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CONTEXT

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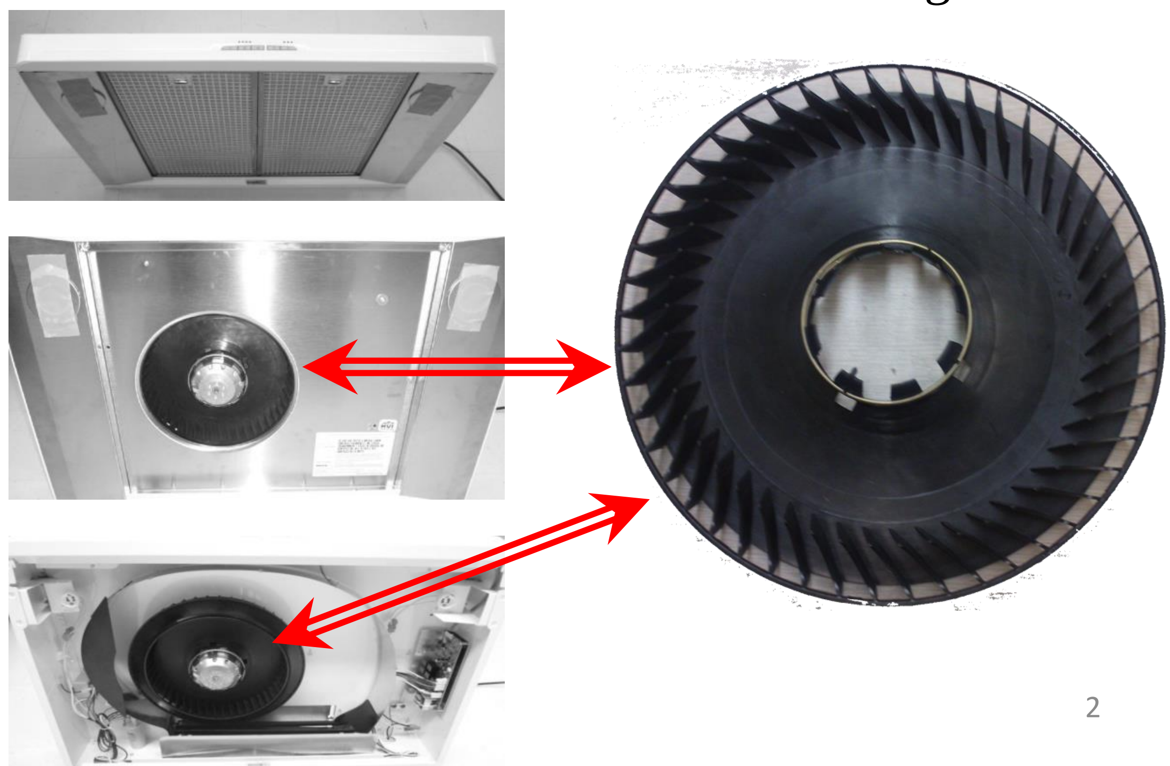
GROUPE D'ACOUSTIQUE DE L'UNIVERSITÉ DE SHERBROOKE

- Control wideband noise of centrifugal fan used in kitchen hood
- Noise produced by interaction between turbulent subsonic flow and blade
- Noise can be reduced by appropriate geometry of the profile
- No general solution. Each fan needs a specific design.

Kitchen

Kitchen hood

Centrifugal fan



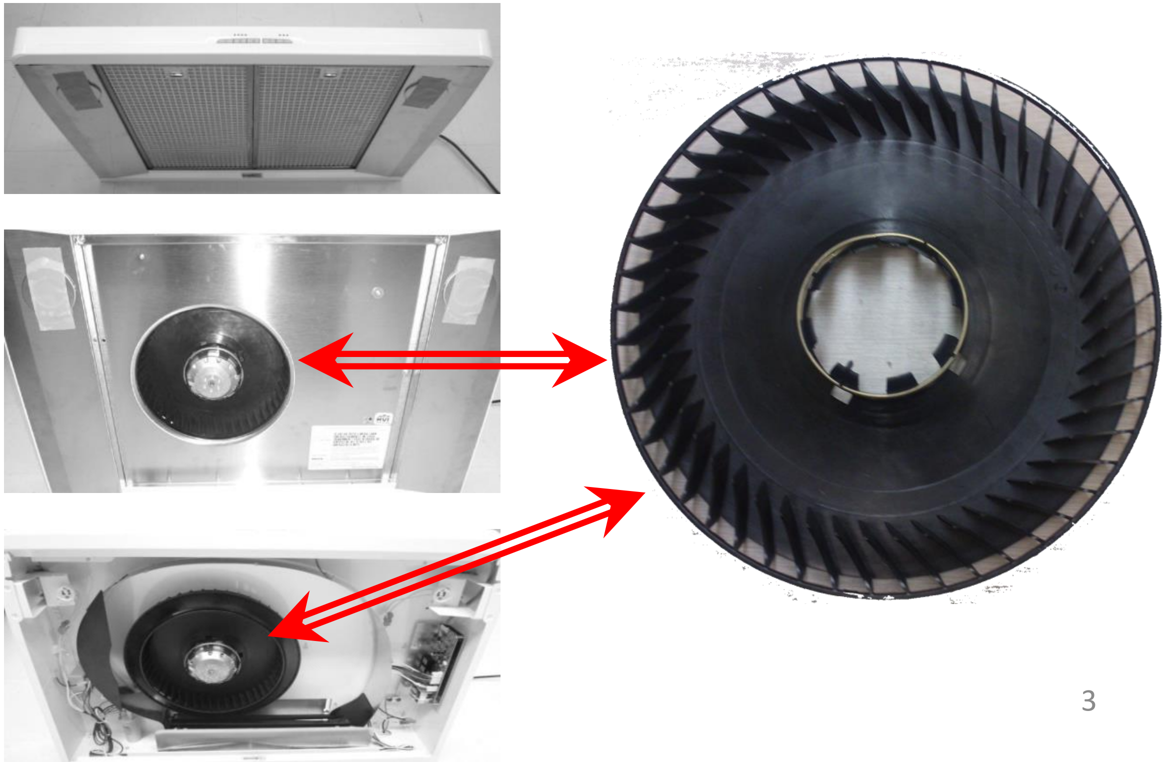
OBJECTIVE

Develop a numerical approach to identify the noisiest zones at the surface of the blade of the centrifugal fan

Kitchen

Kitchen hood

Centrifugal fan



OVERVIEW

OPENFOAM CFD (Computational Fluid Dynamics)

- Geometry
- Boundary conditions
- Mesh (Spatial discretization)

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AEROACOUSTICS

- FW&H (Ffowcs Williams and Hawkings) analogy
- POD (Proper Orthogonal Decomposition) approach
- SVD (Singular Values Decomposition) approach

