PERFORMANCE EVALUATION OF A
POWER RECOVERY WASTE HEAT BOILER

A. J. Szady

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OAK RIDGE GASEOUS DIFFUSION PLANT
OAK RIDGE, TENNESSEE

*prepared for the U.S. ATOMIC ENERGY COMMISSION
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Oak Ridge Gaseous Diffusion Plant
Union Carbide Corporation
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ABSTRACT

A computer program has been developed to numerically simulate the performance of an exhaust heat, power recovery boiler, operating in a refrigerant Rankine cycle. The boilers studied are either of the tube-and-fin or the plate-and-fin design. Each boiler or cooler is made up of a preheat section in series with the actual boiling section. Only one exhaust gas pass is considered; however, the coolant side can have any number of specified passes in the preheat section with one final boiling pass. Each individual pass may contain any number of tube rows.

The modes of heat transfer considered are: (1) forced convection - liquid and gas (2) conduction - fin-and-tube, and (3) boiling and sub-cooled boiling. The cooler boiling temperature is determined for any specified waste heat load and cycle condensing temperature. Each tube row within the boiler and preheater is considered individually and broken into a number of specified increments. A heat balance is computed for each increment, allowing a complete temperature profile to be determined for the exhaust gas and refrigerant within the cooler. An example case is presented.

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CONTENTS

	<u>Page</u>
NOMENCLATURE	7
INTRODUCTION	9
SUMMARY	9
BOILER DESCRIPTION	10
OVERALL PROGRAM LOGIC	10
BASIC HEAT TRANSFER ANALYSIS	14
NUMERICAL ANALYSIS	17
Boiling With Net Vapor Generation, Starts at Inlet of	
Boiling Pass	17
Boiling Section Too Large, Subcooled Liquid Enters to Reduce	
Boiling Area	20
Boiling Section Too Small, Additional Area Taken from	
Adjoining Preheat Pass	22
Subcooled Passes	22
COOLANT-SIDE PRESSURE DROP	23
EXAMPLE CASE	24
COOLER DESCRIPTION	24
OPERATING CONDITIONS	24
REFERENCES	27

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NOMENCLATURE

A_I	= inside heat transfer area per increment.
A_O	= outside heat transfer area per increment.
A_R	= area ratio.
C_{pc}	= specific heat coolant.
C_{pg}	= specific heat gas.
E_i	= error in heat balance.
E_{Ei}	= error in pressure drop.
h_{IB}	= inside boiling coefficient.
h_{IC}	= inside convective coefficient.
h_O	= outside convective gas coefficient.
$\dot{m}_B$	= coolant flow in boiling section.
$\dot{m}_{Ci}$	= coolant flow rate per row in boiling section.
$\dot{m}_g$	= gas flow rate per increment.
$\dot{m}_S$	= coolant flow subcooled section.
Q_I	= heat transfer per increment as evaluated on coolant side.
Q_L	= latent heat load.
Q_O	= heat transfer per increment as evaluated on gas side.
Q_S	= sensible heat load.
Q_{TB}	= $\sum Q_O$ total heat transferred in boiling section.
S	= slope evaluated from chord.
t_I	= coolant inlet temperature.
T_I	= gas inlet temperature.
$t_{I,i}$	= trial value of increment coolant exit temperature.
t_O	= coolant outlet temperature.

T_o = gas outlet temperature.

$T_{o,i}$ = trial value of increment gas exit temperature.

t_v = boiling temperature.

t_w = wall temperature.

$t_{w,i}$ = trial value of wall temperature.

X_o = average cooler exit quality.

X_{pi} = trial value of quality entering boiler section.

Δh = change in coolant enthalpy corresponding to sensible heat load.

Δp_{Ri} = row pressure drop in boiling section.

GREEK SYMBOLS

λ = latent heat.

Δ = change.

η_g = gas side fin efficiency.

η_c = coolant side fin efficiency.

SUPERSCRIPIT

' = reevaluation by alternate expression.

SUBSCRIPTS

i = iteration number or row number.

N = number of parallel rows in boiling section.

