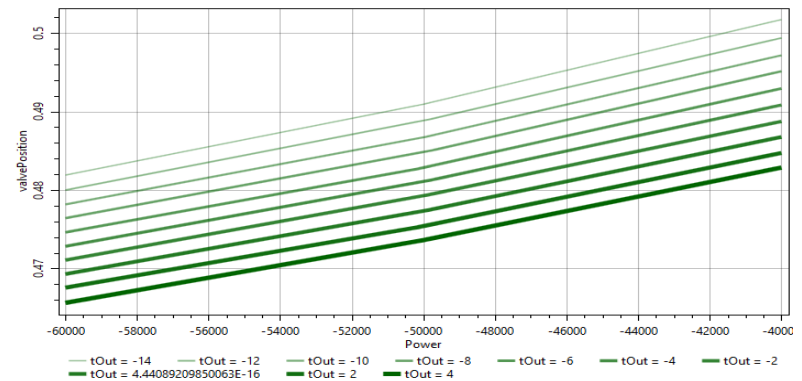
# Results: AIS valve regulation law

## Required level of opening of the damper when conditions change

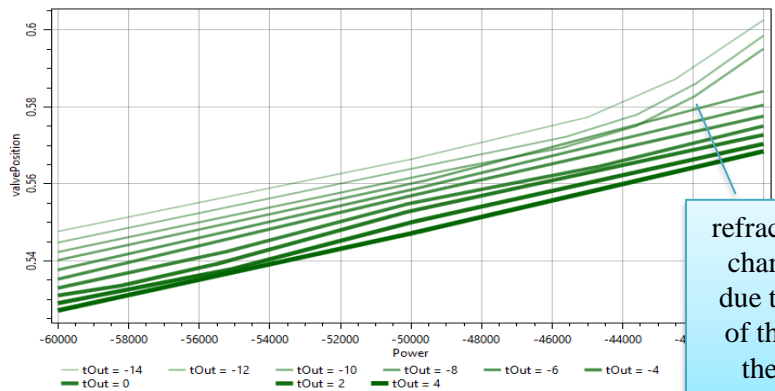
at ambient humidity of 80%



at ambient humidity of 90%



at ambient humidity of 100%



refraction of the characteristic due to the start of the lock on the throttle

# Case Study - Aircraft Fuel System

# Modeling and Analyzing Aircraft Fuel System

## Goals:

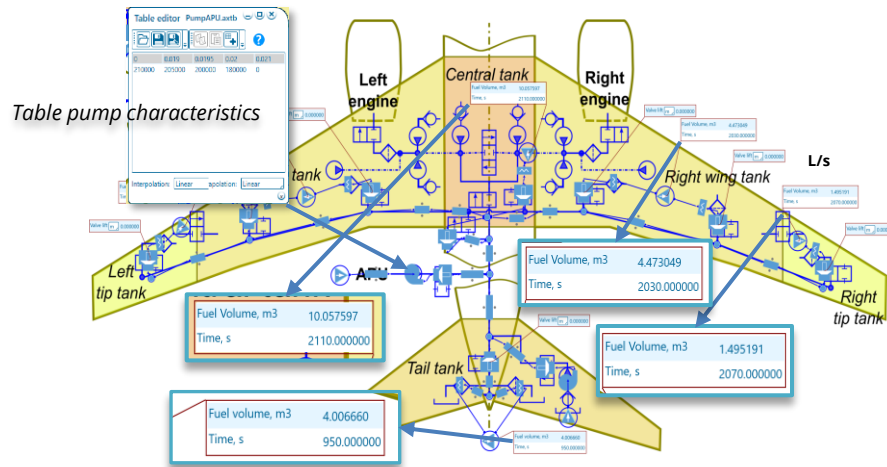
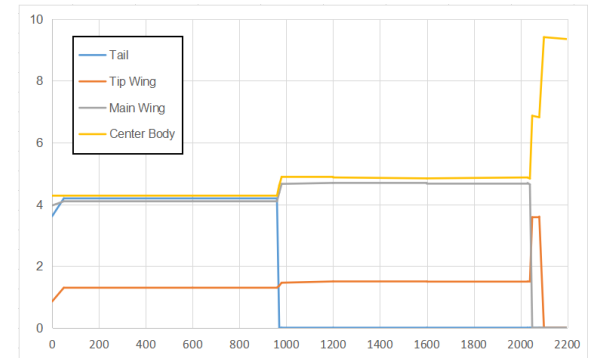
Creation of digital twin of fuel system for two-engine aircraft including APU pump, valve and tank characteristics