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RESULTS

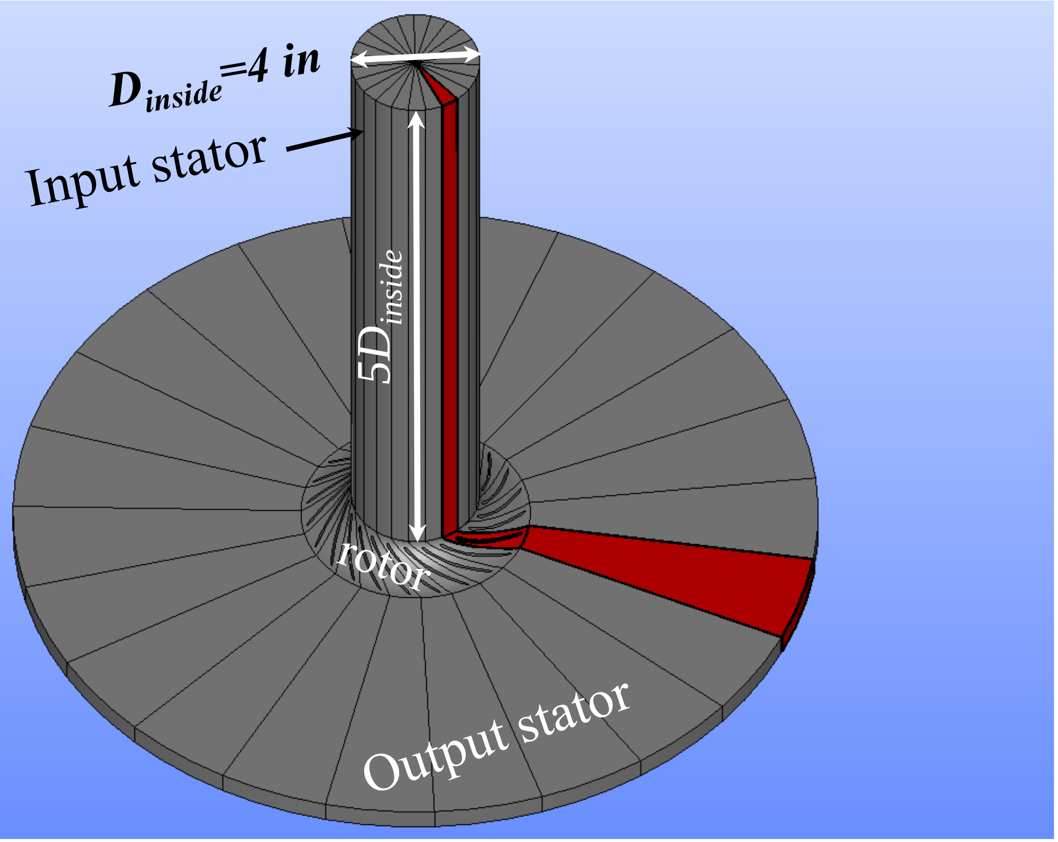
- Radiated sound power
- SVD eigenvector mapping

CONCLUSION

❑Definition of the geometry

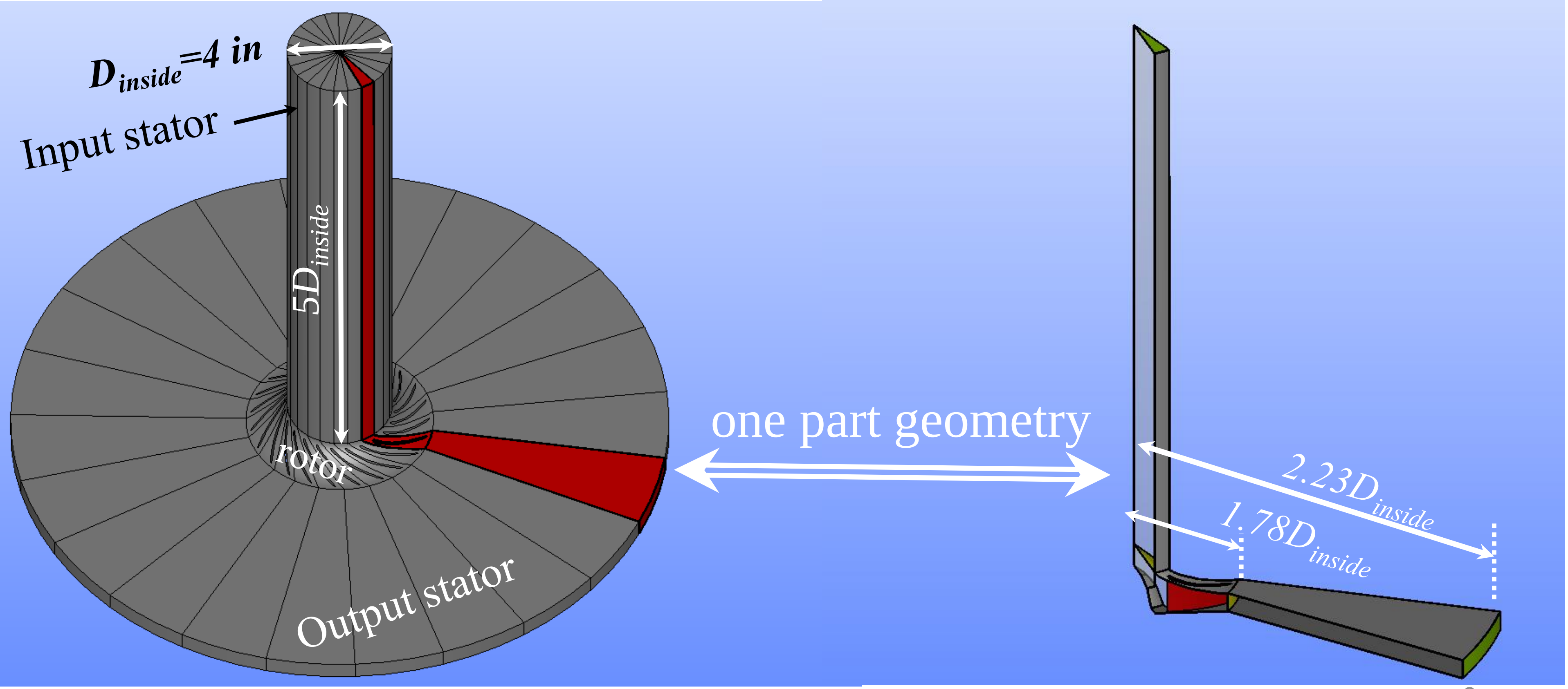
➢ The studied geometry has 2 stators (input and output) and 1 rotor

- The input stator is a cylinder of 4-in diameter
- The output stator is a hollow disk