PERFORMANCE EVALUATION OF A POWER RECOVERY WASTE HEAT BOILER

INTRODUCTION

Within the chemical processing industry, there are many processes which reject large quantities of low-grade waste heat to the atmosphere; an example common to many industries is the exhaust gas from industrial gas turbines. With the rapid increase in power costs presently being experienced, power recovery from waste heat is now becoming economically feasible. One method of power recovery that has received attention both in this country and abroad is the familiar Rankine cycle employing a refrigerant as the working fluid. The key to the economic operation of such cycles is the development of a low-cost refrigerant boiler which can produce high-pressure vapor while exchanging heat with a low-grade source. To facilitate the development of such boilers, a computer program has been developed to numerically simulate boiler performance. Performance simulation is important during development for several reasons:

1. Overall cycle efficiency can be more accurately predicted,
2. Boiling and flow stability can be studied by determining the point at which boiling actually starts,
3. Predicted temperature distributions within the boiler are necessary for a complete boiler thermal stress analysis, and
4. Off-design performance can be evaluated.

SUMMARY

The boiler performance simulation is only part of the overall power recovery system evaluation. Using the boiler simulation computer code, a family of performance lines can be generated for a range of boiler exit qualities and cycle condensing temperatures, which must be integrated with similar results for the flow characteristics of the refrigerant expansion turbine and pump together with the condensing performance of the heat sink. The results will determine the cycle performance and give items such as (1) boiling and condensing temperature; (2) boiler exit quality; (3) coolant flows; and (4) power recovered.

In the case of the natural circulation boiler shown in figure 2, the boiler exit quality can be determined by a cross plot of boiler pressure drop versus quality and the pressure drop characteristics of the downcomer. The performance of the boiler feed pump is needed for forced flow.

It has been stated that tube wall dry out with a transition from annular to mist flow is assumed to take place at some specified quality. Provisions have been made in the program to calculate the dry out quality as a function of tube length, diameter and heat flux, provided such correlations are available for the particular refrigerant in use.

The actual computer program, written in Fortran IV, is quite general in that all heat transfer correlations and fluid property relations are contained in separate subroutines which can be easily changed or modified, thus making the code quite versatile.

BOILER DESCRIPTION

The types of boilers considered are of the multipass cross-flow design shown schematically in figures 1, 2, and 3. Each boiler or cooler consists of the actual boiling section together with the refrigerant preheat section. The boiling section consists of only one coolant pass; however, the pass can contain any number of tube rows in parallel. The preheat section consists of multiple coolant passes containing any number of parallel rows within each pass.

Owing to the rather poor heat transfer characteristics of the refrigerants and the availability of only low-temperature waste heat, boiler superheat is not considered. Consequently, the refrigerant or coolant is assumed to leave the boiling section at some specified average exit quality. The entrained liquid can be returned to either the entrance of the boiling section with forced or natural circulation as shown in figures 1 and 2, or mixed with the coolant entering the preheat section as in figure 3. In the latter case both the preheat and boiling sections have the same coolant flow. The boiler heat transfer surface can be made up of circular fin and tubes, plate fin and tubes, or plate and fins.

OVERALL PROGRAM LOGIC

The steady state performance of a given boiler design is determined as a function of the following:

1. Waste gas flow,
2. Waste heat gas temperature,
3. Desired waste heat gas temperature at boiler exit,
4. Refrigerant temperature at boiler entrance (approximate cycle condensing temperature), and
5. Boiler exit quality.

Figure 1
LIQUID RETURNED TO BOILER INLET

Figure 2
NATURAL RECIRCULATING BOILER

Figure 3
LIQUID RETURNED TO COOLER INLET

The object of the computer program is to determine the coolant boiling temperature, and the temperature profiles within the cooler for the specified conditions.

