Method for Quantifying Heat Duty Derating Due to Interpass Leakage in Bolted Flat Cover Heat Exchangers

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Functional relations are derived to determine the reduction in the heat duty of a multipass heat exchanger due to tube-side fluid bypass. A simplified structural model of the channel cover is constructed to determine the interpass leakage due to cover deflection under pressurized (operating) conditions. A numerical example shows that design rules based on stress limits are adequate to keep interpass leakage within tolerable limits.

INTRODUCTION

A removable flat cover is frequently the main closure for the tube-side chamber in tubular heat exchangers. Since the cover is a pressure-retaining member, its minimum thickness is established by the governing pressure vessel code (the ASME Boiler and Pressure Vessel Codes in the United States). However, in addition to its pressure-retaining function, the cover plays an implicit role in the thermal-hydraulic performance of multipass heat exchangers. The operating pressure causes the cover to flex, thereby creating a crevice between the pass partition plate and cover bearing surface. The size of the crevice is somewhat diminished by the gasket rib springback, but it is seldom quite eliminated in moderate to large (diameter over 30 in) units. Consequently, the differential interpass pressure causes a portion of the tube-side fluid to short-circuit the heat exchanger. This tube-side stream bypass is somewhat analogous to the so-called E-stream in the shell side [1] of the heat exchanger, and equally deleterious to its thermal performance. Most exchanger design standards, notably TEMA [2], take note of this fact and require that the cover be designed to meet a certain deflection criterion in addition to the code-mandated stress limits. Such deflection-based formulas lead to excessive cover thickness (over those based on stress considerations) for large-diameter heat exchangers. Increased cover thickness implies increased hardware cost and added difficulties in handling during cover removal and installation. Hence it is of some value to seek the optimal cover thickness—one that yields a tolerable performance derating for the minimum thickness. Our object here is to develop general relations to enable the determination of such an optimum. The analysis is restricted to two-tube-pass configurations. The treatment can be extended to multipass geometries, albeit at the expense of increased mathematical complexity.

The solution procedure presented here is intended to be used as a design tool to determine the appropriate cover thickness corresponding to the thermal performance requirements for the heat exchanger. At present, no such correlations exist. Simplifying assumptions made in developing the analysis are stated at the appropriate points in the text.

THERMAL PERFORMANCE

Before determining the bypass stream ΔW_t as a function of basic system parameters, it is appropriate to relate it to the overall exchanger performance. The outcome of the derivation is slightly different, depending on whether the coolant fluid is on the shell side or the tube side. First, let us assume that the coolant is on the tube side. Then the thermal performance of a heat exchanger expressed in terms of its heat duty Q is defined as

$$Q = W_t C_t (t_2 - t_1) \tag{1}$$

or

$$Q = W_t C_t P (T_1 - t_1) \tag{2}$$

where P is the temperature effectiveness, defined as

$$P = \frac{t_2 - t_1}{T_1 - t_1} \tag{3}$$

If the tube-side flow rate is reduced by ΔW_t, the corresponding heat duty $Q - \Delta Q$ is given by

$$Q - \Delta Q = (W_t - \Delta W_t) C_t (P - \Delta P)(T_1 - t_1) \tag{4}$$

From Eqs. (2) and (4) we have, neglecting terms of higher order,

$$\Delta Q = (\Delta W_t C_t P + \Delta P\, W_t C_t)(T_1 - t_1)$$

Hence

$$\frac{\Delta Q}{Q} = \frac{\Delta W_t}{W_t} + \frac{\Delta P}{P} \tag{5}$$

The NTU of the heat exchanger is defined as

$$\eta = \frac{UA}{W_t C_t}$$

Hence

$$\eta - \Delta\eta = \frac{UA}{(W_t - \Delta W_t) C_t}$$

Hence

$$\Delta\eta = \frac{-UA}{C_t}\left(\frac{1}{W_t - \Delta W_t} - \frac{1}{W_t}\right)$$

In the derivation above, we have tacitly assumed that the overall heat transfer coefficient U remains unchanged as the effective tube-side flow rate is perturbed. This assumption is supportable as long as the bypass stream is a small fraction of the total flow rate, i.e., $\Delta W_t / W_t \ll 1$.