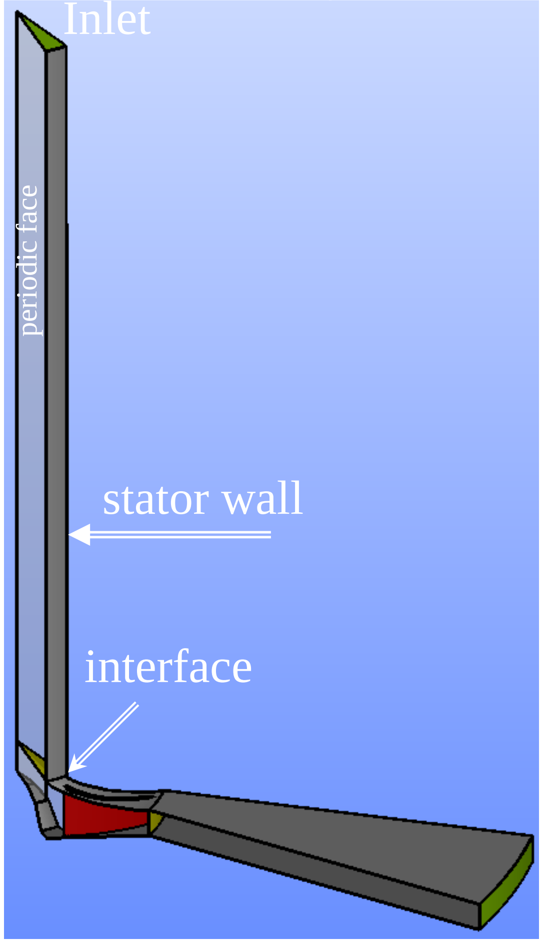
❑Definition of the geometry

- The studied geometry has 2 stators (input and output) and 1 rotor
  - The input stator is a cylinder of 4-in diameter
  - The output stator is a hollow disk
  - The rotor has 23 blades identical
    - In next we use 1/23th of this geometry



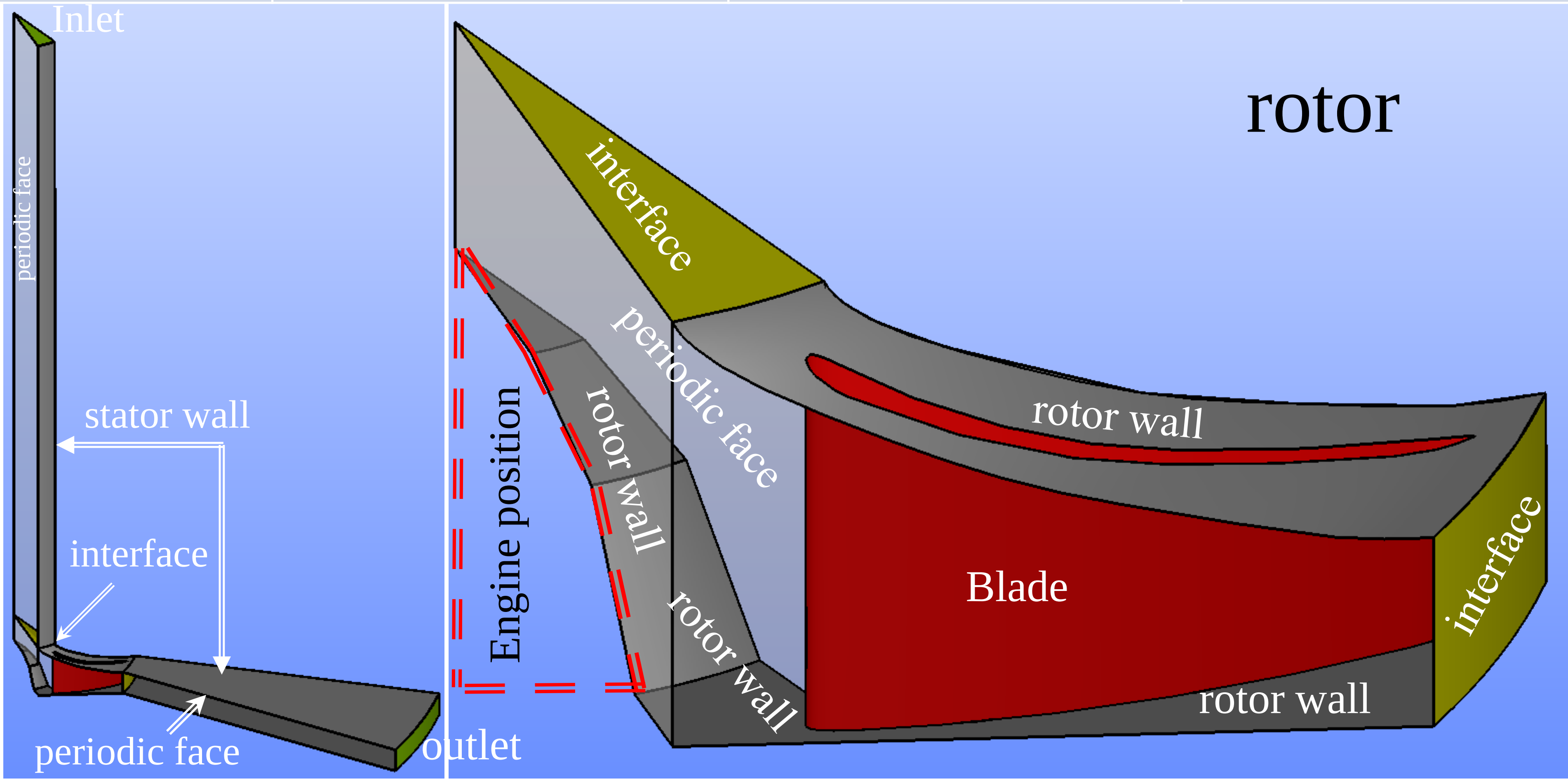
❑Boundary faces

	Input stator		
Boundary faces	Inlet, periodic, stator wall and interface		



❑Boundary faces

	Input stator	Rotor	Output stator
Boundary faces	Inlet, periodic, stator wall and interface	2 interfaces, rotor wall, blade and periodic	Interface, periodic, stator wall and outlet



❑Boundary conditions

<table>
<tr><th></th><th>U</th><th>p</th><th>k</th><th>nuSgs</th></tr>
<tr><td>inlet</td><td>flowRateInletVelocity</td><td rowspan="4">zeroGradient</td><td rowspan="2">fixedValue</td><td>zeroGradient</td></tr>
<tr><td>blade</td><td rowspan="2">movingWallVelocity</td><td>calculated</td></tr>
<tr><td>Rotor walls</td><td rowspan="2">kqRWallFunction</td><td rowspan="2">nuSgsWallFunction</td></tr>
<tr><td>Stator walls</td><td>fixedValue</td></tr>
<tr><td>periodic face</td><td colspan="4">cyglicGgi</td></tr>
<tr><td>interface</td><td colspan="4">overlaGgi</td></tr>
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<tr><td>Solver / preconditiner</td><td>BiCGStab/DILU</td><td>PCG/DIC</td><td>BiCGStab/DILU</td><td></td></tr>
</table>