Given the coolant entering temperature, average boiler exit quality, and mode of liquid recycle, the amounts of sensible and latent heat, together with coolant flows can be determined by the following:

$$\frac{Q_L}{Q_T} = \frac{\lambda}{\lambda + \Delta h} \quad \begin{array}{l} \text{(entrained liquid returned to either boiler} \\ \text{or preheat section),} \end{array} \quad (1)$$

$$\dot{m}_B = \frac{Q_T}{X[\lambda + \Delta h]}, \quad (2)$$

$$\dot{m}_S = \dot{m}_B \quad \text{(entrained liquid returned to preheat section), and} \quad (3)$$

$$\dot{m}_S = X\dot{m}_B \quad \text{(entrained liquid returned to boiler inlet).} \quad (4)$$

The performance analysis is based on determining the amounts of sensible and latent heat for an assumed boiling temperature and specified exit quality. The performance of the designated boiling section is then computed by using a step-by-step finite difference analysis for each boiler row. If it is found that the boiling section is too small to handle the latent heat load implied by the assumed boiling temperature, additional surface area is taken from the adjoining preheat pass. On the other hand, if the boiling section is too large, subcooled liquid is allowed to enter the boiling pass. In this manner, enough heat transfer area is always taken to handle the latent heat load at the assumed boiling temperature. The performance of the remaining cooler surface is then determined and compared against the sensible heat load. If agreement is not established within 1%, a new boiling temperature is chosen and the entire analysis repeated. This procedure allows the point at which boiling actually starts to be determined, whether it be somewhere within the designated boiling section or in a preheat pass.

When the tube wall or coolant channel wall temperature within any increment exceeds the boiling temperature, either boiling or subcooled boiling takes place, depending on the bulk temperature at that point. It is assumed that subcooled boiling does not produce net vapor generation; consequently, there is no contribution to the latent heat load. Above a certain predetermined quality, tube wall dry out is assumed to take place, at which point mist flow is established with a subsequent decrease in the boiling coefficient.

BASIC HEAT TRANSFER ANALYSIS

The numerical analysis is based on subdividing each tube row into a number of specified increments as shown in figure 4. A heat balance is computed for each increment using the following generalized equations:

Figure 4
FINITE ROW ELEMENT

$$Q_I = \dot{m}_S C_{pc} (t_o - t_I) \text{ sensible heat,} \quad (5)$$

$$Q_o = \dot{m}_g C_{pg} (T_I - T_o), \quad (6)$$

$$Q_I = \dot{m}_{Ci} (\Delta X) \lambda \text{ latent heat,} \quad (6A)$$

$$Q_o = UA_o \left(\frac{T_I + T_o}{2} - \frac{t_I + t_o}{2} \right), \text{ and} \quad (7)$$

$$U = \frac{1}{\frac{1}{h_o \eta_g} + \frac{A_R}{h_I \eta_c}} \quad (8)$$

Starting with the first increment in the first row, where the gas inlet temperature and flow rate are known and the coolant temperature and flow rate are set, the heat balance equations are solved successively for each increment in the row. The gas temperatures leaving the first row then become the known entering temperatures for the second row. This stepwise analysis is carried out for the entire cooler. Since each tube row is subdivided into M increments, the gas flow is also split uniformly into M flows. Any lateral flow that might take place within the cooler is not accounted for.

The outside gas coefficient h_o is determined for each increment by using appropriate Colburn j factors and surface parameters stored in a surface library routine. This library contains data available from Kays and London¹ for both fin-and-tube and plate-and-fin surfaces. Gas properties used to evaluate the Stanton, Reynolds, and Prandtl numbers are determined at the average gas temperature within the increment. A row-by-row correction factor has been incorporated to correct the infinite tube bank j-data given in Kays and London. This correction is applied to each row as the analysis steps through the cooler from the first to the last row.

Gas and coolant side surface fin efficiency is calculated in a separate subroutine capable of handling the following fin types:

1. Circular,
2. Plate fin-and-tube,
3. Plate-and-fin - compacts, and
4. Special fin shapes.

Fin dimensions are also stored in the surface library. Compact surfaces can be made up of either rectangular or triangular fins having gas-side to coolant-side passage ratios ranging from one to four.

The inside surface coefficient h_I can be either: (a) forced convection for subcooled liquid or vapor mist flow, in which case the Sieder-Tate equation is used; (b) subcooled boiling evaluated by a combination of the Foster-Grief pool boiling relation and Sieder-Tate forced convection equation; (c) boiling, evaluated by the Chen² correlation.

