SIZE AND WEIGHT OF WIND TURBINES

EX1: VESTAS V90, 1.8 MW

TOWER HEIGHT : 120 m

BLADE LENGTH : R=45 m

NACELLE WEIGHT : 75 t
BLADES (ALL 3) " : 40 t

$\left.\begin{array}{l}75\,t\\40\,t\end{array}\right\} = 115\,t$

TOWER " : 152 t

EX2: MHI-VESTAS V164, 9.5 MW

TOWER/HUB HEIGHT : 105 m

BLADE : R = 82 m

NACELLE WEIGHT : 390 t
BLADES (3) : 105 t

$\left.\begin{array}{l}390\,t\\105\,t\end{array}\right\} 500\,t$

TOWER WEIGHT : ~ 600 t

TOWER BASE DIAMETER : 6.5 m

4.3.6 STIFF AND SOFT TOWERS

LOWEST EXCITATION FREQUENCIES: 1P , B·P

(WITH B = NO OF BLADES, E.G. B=3)

1P: "ROTOR ROTATION FREQUENCY"
SINGLE BLADE EXCITATIONS,
BLADE ASYMMETRIES

B·P: "BLADE PASSING FREQUENCY"

FOR TIP SPEED RATIO λ AND RADIUS R AND WIND SPEED u_∞ WE HAVE

$$\omega_{1P} = \frac{\lambda \cdot u_\infty}{R} \quad , \quad \omega_{B \cdot P} = B \cdot \frac{\lambda u_\infty}{R}$$

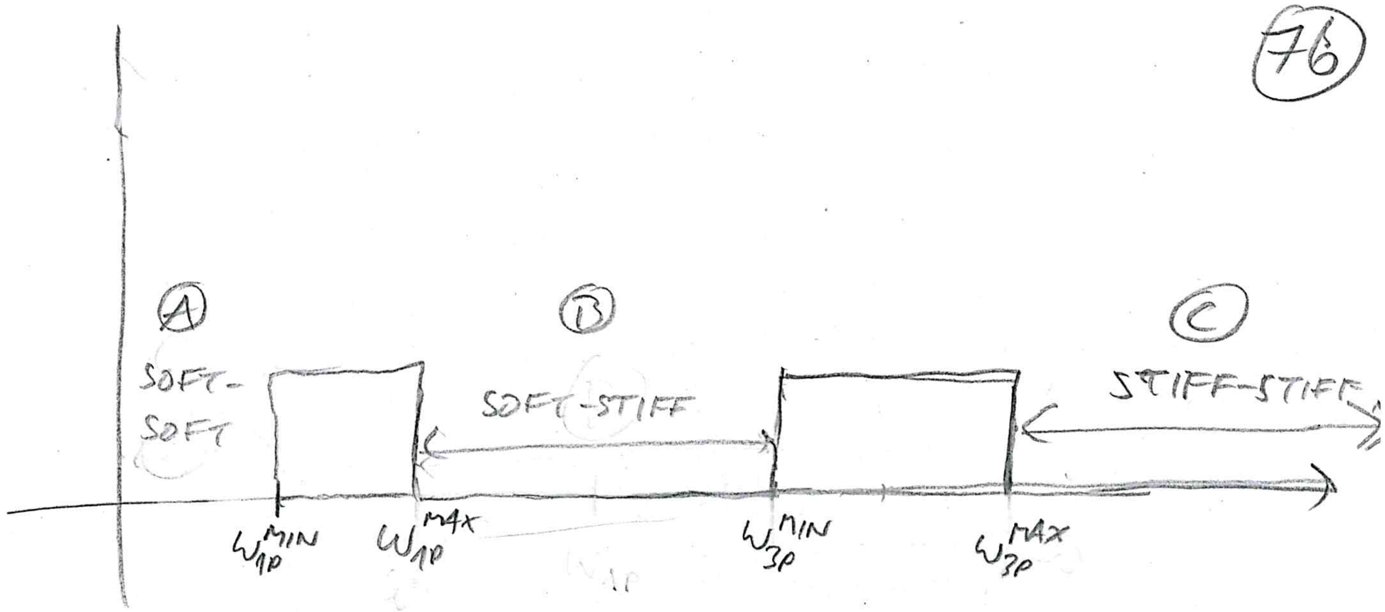
EX: $R = 80\,m$, $\lambda = 8$, $u_\infty = 10\,m/s$: $\omega_{1P} = 1\,\frac{rad}{s}$

$$\omega_{3P} = 3\,\frac{rad}{s}$$

NOTE THAT WE ALWAYS HAVE $\boxed{\omega_{BP} = B \cdot \omega_{1P}}$

AND THAT ω_{1P} TYPICALLY VARIES WITH WIND SPEED.