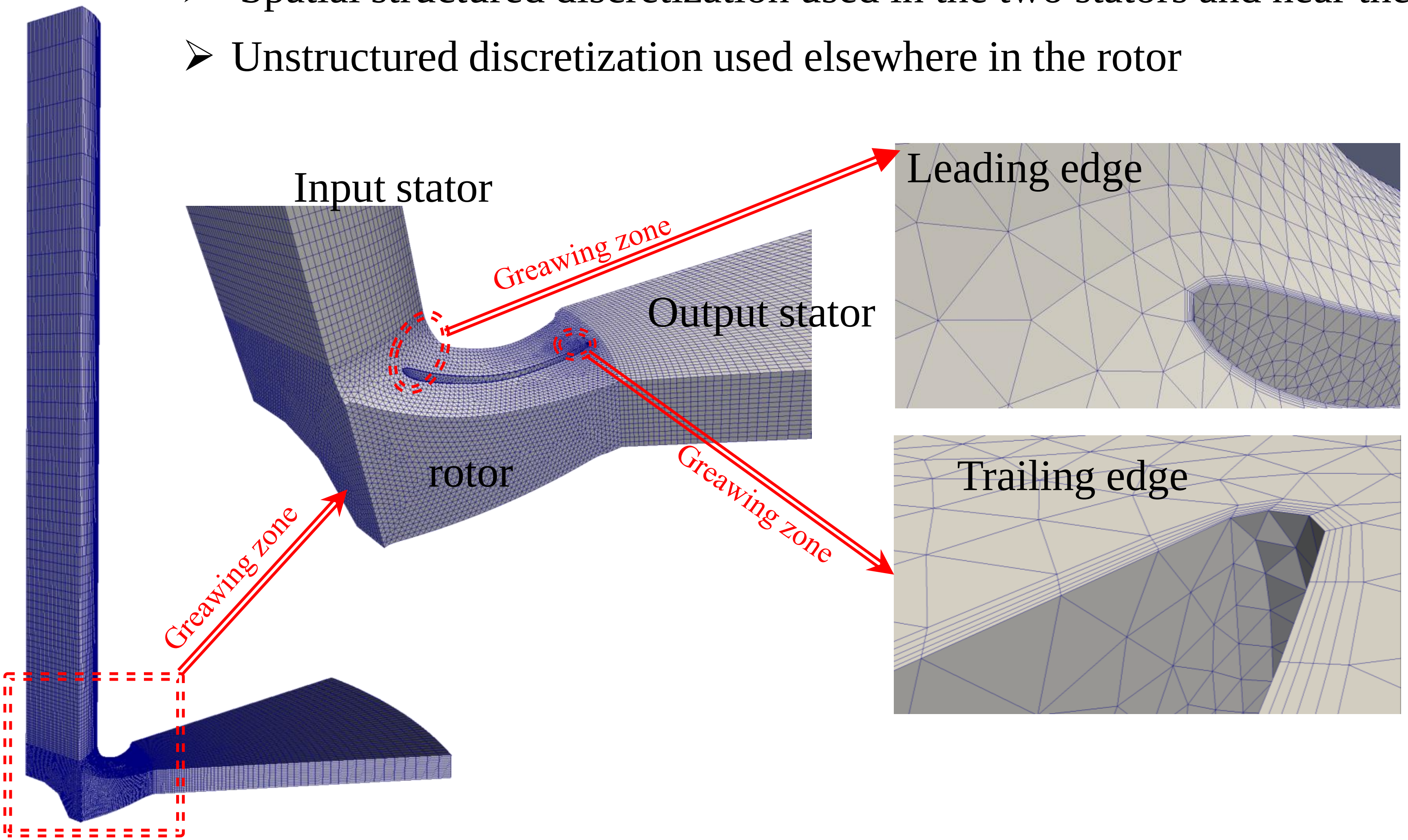
- ➢ BiCGStab: BiConjugate Gradient Stabilized
- ➢ PCG: Preconditioned Conjugate Gradient
- ➢ DILU: Diagonal Incomplete-LU
- ➢ DIC: Diagonal Incomplete Cholesky

❖ LES model (OneEqEddy) is used for modeling turbulent

- The boundary conditions are in the following table
- The flow rate condition is used at the inlet and the zero pressure condition at the outlet

❑ Mesh

- ❖ Salome software is used for mesh
- ❖ Hybrid spatial discretization is used
  - ➢ Spatial structured discretization used in the two stators and near the blade
  - ➢ Unstructured discretization used elsewhere in the rotor



❑ Mesh parameters

- 3 types of mesh are used

Case	Rotor maxsize (in)	Blade maxsize (in)	Total number of elements	Number of elements on the blade
0	0.05	0.05	462 910	22 813
1	0.025	0.05	1 118 234	22 813
2	0.025	0.025	1 381 228	44 972

- Flow rate in inlet: Q = 134.22 cfm
- Outlet pressure value: 0 Pa
- Rotation velocity: Ω = 2 800 rpm
- CFD time step: $\Delta t = 1.05 \times 10^{-7}$ s
- OpenFoam 1.6 extand (pimpleDyMFoam)

❖ Once the calculation converges, source pressure fluctuations are extracted.
❖ Then, they are injected into the acoustic analogy (FW&H).

- Acoustical time step: $\Delta t_{\text{acoustic}} = 8\Delta t_{CFD}$
- Sampling number per revolution: $N_{sampling} / revolution \simeq 4050$

❑Correlation matrix

❖ Ffowcs Williams & Hawkings (FW&H)

- Only the dipole term of the FW&H analogy is considered, the other terms are negligible

$$p_L(\vec{x},t) = \int_{f=0} \frac{1}{4\pi}\left[\frac{\dot{\ell}_r}{cr(1-M_r)^2} + \frac{\ell_r r\dot{M}_r}{cr(1-M_r)^3}\right]_\tau dS \qquad \text{with } \ell_i = pn_i$$

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$f(\vec{x},\vec{y}_i,t_j)$ with $0 \le i < m$ and $0 \le j < N$

- Considering the terms of the integral, matrix [A] is built

$$[A] = \begin{bmatrix} f(\vec{x},\vec{y}_0,t_0) & f(\vec{x},\vec{y}_0,t_1) & \cdots & f(\vec{x},\vec{y}_0,t_{N-1}) \\ f(\vec{x},\vec{y}_1,t_0) & f(\vec{x},\vec{y}_1,t_1) & \cdots & f(\vec{x},\vec{y}_1,t_{N-1}) \\ \vdots & \vdots & \ddots & \vdots \\ f(\vec{x},\vec{y}_{m-1},t_0) & f(\vec{x},\vec{y}_{m-1},t_1) & \cdots & f(\vec{x},\vec{y}_{m-1},t_{N-1}) \end{bmatrix}$$

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- ➢ Each line vector of the matrix [A] represents a sound pressure of one single source along the receiving time at receiver $\vec{x}$
- ➢ Each columns vector of the matrix [A] is the sound contribution of all sources at given reception time at receiver $\vec{x}$