As pointed out previously, the analysis is begun by assuming a boiling temperature; this together with the specified exit quality establishes the coolant flows for both the boiler and preheat sections, thus enabling the inside coefficients to be evaluated. The performance of the designated boiling section is then determined by an increment-by-increment finite-difference analysis starting with the first row, where the entering gas temperature is known and the entering coolant quality is set at zero. After the performance of this section is established, three possible branches exist: (1) the boiling section can exactly handle the latent heat load in which case the preheat section is evaluated next; (2) the boiling pass is too large for the assumed boiling temperature so that subcooled liquid must enter to reduce the boiling area; and (3) the boiling section is too small and must be supplemented with area from the adjoining preheat pass.

After the performance of the entire cooler has been determined, its heat transfer capability at the assumed boiling temperature is compared against the required performance as specified by the gas exit temperature. The boiling temperature for the next iteration is then adjusted depending on whether the calculated performance was higher or lower than that required.

NUMERICAL ANALYSIS

BOILING WITH NET VAPOR GENERATION STARTS AT INLET OF BOILING PASS

This section is characterized by a constant coolant temperature t_v and an entering quality set at zero. The finite difference equations are:

$$Q_o = \dot{m}_g C_{pg} (T_I - T_o), \quad (6)$$

$$Q_o = h_o A_o \eta_g \left(\frac{T_I + T_o}{2} - t_w \right), \quad (9)$$

$$h_o = f(G, T_{avg}), \quad (10)$$

$$\Delta X = \frac{Q_o}{\dot{m}_{ci} \lambda}, \quad (11)$$

$$Q_I = h_{IB} A_I \eta_c (t_w - t_v), \quad (12)$$

$$h_{IB} = f(t_v, t_w - t_v, \dot{m}_{ci}, X_{avg}), \quad (13)$$

$$h_{IC} = f(\dot{m}_{Ci}, t_{avg}), \text{ and} \quad (13A)$$

$$Q_o - Q_I = 0. \quad (14)$$

The unknowns in this system are Q_o , Q_I , t_w , T_o , h_o , h_{IB} , and X_{avg} . To solve this nonlinear system, an iterative routine is used where a trial value of T_o is chosen. Equation (14) then becomes

$$E_i = Q_o - Q_I. \quad (15)$$

The correct value of T_o is found when $E_i \approx 0$ or equation (14) is satisfied within a reasonable limit.

The above system of equations is coupled such that if some value for T_o say $T_{o,i}$ is chosen, all remaining expressions can be evaluated resulting in a value for E_i in equation (15). In essence, this system can be reduced to but one equation relating E_i as a function of $T_{o,i}$. The method of false position³ is used to find a zero of this nonlinear function. A chord of the function E_i is found by assuming a second value for $T_{o,i}$ 5° higher than the first value. The slope of the chord can then be evaluated as

$$S = \frac{E_i - E_{(i+1)}}{T_{o,i} - T_{o,(i+1)}}, \quad (16)$$

and the new estimate for T_o

$$T_{o,(i+2)} = T_{o,(i+1)} - \frac{E_{(i+1)}}{S}. \quad (17)$$

This procedure is shown graphically in figure 5.

The slope S is continually reevaluated as new values of $T_{o,i}$ are computed, the last two values of $T_{o,i}$ used in equation (16). This routine is repeated until the error E_i is reduced to within 1% of Q_o . For the first increment analyzed, the initial estimate for T_o ($T_{o,1}$) is computed by assuming an inside coefficient h_{IB} of $1000 \frac{\text{Btu}}{\text{hr-sq ft-}^\circ\text{F}}$. This allows equations (6), (7), (8), and (10) to be solved simultaneously for $T_{o,1}$. (The outside coefficient h_o is evaluated at $T_{I,1}$.) For subsequent increments, the initial guess of 1000 is replaced by the actual inside coefficient as determined from the preceding increment. The method of false position converges rapidly in this application since the initial trial value of $T_{o,i}$ as computed is generally close to the correct value.

Figure 5
METHOD OF FALSE POSITION

The above analysis is carried out for each increment in the boiling section. At the end of the routine, the summation of Q_o is set equal to Q_{TB} which is then compared against the latent heat load Q_L previously established by the assumed boiling temperature. At this point one of the three branches previously referred to can be taken.