Simplifying further, and neglecting terms of higher order, we have

$$\frac{\Delta\eta}{\eta} = \frac{-\Delta W_t}{W_t} \tag{6}$$

Substituting in Eq. (5), we have

$$\frac{\Delta Q}{Q} = -\frac{\Delta\eta}{\eta} + \frac{\Delta P}{P} \tag{7}$$

Or, in differential form,

$$dQ = -\frac{d\eta}{\eta} + \frac{dP}{P}$$

As shown in [3],

$$dP = \frac{\partial P}{\partial \eta} d\eta + \frac{\partial P}{\partial R} dR$$

Hence

$$\frac{dQ}{Q} = \frac{-d\eta}{\eta} + \frac{1}{P}\left[\frac{\partial P}{\partial \eta} d\eta + \frac{\partial P}{\partial R} dR\right] \tag{8}$$

The derivatives $\partial P/\partial\eta$ and $\partial P/\partial R$ follow directly from the characteristic equation for the heat exchanger [3].

Finally, the reduction dR in the heat capacity rate ratio R due to a reduction ΔW_t in tube-side flow rate is given by

$$dR = \frac{W_t C_t}{W_s C_s} - \frac{(W_t - \Delta W_t) C_t}{W_s C_s}$$

or

$$dR = \frac{\Delta W_t C_t}{W_s C_s} = R \frac{\Delta W_t}{W_t}$$

This, along with Eq. (6), yields

$$\frac{\Delta W_t}{W_t} = \frac{dR}{R} = \frac{d\eta}{\eta} \tag{9}$$

Substituting for $d\eta$ and dR in Eq. (8), we have

$$\frac{dQ}{Q} = \frac{\Delta W_t}{W_t} + \frac{1}{P}\left(-\frac{\partial P}{\partial \eta} + \frac{\partial P}{\partial R} R\right)\frac{\Delta W_t}{W_t}$$

or

$$\frac{dQ}{Q} = f\,\frac{dW_t}{W_t} \tag{10a}$$

where

$$f = \frac{1}{P}\left(-\eta\frac{\partial P}{\partial \eta} + R\frac{\partial P}{\partial R}\right) + 1 \tag{10b}$$

If the cooling medium is on the shell side, a similar procedure yields Eq. (10*a*) with the function f defined as

$$f = \frac{1}{P}\left(-\eta\frac{\partial P}{\partial \eta} + R\frac{\partial P}{\partial R}\right) \tag{10c}$$

Thus, for given heat exchanger geometry and operating data, the fractional reduction in Q due to ΔW_t can be readily determined. We will now proceed to quantify ΔW_t for a given mechanical design.

DEFLECTION RELATION

Figure 1 shows a two-tube-pass channel sealed with a gasketed joint. The flat cover is bolted against the raised face flange integral with the channel barrel. This construction is typical of flat-cover bolted joint designs.

Under the prestressed condition, the bolt load W is entirely borne by the ring gasket. The loading on the diametrial rib is neglected in this analysis. Upon pressurization, the bearing load on the gasket drops to a reduced value W_1. The total bolt load W is assumed to remain unchanged from the prestressed to the pressurized condition. This assumption is a basic premise in the ASME Code flange design rules [4], although its validity in many design cases is open to question [5]. We make this assumption here for the purpose of simplifying the structural analysis. In principle, this assumption can be eliminated and the appropriate bolt load for the pressurized condition calculated without altering the rest of the analysis presented here.

Axial equilibrium yields

$$W = W_1 + W_2 \tag{11}$$

where W_1 is the residual gasket load and W_2 is the pressure-induced header load, i.e.,

$$W_2 = \pi r_0^2 p_t$$

where r_0 is the gasket line effective radius and p_t is the tube-side pressure. Assuming that the position of the cover outside the bolt circle offers no rotational restraint, the lateral deflection of the cover under the prestressed condition at radius r is given by [6]