Correlation matrix

- Multiplying matrix [*A*] by its transpose gives a correlation matrix at receiver $\vec{x}$

$$[W]=\frac{1}{N}[A][A]^T$$

Proper Orthogonal Decomposition (POD)

- The POD matrix [*W*] given the eigenvalues λ_k and eigenvectors $\vec{\Phi}_k$ associated

$$[W][\Phi]=[\lambda][\Phi] \Longleftrightarrow [W]\left[\vec{\Phi}_1,\vec{\Phi}_2,\cdots,\vec{\Phi}_M\right]=\left[\lambda_1\vec{\Phi}_1,\lambda_2\vec{\Phi}_2,\cdots,\lambda_M\vec{\Phi}_M\right]$$

$$[W]=\sum_{k=0}^{m-1}\lambda_k\vec{\Phi}_k\vec{\Phi}_k^T \qquad [\Phi][\Phi]^T=[I]$$

Radiated sound power by POD

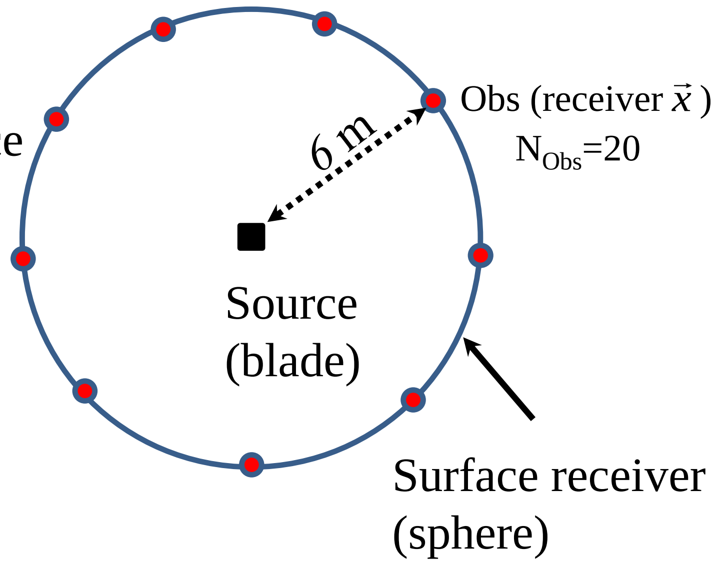
- Using POD on matrix [*W*], the radiated sound power can be computed with only the ***q*** dominant eigenvalues

$$P=\frac{S}{N_{obs}\rho c}\sum_{obs=0}^{N_{obs}-1}\left[\sum_{k=0}^{q-1}\lambda_k^{obs}\left(\sum_{i=0}^{m-1}\Phi_{k,i}^{obs}\right)^2\right]$$

❑ Definition of radiated sound power

- Considering a receiver distribution on a sphere around the source [ISO 3447]

- Acoustical far field hypothesis:
  - the radius is very large as characteristic size to the source
  - the radius is very large as wavelength



- Under this hypothesis the sound intensity at a receiver is:

$$I = \frac{p_{obs}^2}{\rho c}$$

- Sound power of the source is the sound intensity passing through the surface receiver S

➢Direct FW&H $P = \frac{S}{N_{obs}\rho c}\sum_{obs=0}^{N_{obs}-1} p_{obs}^2$

where $p_{obs} = p_L(\vec{x},t) = \int_{f=0} \frac{1}{4\pi}\left[\frac{\dot{\ell}_r}{cr(1-M_r)^2} + \frac{\ell_r r\dot{M}_r}{cr(1-M_r)^3}\right]_\tau dS$

with $\ell_i = pn_i$ c is the sound speed M Mach number

Singular Value Decomposition (SVD) approach

- The vertical superposition of all correlation matrix ($[W_i]$) of all the receivers provides the SVD rectangular matrix ($[W_{SVD}]$)

$$[W_{SVD}] = \begin{bmatrix} [W_0] \\ [W_1] \\ \vdots \\ [W_{Nobs-1}] \end{bmatrix}$$ with $[W_i]$ is a correlation matrix at receiver $\vec{x}_i$

➢ SVD $[W_{SVD}] = [U][\sigma][V]$

$[U]$ Left eigenvector matrix . It account for the difference between the receivers

$[\sigma]$ Diagonal eigenvector matrix

$[V]$ Right eigenvector matrix. It gives the average acoustic information from the source in free field

Radiated sound power by SVD

- Using SVD on matrix $[W_{SVD}]$, the radiated sound power can be computed with only the $\boldsymbol{q}$ dominant eigenvalues

$$P = \frac{S}{N_{obs}\rho c} \sum_{k=0}^{q-1} \left(\sigma_k \left(\sum_{i=0}^{N_{obs}*m-1} U_{k,i} \right) \left(\sum_{j=0}^{m-1} V_{k,j} \right) \right)$$

RESULTS

❑Radiated sound power

- Comparing the sound power of the:
  - Direct FW&H $$P = \frac{S}{N_{obs}\rho c}\sum_{obs=0}^{N_{obs}-1} p_{obs}^2$$
  - POD $$P = \frac{S}{N_{obs}\rho c}\sum_{obs=0}^{N_{obs}-1}\left[\sum_{k=0}^{q-1}\lambda_k^{obs}\left(\sum_{i=0}^{m-1}\Phi_{k,i}^{obs}\right)^2\right]$$
  - SVD $$P = \frac{S}{N_{obs}\rho c}\sum_{k=0}^{q-1}\left(\sigma_k\left(\sum_{i=0}^{N_{obs}*m-1}U_{k,i}\right)\left(\sum_{j=0}^{m-1}V_{k,j}\right)\right)$$

m is the blade cellule number

N_{obs} is the receivers number

c is the sound speed

ρ is the flow density

S is the receiver surface

- The calculated radiated sound power in each mesh case is in the following table

	Case 0 – coarse			**Case 1 – medium**			**Case 2 - fine**		
	FW&H	POD	SVD	FW&H	POD	SVD	FW&H	POD	SVD
q	*22 813*	**10**	**3**	*22 813*	**10**	**3**	*44 972*	**10**	**3**
P(dB)	65.0	64.7	64.6	61.1	60.8	60.4	63.8	63.5	63.4