BOILING SECTION TOO LARGE, SUBCOOLED LIQUID ENTERS TO REDUCE BOILING AREA

The object here is to determine the proper coolant inlet temperature to the boiler section which still allows the latent heat to be removed. That is, some coolant inlet temperature (t_I) must be determined so that

$$Q_{TB} = \dot{m}_{Ci} C_{pc} (t_v - t_I) + \dot{m}_{Ci} \lambda X_o, \quad (18)$$

where

$$\dot{m}_{Ci} \lambda X_o = \text{required latent heat load}, \quad (19)$$

is satisfied for the entire boiling pass.

Equation (18) is solved by assuming some coolant inlet temperature $t_{I,i}$. The performance of each increment in the boiling section is then computed and the summation of Q_o again set equal to Q_{TB} . A check on the assumed value of $t_{I,i}$ is computed from

$$t'_{I,i} = t_v - (Q_{TB}/\dot{m}_{Ci} - X_o \lambda)/C_{pc}. \quad (20)$$

If $t'_{I,i} > t_v$, the initially assumed value of $t_{I,i}$ is too low, the next trial value of $t_{I,i}$ is computed as

$$t_{I,(i+1)} = t_v - 0.5 (t_v - t'_{I,i}). \quad (21)$$

However, if $t'_{I,i} < t_v$ but $t_{I,i} \neq t'_{I,i}$, then

$$t_{I,(i+1)} = \frac{t_{I,i} + t'_{I,i}}{2}. \quad (22)$$

This modified method of successive substitution is carried out until agreement between $t_{I,i}$ and $t'_{I,i}$ is found within 0.2 of a degree.

The finite difference equations for the boiling increments are solved as stated in the first section of NUMERICAL ANALYSIS. The equations for the subcooled increments,

$$Q_o = \dot{m}_p C_{pg} (T_I - T_o) = \dot{m}_{Ci} C_{pc} (t_o - t_I), \quad (5)$$

$$Q_o = UA_o \left(\frac{T_I + T_o}{2} - \frac{t_o + t_I}{2} \right), \text{ and} \quad (7)$$

$$U = \frac{1}{\frac{1}{h_o \eta_g} + \frac{A_R}{h_I \eta_c}} \quad (8)$$

can be solved directly in closed form for t_o , T_o , and Q_o . One additional iteration, however, is required to evaluate both the inside and outside coefficients at approximately the average temperature within the increment. This also allows the viscosity ratio term in the Sieder-Tate equation to be evaluated.

If the tube wall temperature for a subcooled increment is found to exceed the boiling temperature, subcooled boiling is assumed to take place, and consequently the inside coefficient can no longer be evaluated by using only the Sieder-Tate equation. In this case, forced convection is supplemented by boiling which is evaluated with the Foster-Grief pool boiling correlation. The analysis of a subcooled boiling increment is similar to a boiling increment in that a tube wall temperature t_w must be determined to evaluate the inside coefficient. In this case equation (12) is replaced by

$$Q_I = h_{IC} A_i \eta_c \left(t_w - \frac{t_I + t_o}{2} \right) + h_{IB} A_i \eta_c (t_w - t_v). \quad (23)$$

The overall iteration technique for subcooled boiling increments differs from that of the boiling increments since a trial value of the wall temperature $t_{w,i}$ is taken directly without working back from a trial value of T_o . By assuming a trial value for the wall temperature $t_{w,i}$, Q_I is determined from (5), (13), (13A), and (23) which then allows a reevaluation of $t_{w,i}$ through equations (6), (9), and (14) resulting in a new value of the wall temperature $t'_{w,i}$. The wall temperature at the start of the iterative routine is well bounded since it must be below the wall temperature calculated using only forced convection and above the boiling temperature t_v . Therefore, as a first trial value

$$t_{w,i} = \frac{t_w \text{ (forced convection only)} + t_v}{2}. \quad (24)$$

If $t'_{w,1} < t_{w,1}$ the trial value $t_{w,1}$ was too high and consequently, $t_{w,1}$ becomes an upper bound, and

$$t_{w,2} = \frac{t_{w,1} + t_v}{2}. \quad (25)$$

Conversely, if $t'_{w,1} > t_{w,1}$ the trial value of $t_{w,1}$ was too low and $t_{w,1}$ becomes a lower bound, and

$$t_{w,2} = \frac{t_w \text{ (forced convection only)} + t_{w,1}}{2}$$

This half interval search method is carried out until the values of $t_{w,i}$ and $t'_{w,i}$ agree with a specified tolerance.