Determination of refueling time (unsteady regime)

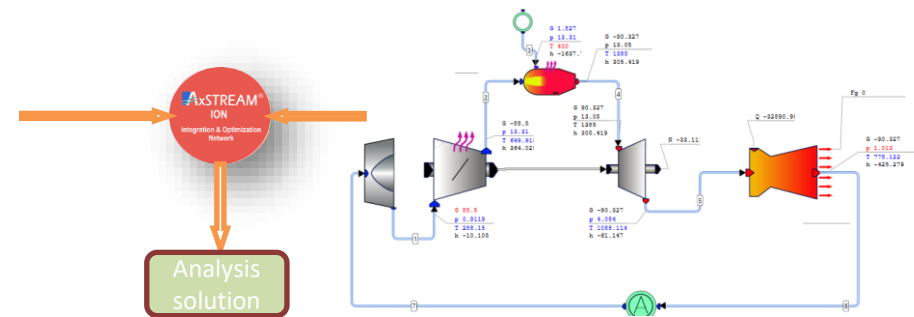
## Tasks:

- Determining mass flow rates through the pipelines.
- Calculating pressure distribution at steady state mode and select APU pump.
- Simulating unsteady state mode and refueling duration determination.

Volume flow rate vs. time



Thermal model of a gas turbine engine



# System Characteristics & Initial Data

- Required refueling time < 40 min for 26 m<sup>3</sup> of total tanks volume (650 L/min or more)
- Refueling time depends on pipe diameters, pressure losses, APU pump characteristics, tank volumes and location, etc.
- Fuel used is JP-7
- Tank pressurization to 2.5 bar
- Arbitrary gear pump performance maps taken from Parker website
- Performance maps can be edited to account for phenomena like aging of equipment
- Effective pipe diameter and/or surface roughness can be modified to simulate deposit buildup which both increase flow resistance and therefore require more pumping power
  - Life cycle analysis (digital thread)

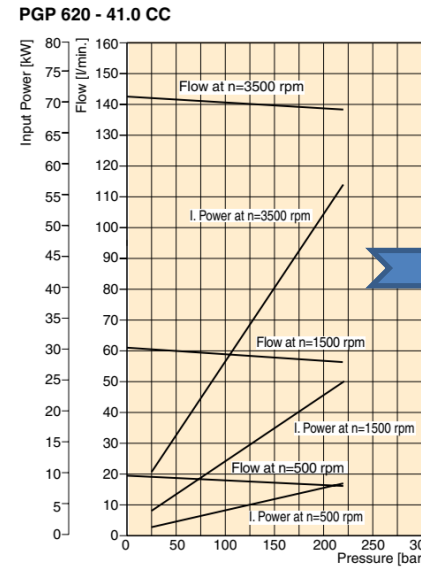


Table editor PGP620\_n.axtb

|           | 1500     | 3500     |
|-----------|----------|----------|
| 0         | 25000000 | 25000000 |
| 0.0009333 | 22000000 | 23783000 |
| 0.0010167 | 0        | 23674000 |
| 0.0023    | 0        | 22000000 |
| 0.0023833 | 0        | 0        |

**Pressure rise**

Interpolation:  Extrapolation:

Table editor PGP620\_n.Paxtb

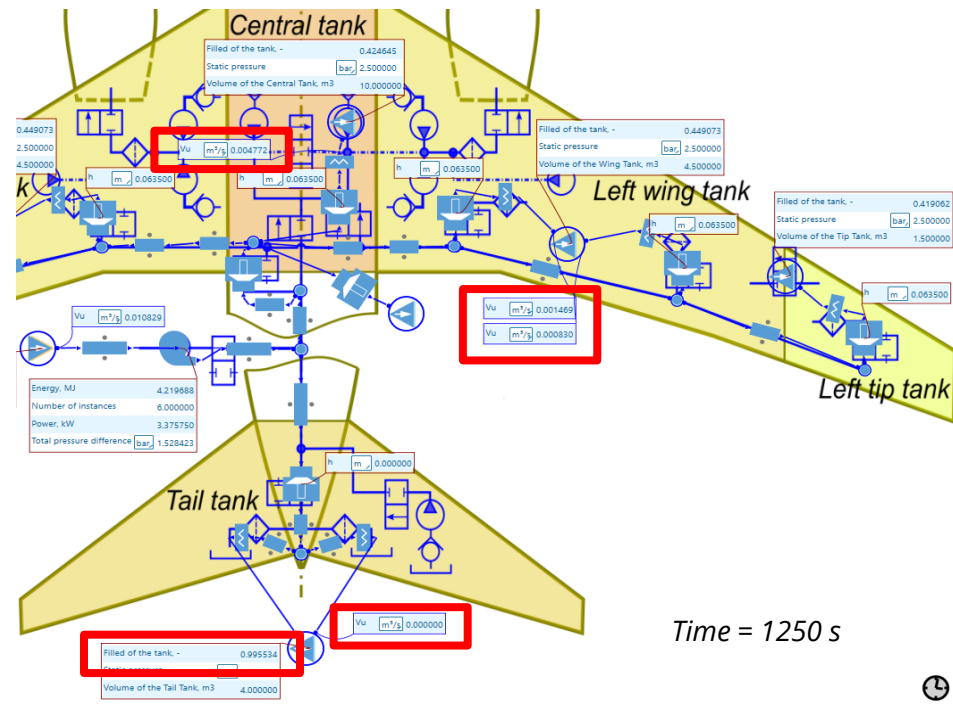
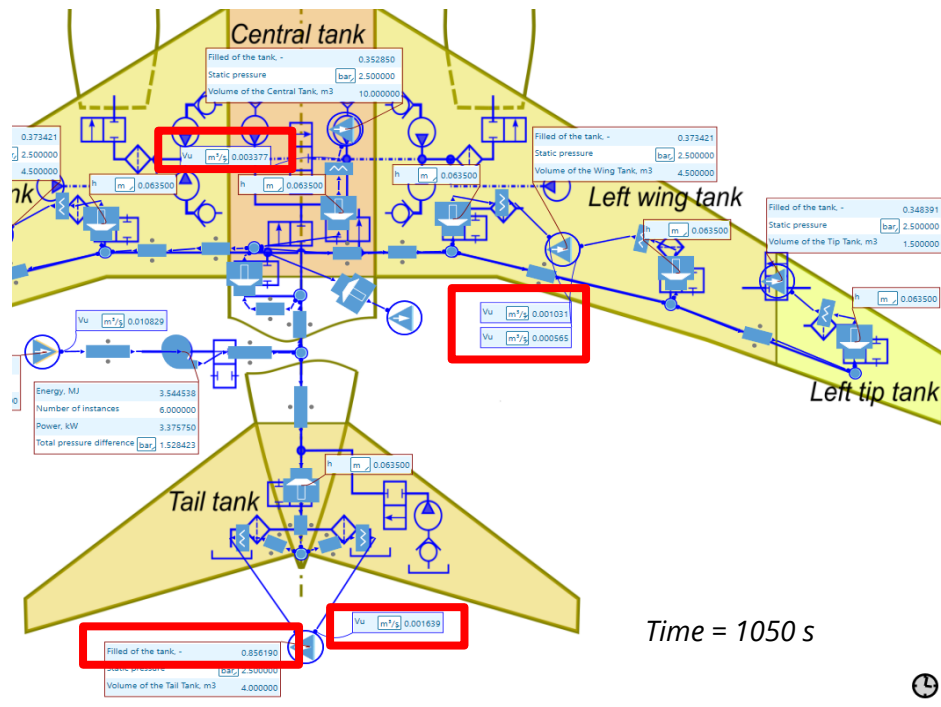
|           | 1500 | 3500     |
|-----------|------|----------|
| 0         | 28.4 | 25000000 |
| 0.0009333 | 25   | 61.6196  |
| 0.0010167 | 3    | 61.337   |
| 0.0023    | 3    | 57       |
| 0.0023833 | 3    | 0        |

**Power**

Interpolation:  Extrapolation:

# Refueling Process

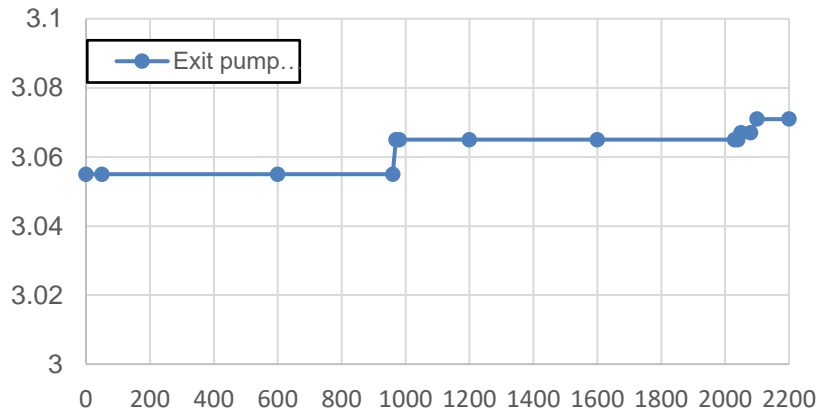
- Transient calculation
- Valves automatically shuts down flow going to tanks that are already full



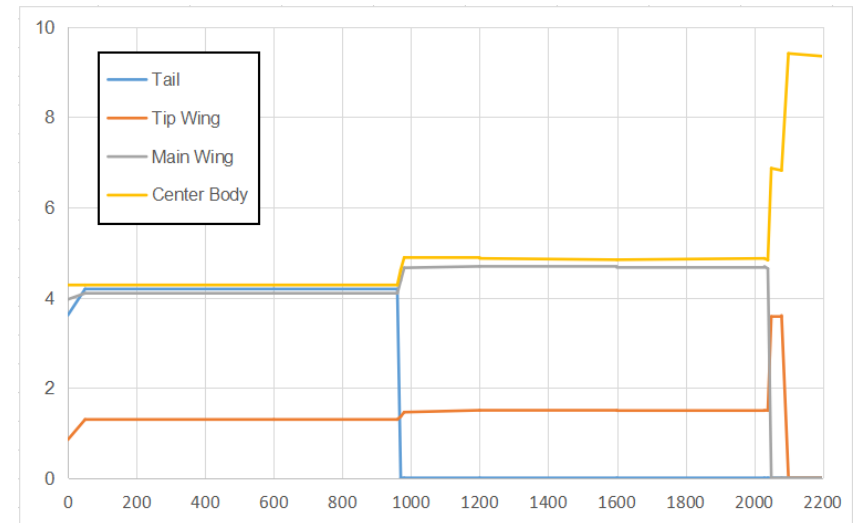
# Refueling Results

- Tail tank filled first at ~ 17 min
- Next, wing tanks filled at ~34 min
- Central tank filled last ~37 min
  - Total time close to the requirement
- Pressure changes in system due to valve shut-offs shown below

| Tank         | Volume, m <sup>3</sup> |
|--------------|------------------------|
| Center       | 10.057                 |
| Inner (Wing) | 2 x 4.473              |
| Outer (Tip)  | 2 x 1.4952             |
| Trim (Tail)  | 4.0066                 |
| <b>Total</b> | <b>26.022</b>          |