Ref 10^{-12} W

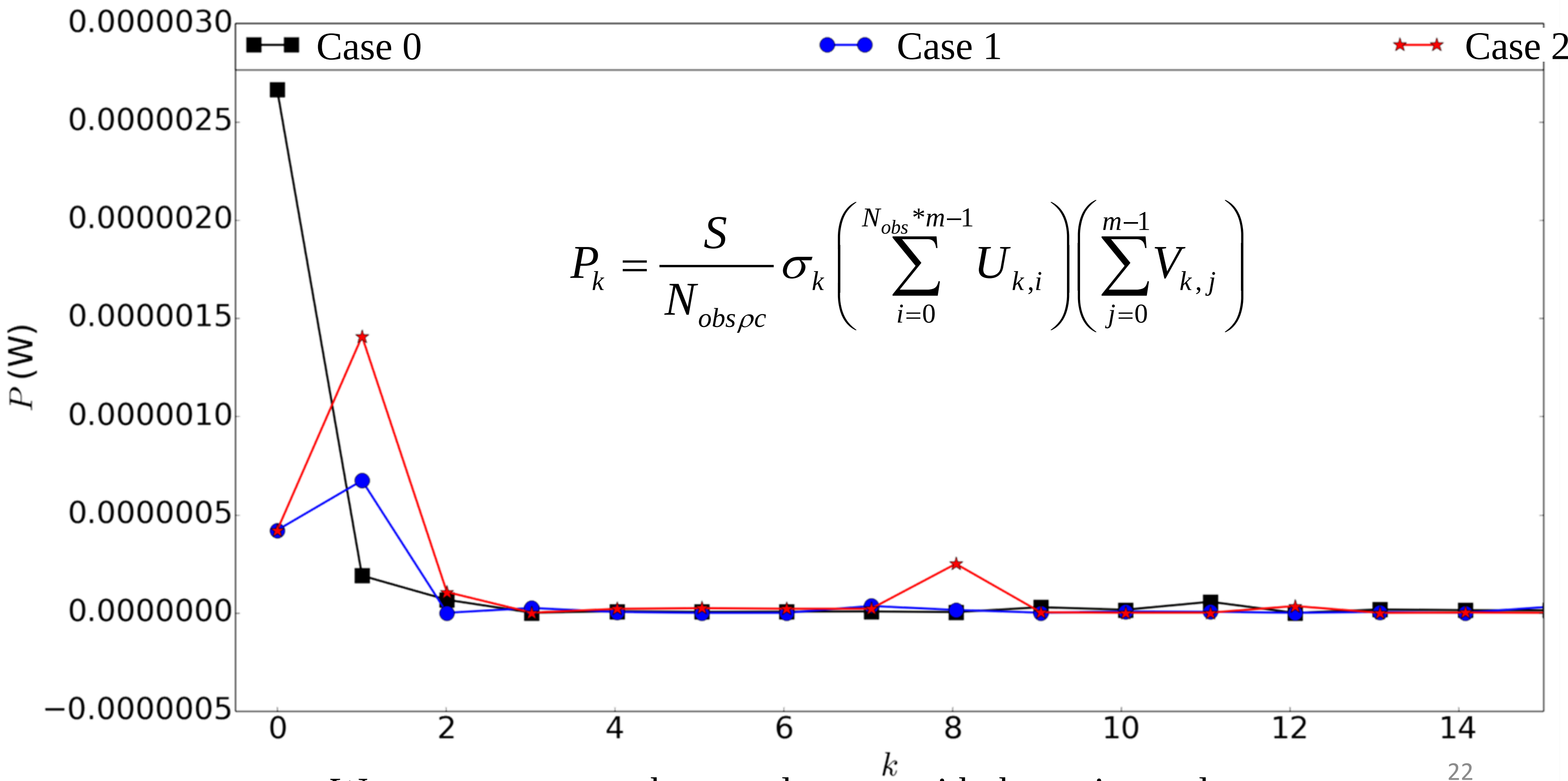
- ➢ Sound power is almost identical for all cases
- ➢ Sound power can be reconstructed with a few eigenvalues only

RESULTS

❑Radiated sound power by SVD

$$P = \frac{S}{N_{obs}\rho c}\sum_{k=0}^{q-1}\left(\sigma_k\left(\sum_{i=0}^{N_{obs}*m-1}U_{k,i}\right)\left(\sum_{j=0}^{m-1}V_{k,j}\right)\right)$$

- The sound power of each eigenvalues is given by the following graph



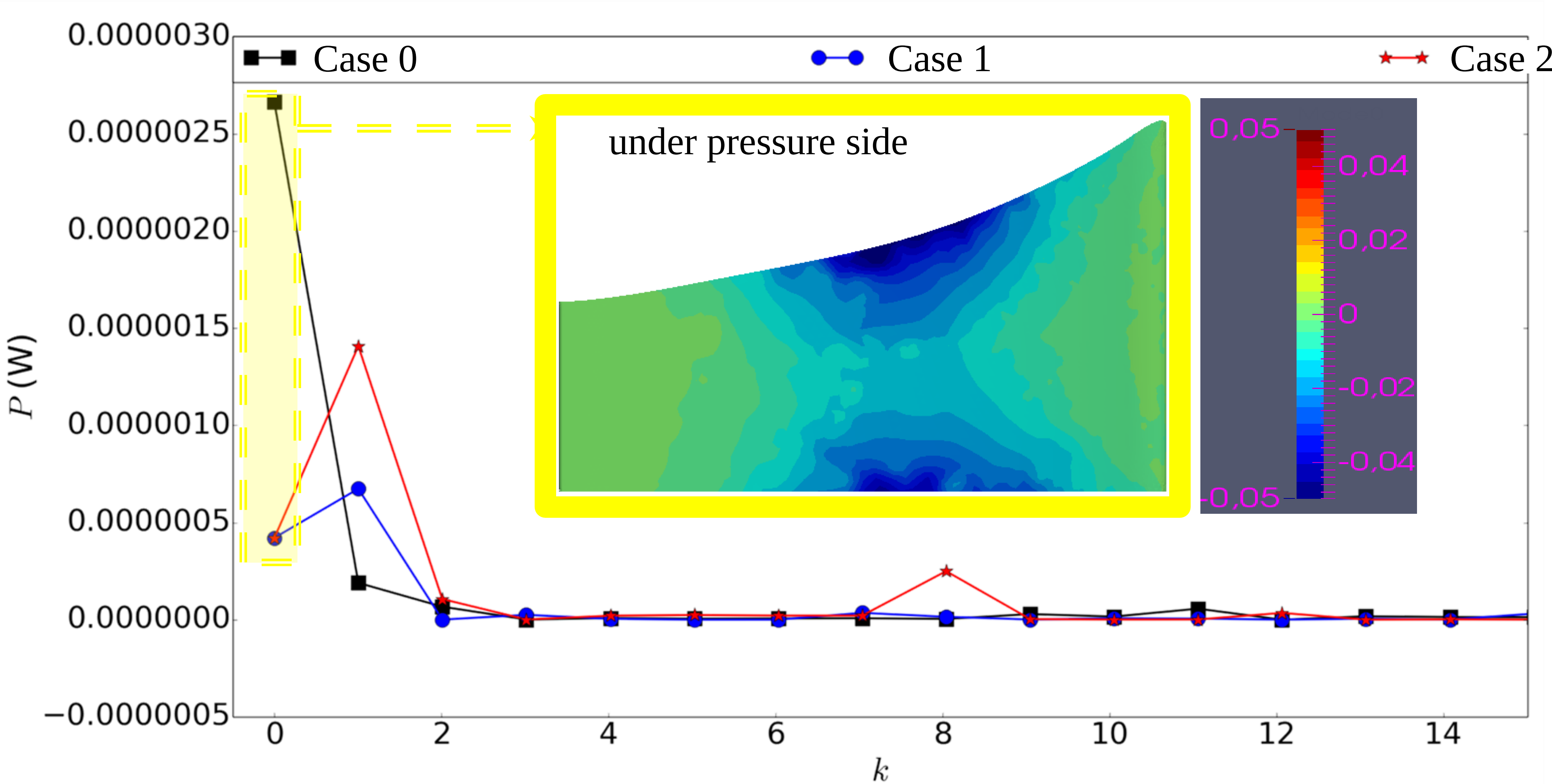
$$P_k = \frac{S}{N_{obs\rho c}}\sigma_k\left(\sum_{i=0}^{N_{obs}*m-1}U_{k,i}\right)\left(\sum_{j=0}^{m-1}V_{k,j}\right)$$

