Computer programs aid design work

Fig. 1—

Shell-and-tube output — page 1

```
INQUIRY NO.
ITEM NO.     X-109
SIZE   33.- 192.      TYPE     U-TUBE      PARALLEL    1 SERIES      2
SQ.FT. PER UNIT     6379.     SHELLS PER UNIT     2.    SQ.FT. PER SHELL    3189.

                         PERFORMANCE OF 1 UNIT
              SHELL SIDE                                     TUBE SIDE

FLUID CIRCULATED
TOTAL FLOW                114191. LB/HR                     1023000. LB/HR
VAPOR                     114191. LB/HR                           0. LB/HR
LIQUID                         0. LB/HR                     1023000. LB/HR
STEAM                          0. LB/HR                           0. LB/HR
VAP. OR COND.             114191. LB/HR                           0. LB/HR
STEAM COND.                    0. LB/HR                           0. LB/HR

GRAVITY IN/OUT             0.000/0.720      AT 60 F         1.000/1.000
AVG. VISCOSITY             0.385  CP                        0.820  CP
VAP MOL WT IN/OUT          96.007   0.00                    0.00/  0.00

TEMPERATURE IN               290.0 F                        70.0 F
TEMPERATURE OUT              100.0 F                       100.0 F
PRESSURE (PSIA)               16.00                         50.00
NO. OF PASSES    DIVIDED FLOW                                2.
VELOCITY                                                     5.1 FT/SEC

PRESSURE DROP   ALLOW  5.00  CALC.  4.83          ALLOW 10.00  CALC.  9.95

FOULING               0.0020 HR-F-SQFT/BTU                 0.0020 HR-F-SQFT/BTU

HEAT EXCHANGED        30635004.  BTU/HR              LMTD  86.6 F  CLMTD  83.7 F
TRANSFER RATE (IN SERVICE)                    57.3 BTU/SQFT-F-HR

                            MECHANICAL DESIGN
SHELL DESIGN PRESSURE      50.                 TUBE DESIGN PRESSURE      85.

URES=    956.     O.D.=0.750     I.D.=0.584     LENGTH=26.00     PITCH=0.9375

TRI. PITCH

SHELL I.D.    33.000

 9.BAFFLES    DOUBLE SEGM.  SPACING=        21.3  CUT=0.249

TUBES OUT IMP=  0

NOZZLES      SHELL     IN      20        OUT       6       ASA RATING
             CHANNEL   IN      12        OUT      12       ASA RATING
```

Fig. 2—

Air-cooled output — page 1

```
                        AIR COOLED HEAT EXCHANGER
                            ENGINEERING DATA
ITEM CONDENSOR   INQ. NO.SAMPLE AIR

NO. SECTIONS             4.
NO. BAYS                 2. EST WIDTH/BAY  15.60 LENGTH  40.00 TOTAL WIDTH  31.21

HEAT DUTY Q   14815002.0                    PQ HEAT LOAD OFF                 0.0
U. EXTERNAL-BARE     4.378     93.695       U CALC   4.420 PO EXCESS AREA    0.9
LMTD-CLMTD          23.751     23.278
EXTERNAL SURFACE 144737.28                  SURFACE/SECTION               36184.3
BARE TUBE SURFACE   6763.4

                             PROCESS SIDE
TOTAL FLOW   VAPOR   LIQUID   STEAM   HC CONDENSED
   113179.
TEMPERATURE IN-OUT  142.0       140.0
PRESSURE IN-OUT
VISCOSITY      AVG.     0.1300   WALL        0.1300
VAPOR MOL WT IN-OUT
SPECIFIC HEAT           0.445
FOULING RESISTANCE      0.00100

                        AIRSIDE AND CONSTRUCTION
TEMPERATURE IN      105.00   OUT         126.78   EVEV 3200.0   QUANT      2833920.
SECTIONS               4.    NO TUBES    162.0    ROWS   4.0    PASSES         1.

NOZZLES IN                6     OUT                 3    SERIES        PSI

TUBE O.D.            1.000   TUBE BWG      0.060  LENGTH                  40.0
HP/FAN              29.10    NO. FANS       4.0   FAN DIAMETER            12.0
ACFM/FAN         186569.6    STATIC PRESSURE DROP    0.368     IN H2O
```


Dale L. Gulley
Dale L. Gulley & Associates
Tulsa

Many computer programs have been developed that cover almost all phases of designing and engineering heat-transfer equipment. Only special cases remain to be programmed.

In trying to achieve a total computer-program package for heat-transfer equipment, we had a choice of two alternatives. One was to prepare one large thermal design and one large mechanical design program encompassing all phases of design. This would cut down on the number of computer programs but it would make a relatively complex computer input.