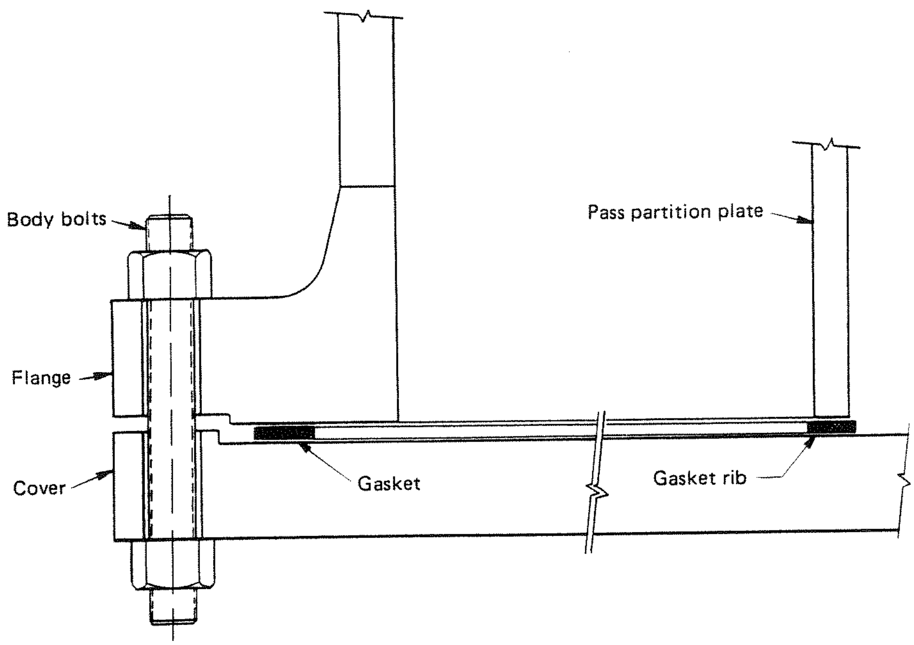

Figure 1 Typical flange-cover geometry.

$$y_s(r) = \frac{\alpha_0 W g(r)}{t^3} \tag{12}$$

where

$$g(r) = \alpha_1(a^2 - r^2) - (r^2 + r_0^2)\ln\frac{a}{r_0} + (r^2 - r_0^2) - \frac{\alpha_2 r_0^2(a^2 - r^2)}{a^2} \tag{13}$$

$$\alpha_0 = \frac{3(1 - \nu^2)}{2\pi E}$$

$$\alpha_1 = \frac{3 + \nu}{2(1 + \nu)} \tag{14a}$$

$$\alpha_2 = \frac{1 - \nu}{2(1 + \nu)} \tag{14b}$$

and a is the bolt circle radius.

The lateral deflection of the cover under the pressurized condition, y_0, consists of two components: $y_1(r)$ due to the ring load W_1 and $y_2(r)$ due to the pressure loading

$$y_0(r) = y_1(r) + y_2(r) \tag{15}$$

where

$$y_1(r) = \alpha_4 W_1 g(r) \tag{16}$$

The expression for $y_2(r)$ may also be found in standard textbooks on plate theory [6],

$$y_2(r) = \frac{\alpha_0 W_2}{8t^3} h(r) \tag{17}$$

where

$$h(r) = 4a^2 - 5r_0^2 + \frac{r^4}{r_0^2} - (8r^2 + 4r_0^2)\ln\frac{a}{r_0} - \frac{4\alpha_2 r_0^2(a^2 - r^2)}{a^2} + \alpha_3(a^2 - r^2) \tag{18}$$

$$\alpha_3 = \frac{8}{1 + \nu} \tag{19}$$

Assuming zero rib gasket springback from the prestressed condition, the total area of the crevice is given by

$$A = 2\int_0^{r_0} (y_0 - y_s)\, dr$$

or

$$A = \frac{2\alpha_0 r_0^3}{t^3}[(W_1 - W)I_1 + W_2 I_2] \tag{20}$$

where

$$I_1 = \frac{1}{r_0^3}\int_0^{r_0} g(r)\, dr$$

or

$$I_1 = \frac{\alpha_1}{3}(3x^2 - 1) - \frac{4}{3}\ln x - \frac{2}{3} - \frac{\alpha_2}{3}(3 - x^{-2}) \tag{21}$$

and

$$I_2 = \frac{1}{r_0^3}\int_0^{r_0} h(r)\, dr$$

or

$$I_2 = 4x^2 - \frac{24}{5} - \frac{20}{3}\ln x - \frac{4\alpha_2}{3}(3 - x^{-2}) + \frac{\alpha_3}{3}(3x^2 - 1) \tag{22}$$

where

$$x = \frac{a}{r_0} \tag{23}$$

I_1 and I_2 are strictly functions of x. Hence the area A can be readily computed from Eq. (20) for a given condition of pressure and cover geometry. Equation (20) shows a striking result. We note that the bypass area A is inversely proportional to the third power of the cover thickness. Hence, the bypass area for any thickness is immediately known if it is computed for one thickness.

The leakage rate ΔW depends on the interpass pressure drop ΔP:

$$\Delta W_t = v_g A \tag{24}$$

where

$$v_g \left(\frac{2g\,\Delta p}{K_g}\right)^{1/2} \tag{25}$$

The loss coefficient K_g depends on the Reynolds number corresponding to the flow through the gap. A detailed iterative method for evaluating K_g and v_g is given in [7]. The pressure drop ΔP due to fluid flow through the tubes can be expressed in the general form

$$\Delta P = (K_1 + K_2 l) \frac{\rho v_t^2}{2g} \tag{26}$$

where K_1 represents loss coefficients due to entry, expansion, and turnaround losses, etc., and

$$K_2 = \frac{4}{d_i} f_F$$

where f_F is commonly referred to as Fanning's friction factor.