BOILING SECTION TOO SMALL, ADDITIONAL AREA TAKEN FROM ADJOINING PREHEAT PASS

Since the boiling coefficient is a function of quality, a boiler section entering quality X_p must be found so that

$$X_p = X_o - \frac{Q_{TB}}{(\lambda) (\dot{m}_{Ci})} \quad (26)$$

is satisfied for the entire boiling pass.

As a first estimate X_{p1} can be evaluated by equation (26) using the previous value of Q_{TB} as determined when the coolant entered the boiling section at a quality of zero. Then using this inlet quality X_{p1} , the boiler section performance is reevaluated as outlined under the first section of NUMERICAL ANALYSIS. A new quality X'_{p1} is determined from equation (26) and compared against the previous value. The method of successive substitution is used until agreement between X_{p1} and X'_{p1} is found within 5%. The adjoining preheat pass is then handled as a boiling pass with subcooled liquid entering at some unknown temperature, the evaluation of which is handled exactly as outlined under the second section of NUMERICAL ANALYSIS.

SUBCOOLED PASSES.

After enough surface area has been taken to satisfy the latent heat load, the remaining surface area constitutes the real preheat section which must handle the sensible heat load. Working back into the cooler, the coolant exit temperature for the pass in question is known since it corresponds to the coolant entering temperature previously calculated for the pass above. Therefore, the coolant temperature entering the pass must be found so that the calculated exit agrees with the entering temperature for the preceding pass within a specified tolerance. A trial value for the coolant entering temperature is assumed and the pass performance evaluated, resulting in a pass coolant exit temperature. The second trial value is then adjusted depending on whether the exit temperature calculated from the first trial was higher or lower than required.

The finite difference equations are the same as those described for subcooled increments in the second section of NUMERICAL ANALYSIS. The inside coefficient can be either forced convection or subcooled boiling depending on the wall temperature.

COOLANT-SIDE PRESSURE DROP

The change in the boiling temperature associated with the pressure drop in the boiling section is not accounted for in the analysis. However, the actual pressure drops are needed to determine the coolant flow distribution between the parallel tube rows in the boiling pass. After the correct boiling temperature has been found, the coolant-side pressure drop is calculated for each tube row in the boiling section. An increment by increment analysis is carried out by using the Martinelli Nelson⁴ correlation. In order to achieve a proper flow distribution, the pressure drops must be equal. The heat transfer performance of the boiling section, and consequently the final boiling temperature is a function of the coolant flow distribution. Therefore, some coolant flow split must be assumed at the very start; a uniform split generally is quite satisfactory. After the entire performance analysis is carried out based on the assumed split, and the pressure drops for the parallel boiler rows computed, an improved split is determined for the next iteration. That is, a split which will tend to equalize the pressure drops. After the results of the uniform flow distribution are obtained the following errors can be computed.

$$\begin{aligned}
 \Delta p_{R1} - \Delta p_{R2} &= E_{R1} \\
 \Delta p_{R1} - \Delta p_{R3} &= E_{R2} \\
 &\vdots \\
 &\vdots \\
 \Delta p_{R1} - \Delta p_{RN} &= E_{RN} \\
 \dot{m}_B - \dot{m}_{C1} - \dot{m}_{C2} - \dot{m}_{CN} &= E_{R(N+1)}
 \end{aligned} \tag{27}$$

where $\Delta p_{RN} = f(\dot{m}_{Ci}, t_v, Q_L)$.