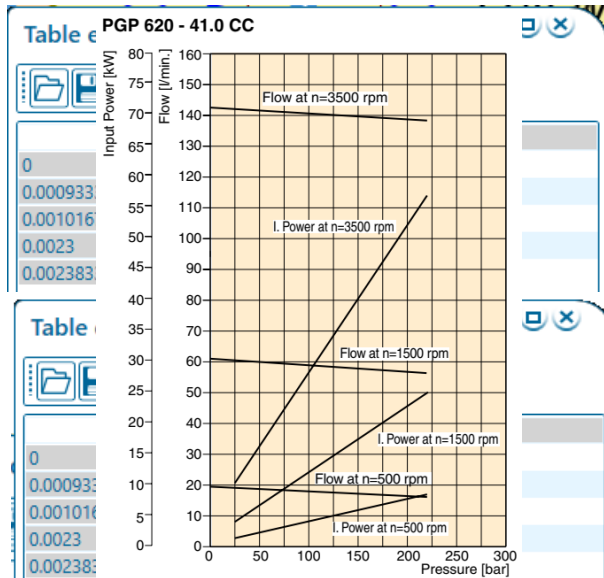


*Evolution of system pressure with time*



*Volume flow rate evolution with time*

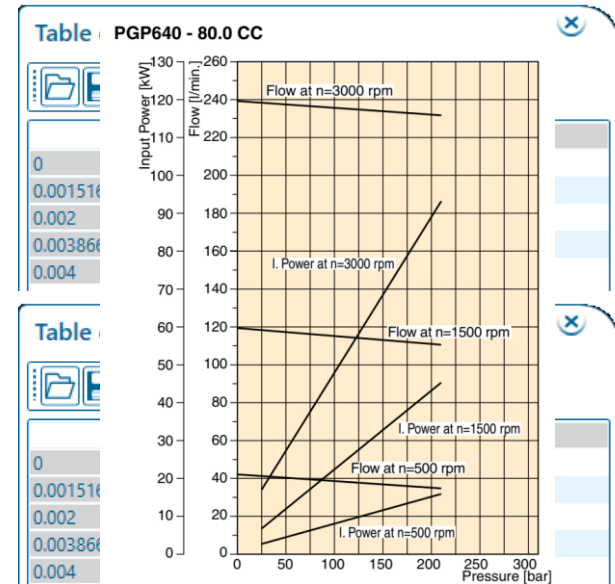
# Pumps Selection & Upgrade



Power (top) & pressure rise (bottom)

- 2 different pumps were investigated
  - PGP620 (left)
  - PGP640 (right)

- Allows simulating tasks such as
  - Equipment replacement
  - Component upgrade



Power (top) & pressure rise (bottom)

# Integration with Gas Turbine Engine

- **Goal:** Accurate integrated system-level modelling of engine and fuel delivery system for any operating conditions to determine pipeline pressure and temperature changes as well as required fuel flow rate to provide desired engine net power
- **How:** Link thermal-fluid network fuel delivery system model to a thermodynamic cycle off-design thermodynamic engine model thanks to an integration software

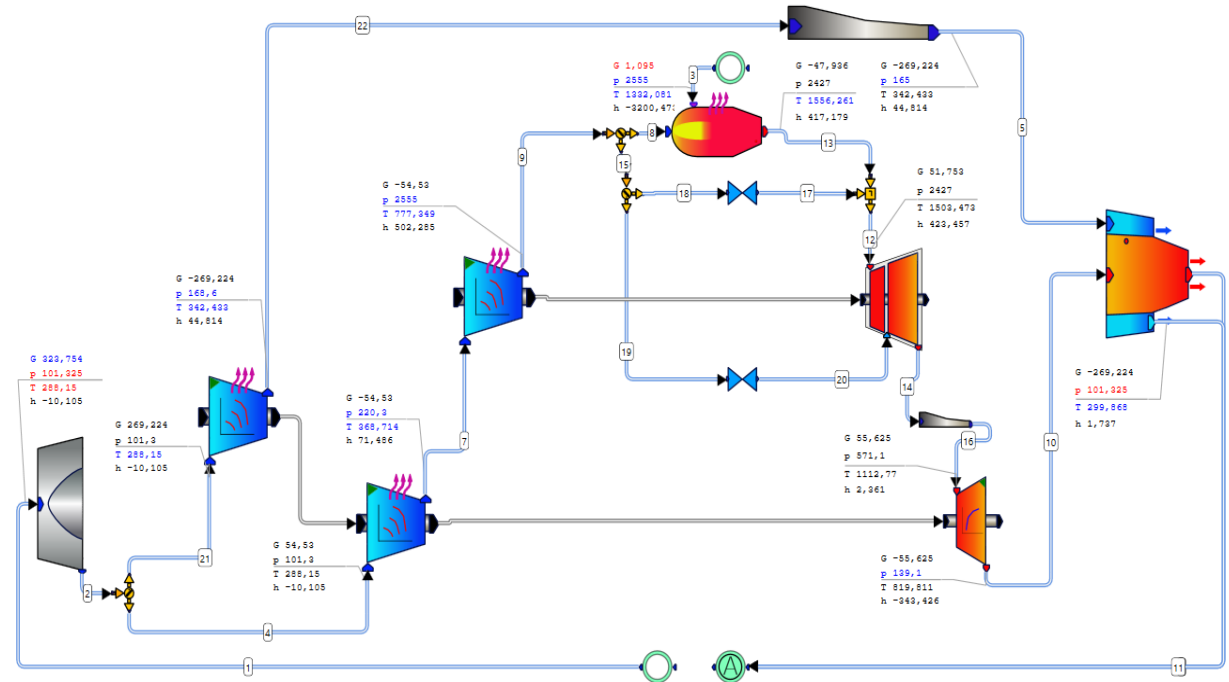
Workflow principle for models integration