Furthermore, $v_t = W_t/A_t$, where A_t is total tube internal flow area per pass.

Hence

$$W_t = \left(\frac{2g\ \Delta P}{K_1 + K_2 l}\right)^{1/2} A_t \tag{27}$$

From Eqs. (24) and (27), we have

$$\frac{\Delta W_t}{W_t} = \left(\frac{K_1 + K_2 l}{K_g}\right)^{1/2} \frac{A}{A_t} \tag{28}$$

The term in parentheses on the right-hand side of Eq. (28) is a function of pass partition plate geometry, tube length, and fluid properties.

Combining Eq. (28) with Eq. (20), we have

$$\frac{\Delta W_t}{W_t} = F \frac{r_0^3}{t^3} \tag{29}$$

where F is independent of cover thickness.

$$F = \frac{2\alpha_0}{A_t} [(W_1 - W)I_1 + W_2 I_2] \left(\frac{K_1 + K_2 l}{K_g}\right)^{1/2} \tag{30}$$

For a given exchanger geometry and fluid flow condition, F can be evaluated from Eq. (30). The bypass flow rate ratio is strongly dependent on the cover thickness. Equation (29) can be used in conjunction with Eq. (10*a*), (10*b*), or (10*c*) to determine the optimal t.

Having developed the necessary relations to determine the thermal performance deterioration for a given cover thickness, we can now proceed to make a quantitative evaluation of the current design practice.

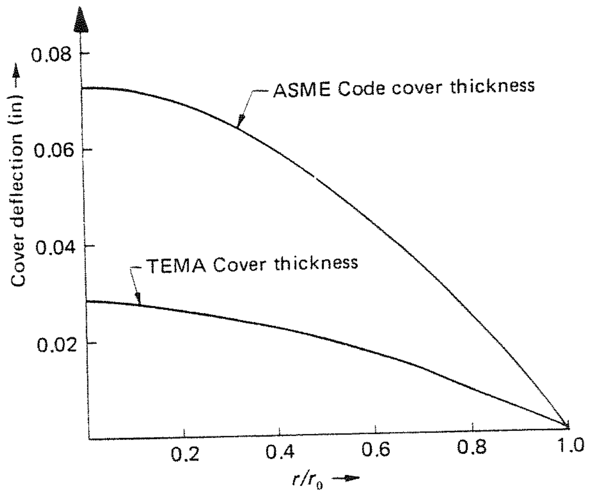

Figure 2 Cover deflection profile for example problem.

EXAMPLE

We consider a one-shell-pass, two-tube-pass heat exchanger operating at a temperature effectiveness $P = 0.54$, $\eta = 1.0$, and thermal flow rate ratio $R = 0.5$. The gasket radius r_0 is 20 in (50.8 cm) and the bolt circle radius a is 20.75 in (52.7 cm). The cover is bolted to the channel flange through 40 bolts of $\frac{3}{4}$ in (19 mm) nominal diameter and 25,000 psi (172.4 MPa) allowable stress. The cover material has an allowable stress of 17,500 psi (120.6 MPa) and a Young's modulus E of 28.2×10^6 psi (194.4 GPa). The fluid pressure is 200 psig (1379 kPa gage).

The cover thickness is calculated as 2.34 in, using the rules of the ASME Code [4], and 3.21 in, using the deflection-based formula of TEMA [2]. The tube-side flow rate W_t is 3×10^6 lb/h (1.36×10^6 kg/h). The deflected cover diametral sections for TEMA and code cover thicknesses are shown in Fig. 2. The discrepancy between the two deflected cover profiles is obvious. Table 1 gives the computed quantities for both thicknesses from the analysis in this paper. It is seen that the loss in heat duty is less than 1% if the cover thickness corresponds to the ASME Code formula; it is approximately 0.33% for the cover thickness based on TEMA. The added cost of a heavier cover does not appear necessary in this case.