Using the Newton-Raphson method, the errors can be approximated by

$$\begin{aligned}
 \frac{\partial(\Delta p_{R1})}{\partial \dot{m}_{C1}} (\Delta \dot{m}_{C1}) - \frac{\partial(\Delta p_{R2})}{\partial \dot{m}_{C2}} (\Delta \dot{m}_{C2}) &= E_{R1} , \\
 \frac{\partial(\Delta p_{R1})}{\partial \dot{m}_{C1}} (\Delta \dot{m}_{C1}) - \frac{\partial(\Delta p_{R2})}{\partial \dot{m}_{C3}} (\Delta \dot{m}_{C3}) &= E_{R2} , \\
 \frac{\partial(\Delta p_{R1})}{\partial \dot{m}_{C1}} (\Delta \dot{m}_{C1}) - \frac{\partial(\Delta p_{RN})}{\partial \dot{m}_{CN}} (\Delta \dot{m}_{CN}) &= E_{RN} , \text{ and} \\
 -\Delta \dot{m}_{R1} - \Delta \dot{m}_{C2} - \dots - \Delta \dot{m}_{CN} &= E_{R(N+1)} .
 \end{aligned} \tag{28}$$

If values for the partial derivatives can be evaluated, the resulting linear equations can be solved simultaneously for the corrections ($\Delta \dot{m}_{CN}$) to the previously assumed split. The partial derivatives however are difficult to obtain, therefore they must be approximated. The approximation is made by varying the uniform split slightly. This results in two values of pressure drop for two flow distributions at nearly the same boiling temperature. The slopes computed from these values allow the linear equations (SYSTEM 28) to be solved for the corrections.

This procedure has been found to converge to an acceptable solution within two or three iterations. When dealing with refrigerants that exhibit a large density ratio and consequently may show a decrease in pressure drop with increasing mass flow this procedure may not converge as rapidly.

EXAMPLE CASE

The results for the numerical case shown schematically in figure 6 are based on the following specification:

COOLER DESCRIPTION

Heat transfer surface	- 5/8-in. Wolverine type H/R fin tubes arranged on a staggered array with a longitudinal and transverse pitch of 1.75 and 1.55 in., respectively. Surface performance based on the data of Jameson. ⁵
Boiler height or tube length	- 10 ft
Boiler width	- 9 ft
Coolant tubes per row	- 69
Parallel tube rows in boiling section	- 4
Number of subcooled passes	- 2
Parallel tube rows per subcooled pass	- 2

OPERATING CONDITIONS

Working fluids	- Air and R-114
Air inlet temperature	- 475°F
Desired air exit temperature	- 215°F

Figure 6
EXAMPLE CASE

Air flow rate	- 290,000 lb/hr
Cycle condensing temperature	- 85°F
Average exit quality	- 0.65
Tube dry out quality	- 0.80

The computed temperature profile shown in the figure indicates that the predicted boiling temperature is 210°F and that the initiation of actual boiling starts well into the boiling pass with 23°F of inlet subcooling at the pass inlet. Subcooled boiling, however, starts with the first increment in the fifth row (increment numbers for each row follow the direction of coolant flow). At this location the tube wall temperature has exceeded the boiling temperature (210°F). This increment, as well as all the other increments on the right side, are influenced by the dry out of the first tube row. Dry out in this case occurred after the coolant quality exceeded the specified tube wall dry out value of 0.80. The remaining rows however did not dry out and show a range in exit quality from 0.68 to 0.40. The almost stepwise change in the tube wall temperature at the dry out increment is somewhat extreme since axial conduction along the tube has not been taken into account. The coolant temperature entering the cooler (131°F) is the result of mixing the returning condensate at 85°F with the externally separated liquid at 210°F. In cases where the separated liquid is returned to the entrance of the boiling section, the coolant enters the cooler at the specified cycle condensing temperature.

The coolant flow distribution in the boiling section for this case was specified as 30, 28, 23, and 19% from the first to the fourth row, respectively. Based on this distribution, the individual heat loads, and a boiler coolant flow rate of 396,300 lb/hr, the computed pressure drops for the parallel rows in the boiling section are:

$$\Delta p_{R1} = 14.0 \text{ ft,}$$

$$\Delta p_{R2} = 11.9 \text{ ft,}$$

$$\Delta p_{R3} = 9.6 \text{ ft, and}$$

$$\Delta p_{R4} = 8.3 \text{ ft.}$$

These losses include elevation, acceleration, friction, and entrance and exit effects. The greater loss in the first row is due to a higher friction loss, which can be compensated for by slightly decreasing the flow in that row for the next iteration. However, a decrease of mass flow in the first row will cause further dry out; consequently, the final boiling temperature will be slightly lower than 210°F.

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