- We can reconstruct the sound power with three eigenvalues

RESULTS

❑Radiated sound power by SVD - eigenvector V_0 mapping

- Mapping eigenvectors show that the middle of the blade is the noisiest

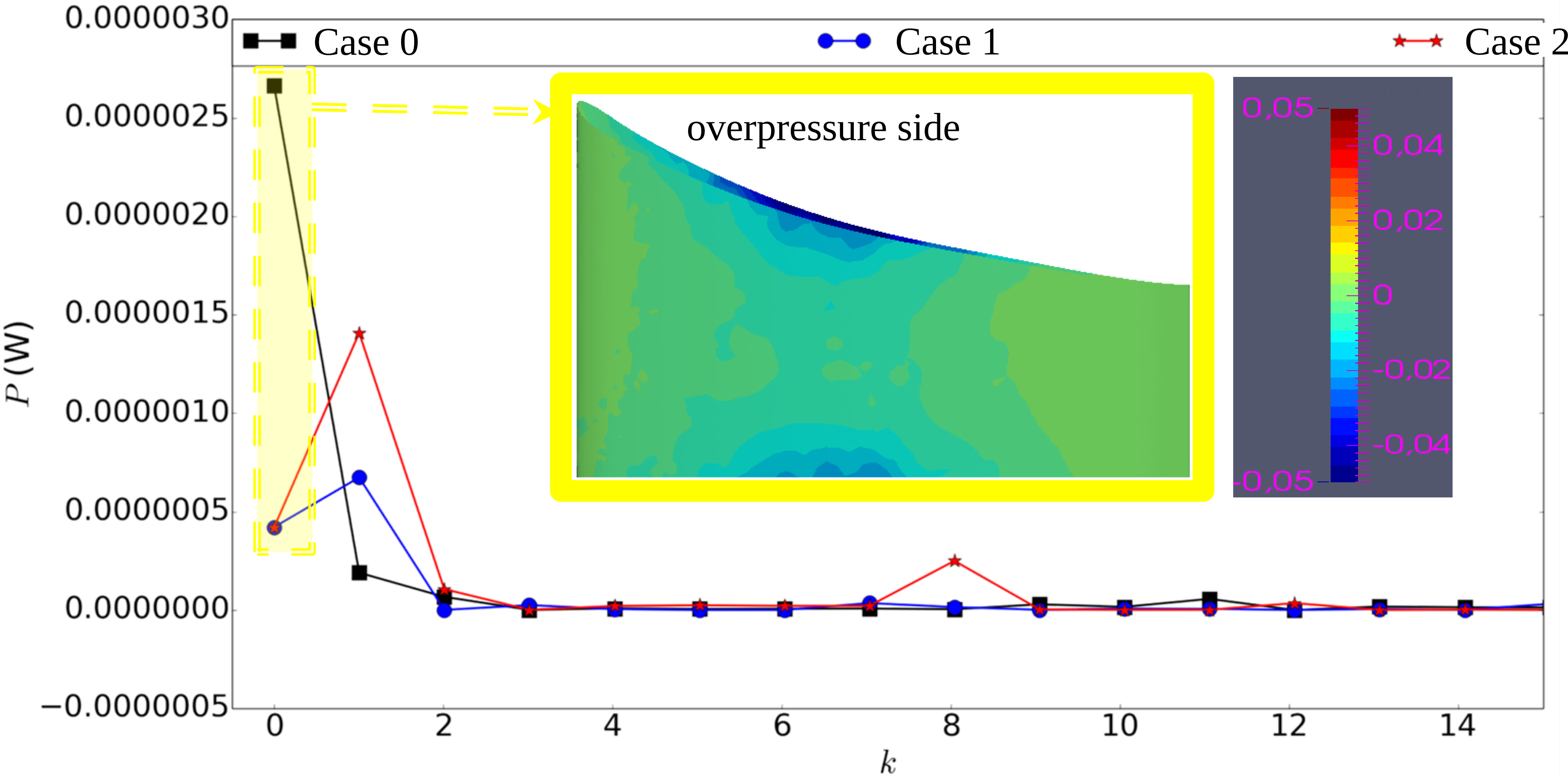


- Seen on the under pressure side with eigenvector 0

RESULTS

❑Radiated sound power by SVD - eigenvector V_0 mapping

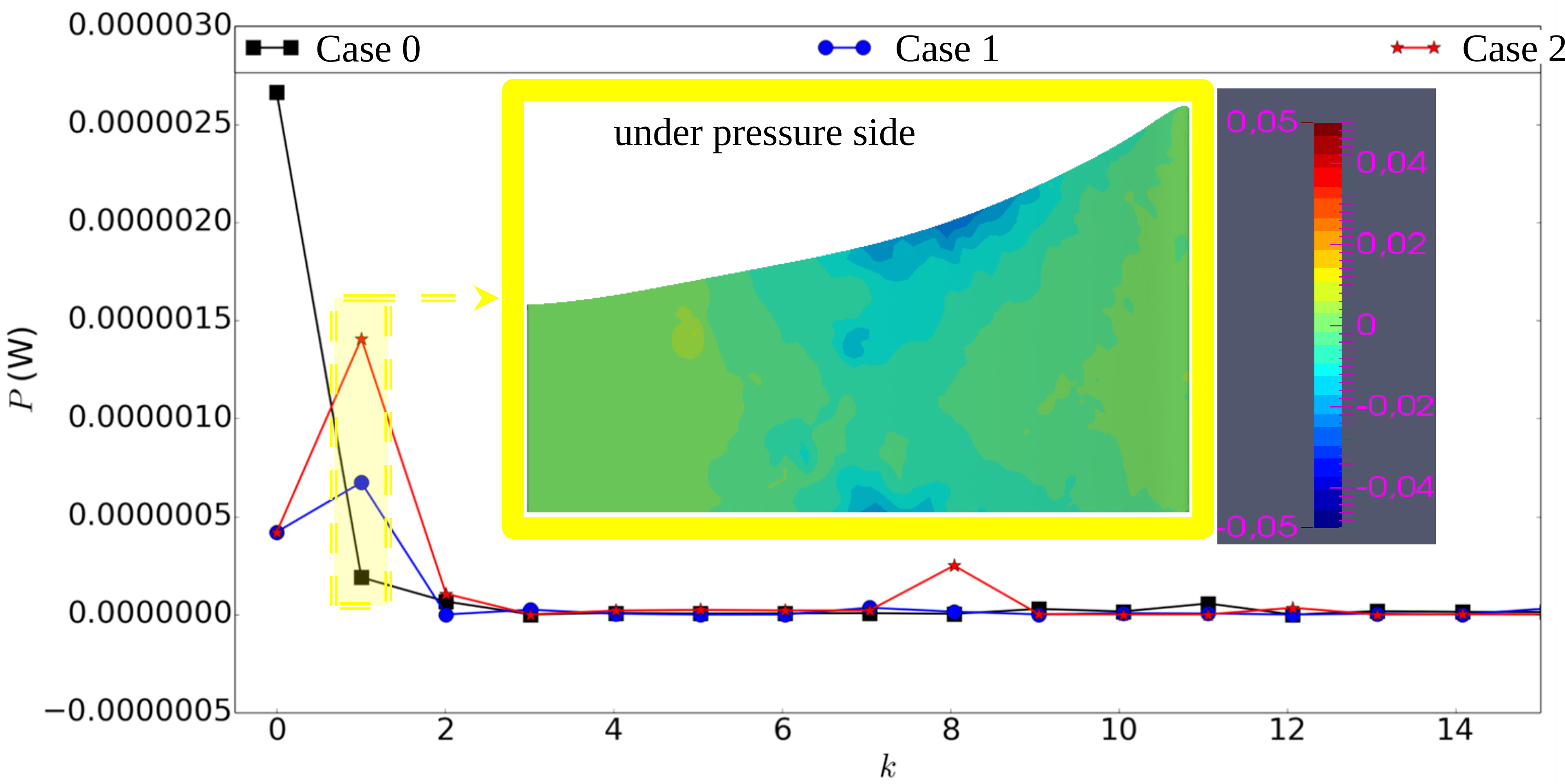
- Uniform contribution to the overpressure side



RESULTS

❑Radiated sound power by SVD - eigenvector V_1 mapping

- Uniform contribution to the under pressure side



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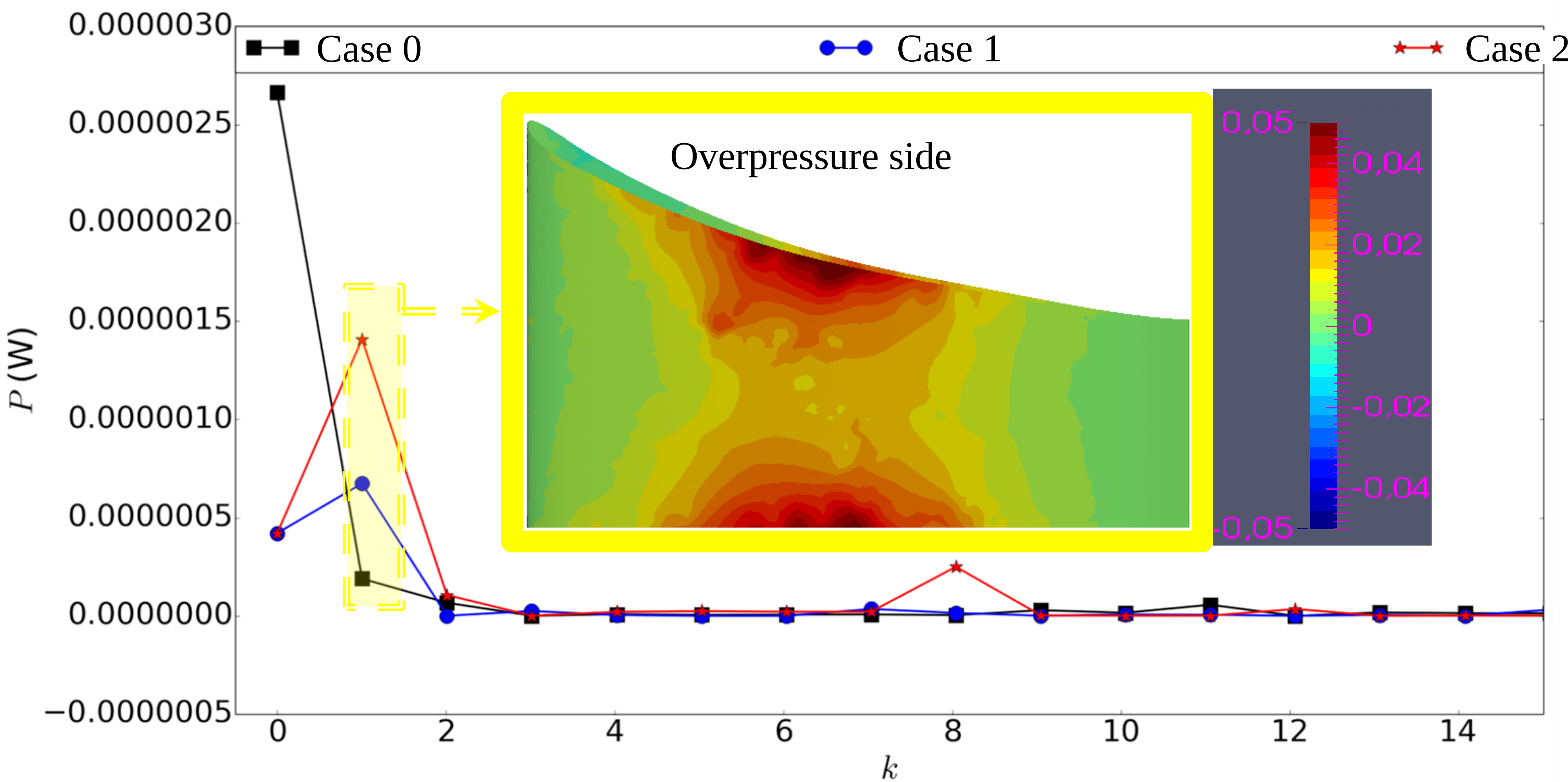

- Seen on the overpressure side with eigenvector 1

CONCLUSION

❑ From this study we can conclude that:

- ➢ Radiated sound power is almost identical for coarse, middle and fine mesh
- ➢ Radiated sound power computed by the modal decompositions (POD & SVD) can be reconstructed with only a few eigenvalues
- ➢ SVD mapping eigenvectors show that the middle of the blade is the noisiest zone

❑ Future work

- ✓ Optimize the geometry of the noisiest zone to minimize the radiated sound power of the centrifugal fan
- ✓ Implement our acoustic analogy code (C++) directly into the OpenFoam plateform

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THANK YOU

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