It would be incorrect to draw any general conclusions solely on the basis of the example above. Each case warrants its own analysis and conclusions. Our object here is to develop an

Table 1 Comparison of TEMA and Code-Based Covers for the Example Problem

Quantity	Code [4]	TEMA [2]	Equation or reference
Cover thickness t	2.34	3.21	–
Leakage area, in^2	1.67	0.65	Eq. (20)
Leakage velocity, ft/s	30.2	27.5	[7]
Leakage flow rate ΔW_t, lb/h	78,500	27,900	Eq. (24)
Percent leakage $100\ \Delta W_t/W_t$	2.62	0.93	–
$\partial P/\partial \eta$	0.263	0.263	[3]
$\partial P/\partial R$	−0.169	−0.169	[3]
f	0.36	0.36	Eq. (10*b*)
dQ/Q	0.94×10^{-2}	0.33×10^{-2}	Eq. (10*a*)

analytic tool for determining the most suitable cover thickness for a specific heat exchanger. The example problem is used to demonstrate the application of the final equations. The general interpass leakage problem can be solved using public domain computer codes, such as LAPCOV [8].

CONCLUSION

A method for computing the reduction in heat exchanger thermal performance due to interpass leakage in the tube side has been derived. This leakage may occur at the pass partition plate to tube sheet joint or the pass partition plate to flat cover joint. A formalism for quantifying the former has been published [7]. In this paper, a structural model for the flat cover is proposed which yields relatively straightforward relations for computing the magnitude of interpass leakage due to cover flexural deflection. The final expressions are presented with cover plate thickness as the variable. It is shown that the leakage rate is inversely proportional to the third power of cover plate thickness. The solution technique presented here may be used to establish the correct cover thickness in relation to the heat duty requirements.

NOMENCLATURE

A	leakage gap area
A_t	total in-tube flow area per tube pass
a	bolt circle radius
C_t	specific heat of tube-side fluid
C_s	specific heat of shell-side fluid
d_i	inside diameter of tubes
E	Young's modulus of cover material
g	gravitational constant
K_1, K_2	hydraulic loss coefficients [Eq. (26)]
l	total tube length
NTU	number of heat transfer units
P	temperature effectiveness [Eq. (3)]
p_t	tube-side pressure
ΔP	interpass pressure drop
Q	heat duty
ΔQ	incremental change in heat duty due to interpass leakage
R	thermal flow rate ratio
ΔR	incremental change in R due to interpass leakage
r_0	effective gasket circle radius
r	radial coordinate (from center of cover)
t	thickness of cover
t_1	coolant fluid inlet temperature
t_2	coolant fluid outlet temperature
T_1	hot fluid inlet temperature
v_g	leakage stream velocity through interpass gap
v_t	in-tube flow velocity
W	total bolt load
W_1	residual gasket load under pressurized condition
W_2	header load due to channel pressure
W_s	shell-side stream flow rate
W_t	tube-side flow rate
ΔW_t	leakage rate from tube pass 1 to pass 2 due to bowing of channel cover
x	ratio of bolt circle radius to effective gasket radius
$y_0(r)$	cover deflection as a function of radius under pressurized condition
$y_1(r)$	cover deflection due to W_1 under pressurized condition
$y_2(r)$	cover deflection due to W_2
$y_s(r)$	cover deflection under seating condition (function of r)
ρ	tube-side fluid density
ν	Poisson ratio of cover material
η	number of transfer units (NTU)

REFERENCES

[1] Taborek, J., Design Methods for Heat Transfer Equipment—A Critical Survey of the State-of-the-Art, in *Heat Exchanger: Design and Theory Sourcebook*, eds.

N. Afgan and E. R. Schlünder, Hemisphere, Washington, D.C., 1974.
[2] *Standards of Tubular Exchanger Manufacturers Association*, 6th ed., TEMA, New York, 1978.
[3] Singh, K. P., Some Fundamental Relationships for Tubular Heat Exchanger Thermal Performance, *Trans. ASME, J. Heat Transfer*, vol. 103, pp. 573–578, 1981.
[4] *ASME Boiler and Pressure Vessel Code*, sect. VIII, div. 1, 1980, New York.
[5] Singh, K. P. and Soler, A. I., Design Parameters Affecting Bolt Load in Ring Type Gasketed Joints, *Trans. ASME, J. Pressure Vessel Technol.*, vol. 105, pp. 11–13, 1983.
[6] Timoshenko, S. P., *Theory of Plates and Shells*, 2d ed., McGraw-Hill, New York, 1959.
[7] Singh, K. P., Study of Bolted Joint Integrity and Inter-Tube-Pass Leakage in U-Tube Heat Exchangers, Part II: Applications, *Trans. ASME, J. Eng. Power*, vol. 101, pp. 16–22, 1979.
[8] Singh, K. P. and Soler, A. I., *Mechanical Design of Heat Exchangers and Pressure Vessel Components*, chapter 12, Arcturus Publishers, Cherry Hill, N.J., 1984.