Vibration isolation of dimple plate heat exchangers

by

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SUMMARY

Dimple plate heat exchangers are a new type of welded compact plate heat exchangers. The dimple plates increase the turbulence of the fluid flowing over the plate, increasing the efficiency of the heat exchanger without increasing pressure drop over the heat exchanger. The compact design of the heat exchanger makes it possible to install the heat exchanger at the top of condenser columns, reducing the footprint area of the column by replacing standard shell and tube condensers at the foot of the column.

After the implementation of these condensers in 2008, Sasol experienced failures of 12 column-top dimple plate condensers in unit 300. In these cases damage was observed at the weld between the plates and the bottom header. One possible reason for the damage was vibration caused by the flow over the dimple plates.

The characteristics of flow-induced vibrations in dimple plate heat exchangers were studied in the scientific literature. It was, however, found that although the effect of dimples on channel walls had been well-researched, the fluid-elastic vibration of the bluff body containing the dimples had not been sufficiently studied. A simple aerodynamic model was constructed to determine the characteristics of the combination of vibration caused by the bluff body (plate) and the flow over the dimples on the plate. The experiment showed the generation of two flow-induced vibration amplitudes.

The structure of the heat exchanger was modelled by using mass, stiffness and damping elements. With certain assumptions the model was reduced to a two degrees of freedom system that simulated the most prominent vibration direction. This model was used to simulate the effect of design changes to the response of the structure to a range of forcing frequencies.

An experimental model of the column-top condensers was constructed and the response due to different vibration frequencies was measured. The measured results were compared with the theoretically predicted values for cases with the current design and the cases where the vibration-control concept was implemented. This validated the theoretical model and the mathematical simulation as a tool to design vibration-control systems for real heat exchangers.

With the replacement of the very stiff mounts that are used in current designs with soft rubber mounts, the dynamic forces on the internal plates was reduced by up to 97.8% for certain forcing frequencies. The deflection of the internal plates is a main cause of stress in the plates and, more importantly, the weld fillets connecting the bottom of the plates to a common header. This repeated stress can easily cause fatigue failure in the welds. By therefore reducing the amplitude of vibration of the heat exchangers, the onset of fatigue failure will be substantially delayed, increasing the reliable lifetime of the column-top condenser.

This concept is not only limited to dimple plate heat exchangers. The oscillating stress in any internal component can, therefore, be reduced by isolating the whole system with soft rubber mounts of a determined stiffness and layout.

Studieleier: Dr. C.B. Nel

OPSOMMING

Dimpelplaathitteruilers is ‘n nuwe tipe kompakte hitteruiler met volledig gesweiste nate. Die dimpels verhoog die turbulensie van die vloeier wat oor the plaat vloei, wat die effektiwiteit van die hitteruiler verhoog sonder om die drukval oor die hitteruiler noemenswaardig te verhoog. Die kompakte ontwerp van die hitteruiler maak dit moontlik om die grondoppervlakte wat benodig word vir ‘n kondensasiekolom te verminder, deur die hitteruiler bo in die kolom te installeer en ‘n buis-hitteruiler, wat gewoonlik by die voet van die kolom geïnstalleer is, te vervang. Na die implementering van hierdie kondensators in 2008, het Sasol 12 falings ervaar. In hierdie gevalle is skade aan die sweisnaat tussen die plate en die onderste mondstuk aangemeld. Een moontlike rede vir die skade was vermoeidheidsfaling weens vibrasie, veroorsaak deur vloei oor die plate.

Die eienskappe van vloei-geïnduseerde vibrasie by dimpelplate is nagevors in die wetenskaplike literatuur. Daar is gevind dat, hoewel daar studies was oor die effek van dimpels in vloeikanale, die effek van vloei-geïnduseerde vibrasie op soliede strukture met dimpels op nie genoegsaam bestudeer is nie. ‘n Eenvoudige aërodinamiese model is gebou om die eienskappe van die kombinasie van vibrasie, veroorsaak deur die beweging van die struktuur (plate) en die vloei oor die dimpels op die plate, te bepaal. Die eksperiment toon twee vloei-geïnduseerde vibrasie-amplitudes.

Die struktuur van die hitteruiler is gemodelleer deur gebruik te maak van massa-, styfheids- en dempingselemente. Met sekere aannames is die model vereenvoudig na ‘n twee grade van vryheid-stelsel, wat die mees prominente vibrasierigting simuleer. Die model is gebruik om die effek van ontwerpveranderinge op die respons teen ‘n aantal forserende frekwensies na te boots.

‘n Eksperimentele model is van die kondensator gebou en die respons, as gevolg van verskillende vibrasie-frekwensies, is gemeet. Die gemete

resultate is vergelyk met die teoretiese waardes vanuit die wiskundige model vir die huidige opstelling en die voorgestelde opstelling van die kondensators. Hierdie bekragtiging het getoon dat die wiskundige model gebruik kan word vir die ontwerp van vibrasiebeheerstelsels in werklike hitteruilers.

Met die vervanging van die stywe staal hegtingstelsel wat huidig gebruik word, met ‘n sagte rubber hegtingstelsel, word die dinamiese kragte in die interne plate met tot 97.8% verlaag vir ‘n aantal forserende frekwensies. Die deurbuiging van die plate is ‘n groot oorsaak van spannings in die plate en die sweisnate wat die plate aan ‘n gemeenskaplike mondstuk heg. Hierdie herhaalde spanning kan maklik vermoeidheidsfaling in the sweisnate veroorsaak. Deur die amplitude van die deurbuiging te verminder, word die vermoeidheidsfaling van die hitteruiler uitgestel en die bruikbare leeftyd van die hitteruiler verleng.