PROBLEMS WOULD ARISE IF ω_{1P} OR ω_{BP} BECOME EQUAL TO TOWER EIGENFREQUENCIES, SO THEY ARE TO BE AVOIDED BY (a) TOWER DESIGN AND
(b) CONTROLLER DESIGN



GIVEN THE RANGE OF OPERATIONAL SPEEDS, THE TOWER CAN BE

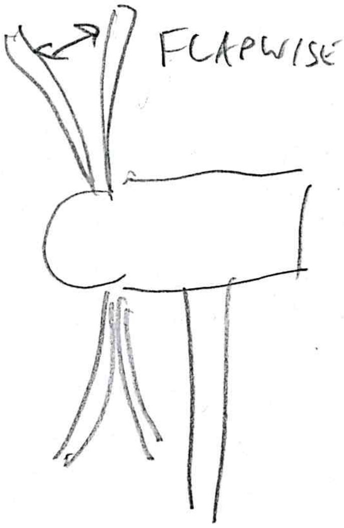
(A) "SOFT-SOFT" IF ITS LOWEST EIGENFREQUENCY ω_{tower} IS IN REGION (A)

(B) "SOFT-STIFF" IF ω_{tower} IS IN REGION (B)

(C) "STIFF-STIFF" IF ω_{tower} IS IN REGION (C), I.E. HIGHER THAN B.P. IN THIS CASE, <u>ALL</u> EIGENFREQUENCIES ARE ABOVE ω_{BP}^{MAX}

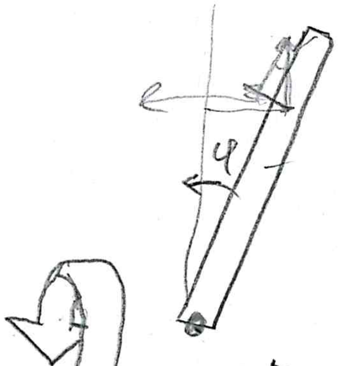
4.4 BLADE OSCILLATIONS AND CENTRIFUGAL STIFFENING

BLADE OSCILLATIONS MOSTLY OCCUR "FLAPWISE"
I.E. FORWARD-BACKWARD



INTERESTINGLY, DUE TO ROTATION, THE BLADE "STIFFENS" AND HAS HIGHER EIGENFREQUENCIES THAN IT WOULD HAVE WITHOUT ROTATION. LET'S SEE WHY.

4.4.1 ROTATING, HINGED BEAM (NO ELASTICITY)



MOMENT OF INERTIA

$$I = \int_0^R \mu(r)\, r^2\, dr$$

FLAPWISE OSCILLATION ANGLE φ
RESTORING MOMENT $M(\varphi)$

$$\boxed{I\ddot{\varphi} = M(\varphi)} \quad (1)$$

ROTATION FREQUENCY Ω

MOMENT

$M(\varphi)$ COMES FROM CENTRIFUGAL FORCE

$$M(\varphi) = -\int_0^R \mu(r)\,\Omega^2 \cdot r \cdot \underbrace{\cos(\varphi)}_{\approx 1}\ \underbrace{\sin(\varphi)}_{\approx \varphi} \cdot r \, dr$$

$$= -\Omega^2 \cdot \int_0^R \mu(r)\, r^2\, dr \cdot \varphi$$

$$= -\varphi \cdot \Omega^2 \cdot I$$

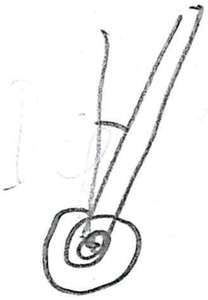
WITH (1) THIS GIVES, MAGICALLY,

$$I\ddot{\varphi} = -\Omega^2 I \varphi \iff \varphi(t) = A\sin(\Omega t)$$

IS SOLUTION,

EIGENFREQUENCY EQUALS ROTOR FREQUENCY!

4.4.2 ROTATING BEAM WITH TORSIONAL SPRING



$$M(\varphi) = -\Omega^2 I \varphi - K\varphi$$

$\boxed{\text{SPRING CONSTANT } K}$

NATURAL RESONANCE

$\omega_{NR} = \sqrt{\frac{K}{I}}$

$$I\ddot{\varphi} = -\left(\Omega^2 I + K\right) \cdot \varphi$$

$$\iff \ddot{\varphi} = -\left(\Omega^2 + \frac{K}{I}\right)\varphi = -\left(\Omega^2 + \omega_{NR}^2\right)\varphi$$

$\omega_R^2 = \omega_{NR}^2 + \Omega^2$

"CENTRIFUGAL STIFFENING"