|         | Thermodynamic cycle calculation tool  | Thermal-fluid network modelling tool   |
|---------|---|--|
| Inputs  | <ul style="list-style-type: none"> <li>• Altitude - fixed</li> <li>• Mach number - fixed</li> <li>• Fuel MFR - guessed</li> </ul> | <ul style="list-style-type: none"> <li>• Fuel tank temperature - fixed</li> <li>• Fuel tank pressure - fixed</li> <li>• Fuel injection <b>static</b> pressure - guessed</li> </ul> |
| Outputs | <ul style="list-style-type: none"> <li>• Fuel injection <b>total</b> pressure</li> </ul>  | <ul style="list-style-type: none"> <li>• Fuel MFR</li> </ul>   |



# Thermodynamic Cycle Modeling of Engine

- Off-design model
- Fixed parameters:
  - Altitude
  - Mach number
  - Fuel mass flow rate
- Injection total pressure compute

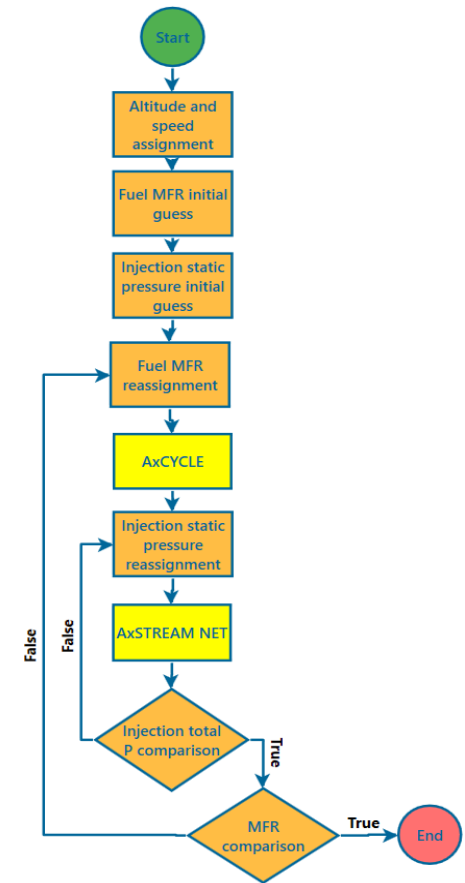


Thermodynamic model of considered engine

# Integration & Automation Workflow

Workflow blocks description

| Block                                   | Type       | Description   |
|---|------------|---|
| Altitude and speed assignment           | Script     | Computation of the intake total pressure and temperature from the altitude and the Mach number  |
| Fuel MFR initial guess                  | Script     | First assignment of fuel MFR for the cycle model in order to initiate the loop  |
| Injection static pressure initial guess | Script     | First assignment of injection static pressure for the thermal-fluid model in order to initiate the loop   |
| Fuel MFR reassignment                   | Script     | Reassignment of the fuel MFR value according to the comparison block  |
| AxCYCLE                                 | Process    | Run of the cycle model  |
| Injection static pressure reassignment  | Script     | Reassignment of the injection static pressure according to the comparison block   |
| AxSTREAM NET                            | Process    | Run of the thermal-fluid model  |
| Injection total pressure comparison     | Comparison | Adjust the injection static pressure of the thermal-fluid model until the total pressure is equal to that of the cycle model. By default, the margin is 0.01 bar and the step varies from 1 bar to 0.01 bar as the gap reduces. |
| Fuel MFR comparison                     | Comparison | Assign the fuel MFR computed by thermal-fluid software to cycle calculation tool until the values are equal   |



Integration model diagram

# Conclusions

# Conclusions

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- In this tutorial of basics session we have seen the basics of modeling secondary flows in turbomachinery systems including
  - What kind of elements are used and how they are modeled
  - What seals are and how leakages are important to accurately evaluate system performance
  - The different heat transfer methods and how they are calculated
  - The importance of cooling in gas turbines and what kind of physical phenomena are involved
  - The interest of integrated systems for off-design calculations
- The different topics were presented along with some visual examples to help understand the principles shown
- Case studies were showcased to bring real-life examples of applications for such calculations and their use

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