Hierdie konsep is egter nie beperk tot dimpelplaathitteruilers nie. Die ossilerende spanning in enige interne komponent kan beduidend verminder word deur die hele struktuur te isoleer met ‘n elastiese voetstuk van ‘n bepaalbare ontwerp en styfheid.

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LIST OF ABBREVIATIONS

Abbreviation	**Meaning**
DOF	Degrees of freedom
FEA	Finite Element Analysis
FEM	Finite Element Method
FFT	Fast Fourier Transform
PIV	Particle image velocimetry
RMS	Root mean squared

LIST OF SYMBOLS

Symbols	***Meaning***	***Units***
$[C]$	General damping coefficient matrix for use in two DOF models	Ns/m
$[C]_M$	Damping coefficient matrix for soft rubber-mounted case	Ns/m
$[C]_R$	Damping coefficient matrix for very stiff steel-mounted case	Ns/m
c	Damping coefficient of an arbitrary component	Ns/m
c_c	Damping coefficient contributed by compensators	Ns/m
c_{ci}	Damping coefficient of an individual compensator	Ns/m
c_m	Damping coefficient contributed by mounts	Ns/m
c_{mi}	Damping coefficient of an individual mount	Ns/m
c_p	Damping coefficient contributed by plates	Ns/m
c_1	Damping coefficient between frame and mass (m_1) in general case	Ns/m
c_{1M}	Damping coefficient between frame and mass (m_1) for soft rubber-mounted case	Ns/m
c_{1R}	Damping coefficient between frame and mass (m_1) for stiff steel-mounted case	Ns/m

c_2	Damping coefficient between masses (m_1 and m_2)	Ns/m
c_3	Damping coefficient between mass (m_2) and frame	Ns/m
D	Dimple print diameter	m
D	Width of frontal area of cylinder	m
$\mathbf{F}$	Force vector of 2 DOF model	N
F_k, F F_{k2}	Resultant force in spring	N
$F(t)$	Time dependent oscillating force	N
F_{1RMS}	RMS value of resultant force in element 1	N
F_{2RMS}	RMS value of resultant force in element 2	N
F_{3RMS}	RMS value of resultant force in element 3	N
F_0	Amplitude of oscillating force $F(t)$	N
f_n	Natural frequency	Hz
f_s	Frequency of vortex shedding	Hz
H	Height of dimple plate channel	m
H/D	Non-dimensional dimple aspect ratio	
$[\mathbf{K}]$	General stiffness matrix of two DOF model	N/m
$[\mathbf{K}]_M$	Stiffness matrix for soft rubber-mounted case	N/m

$[\mathbf{K}]_R$	Stiffness matrix for stiff steel-mounted case	N/m
k	Stiffness of an arbitrary component	N/m
k_c	Stiffness contributed by compensator	N/m
k_{ci}	Stiffness of individual compensator	N/m
k_m	Stiffness contributed by soft rubber mounts	N/m
k_{mi}	Stiffness of individual soft rubber mount	N/m
k_p	Stiffness contributed by plates	N/m
k_1	Stiffness between mass (m_1) and the frame in two DOF model (general case)	N/m
k_{1M}	k_1 value for soft rubber-mounted case	N/m
k_{1R}	k_1 value for stiff steel-mounted case	N/m
k_2	Stiffness between two masses (m_1 and m_2) in two DOF model	N/m
k_3	Stiffness between mass (m_2) and frame	N/m
m	Mass of an arbitrary component	kg
$\mathbf{M}$	Mass matrix of 2 DOF model	kg
m_B	Total effective mass of bottom frame	kg
m_b	Mass of bottom steel structure	kg
m_e	Equivalent mass of plates during vibration	kg
m_m	Mass of electric vibrating motor	kg

m_p	Mass of plates	kg
m_r	Remainder of mass of plates	kg
m_T	Mass of top frame	kg
m_{Tot}	Total mass of vibration model	kg
m_1	Mass connected to top frame in two DOF model	kg
m_2	Mass connected to bottom frame in two DOF model	kg
N_c	Number of compensators in elements	
N_m	Number of mounts in element	
S	Strouhal number	
$\overline{V}$	Mean flow velocity	m/s
x_1	Displacement of mass (m_1) in two DOF model	m
x_2	Displacement of mass (m_2) in two DOF model	m
$\triangle x$	Relative displacement of two arbitrary masses	m
$\dot{x}_1$	Velocity of mass (m_1) in two DOF model	m/s
$\dot{x}_2$	Velocity of mass (m_2) in two DOF model	m/s
$\ddot{x}_1$	Acceleration of mass (m_1) in two DOF model	m/s^2

$\ddot{x}_2$	Acceleration of mass (m_2) in two DOF model	m/s^2
X_{1RMS}	Root mean square value of displacement x_1	m
X_{2RMS}	Root mean square value of displacement x_2	m
δ	Depth of dimple	m
δ/D	Non-dimensional dimple depth	
ξ_c	Damping ratio of rubber pipe compensators	
ξ_m	Damping ratio of rubber mounts	
ξ_p	Damping ratio of plate pack	
ϖ	Frequency of oscillation	rad/s
ϖ_n	Natural frequency	rad/s

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