The other was to keep the different phases in logical blocks. This means a separate program for air-cooled, vertical thermosyphons, service with weighted mean-temperature differences (MTD), etc. Each component program has its own individual input.

For the benefit of the engineer we decided on the second plan. This means that there are many simple input forms, rather than a few complex ones. In this way there can be more explanation on each input form.

Thermal design

Between 1956 and 1961 there were many articles[1 2 3 4 5] on thermally designing heat exchangers by computer. At this time manufacturers were using relatively small and slow computers. There was quite a bit of operator intervention and decision.

Many variables were fed in on input that today are calculated by the computer. The type of heat transfer handled was mainly sensible heat and single-phase flow.

Now programs are quite a bit larger, the computers are much faster, and the whole operation is more automatic. Two-phase condensation and vaporization are programmed. Even the most complicated cases with weighted MTD's can be done in stages.

A recent development which has proven beneficial is an error-checking routine for each computer program. This is particularly advantageous whenever you are running on a large computer where computer time is quite expensive. It is built into the programs, and it catches obvious errors and omission of data.

If the input information is insufficient to run on the computer, the item is rejected and a message is printed out as to what the difficulty is.

However, the computer continues with a complete run. Then the item with the problem can be checked to see if the assumption of the computer is correct.

This can save several unnecessary runs. This feature is also helpful in training new people on filling in the input sheets and it catches obvious keypunch errors.

One portion of thermal design programs which can be quite a problem is the physical-property section. On some items of input the designer is given a choice on whether he wants to supply the physical properties or let the computer do it. Our procedure is used on input:

1. If input is left blank, computer does it.

2. If value is put in, computer will use it.

Take the case of vapor specific heat. Equations have been developed for hydrocarbon vapors and steam. For these fluids the input block for vapor specific heat is left blank. But if you have an exotic organic, hydrogen and hydrocarbon mixture, etc., then the specific value is filled in on input.

Taborek[3] describes generally how a data file can be set up for physical properties. It will automatically supply all the physical properties required for heat-transfer calculations in the form of equations which are functions of temperature pressure.

One physical property which is difficult to handle by the computer is viscosity. When the viscosity is plotted against temperature, the slope will vary with the type of fluid. Viscosity equations for common fluids such as water can be built into the program, but generally it is necessary to supply the viscosity at two different temperatures.

From this the computer can determine the slope and derive an equation.

This is quite helpful because you do not know the wall temperature to evaluate the wall viscosity when the design is started. It is only after the heat-transfer coefficients are developed that the tube-wall temperature can be determined with any accuracy.

Compressibility factors for gases can be fed in on input, or if their spaces are left blank then they can be computed by methods such as reference No. 6.

Shell and tube

Tube count. Generally there are three different methods of developing a tube count by computer. They are:

1. Table look-up
2. Area method
3. Accurate row method.

The table-look-up method is the easiest method to put on the computer. The tube-count tables are stored in a computer file. This has the disadvantage of taking up a lot of storage. It requires files of considerable size to have tables for a large variety of shell sizes and tube patterns.

In the area method, the actual area available for tubes is determined. If you have the full tube circle available for tubes, then it is a fairly simple matter to determine the area of the circle and mathematically calculate the number of tubes. But usually you have to deduct areas for nozzle entrance and exit space, then for multipass units you need to deduct the area lost due to pass plates.

The accurate-row method gives very accurate counts. It is more accurate than tube counts determined by the area method and more flexible than using table look-up procedures.

The exact length of each tube row is calculated. In addition the location of the horizontal pass plates are accurately determined. From this information the exact number of tubes in each row is calculated. If it is desired to knock out tubes for an impingement plate, the number of rows lost is computed by the program. Then the exact number of tubes lost is calculated.

Output. Fig. 1 is the computer for a shell-and-tube condenser. This is a unit with a long condensing range, operating near atmospheric pressure.

The output closely resembles the recommended form of (Tubular Exchange Manufacturers Association) (TEMA). The TEMA specification sheet can be found on page nine of the fifth edition. This item is oversurfaced to some extent because the tube-side pressure drop would be exceeded if we decreased the tube count.

HEAT-TRANSFER REPORT

With a normal shell-flow pattern this item would have excessive pressure drop. As can be seen from the output sheet, the computer used divided flow. This cuts the pressure drop down approximately one-eighth. In addition the computer used double segmental baffles which reduce the pressure drop even further. This is all done automatically by the computer.

There is also other computer output which is not shown here. The second page of output has summarized data such as:

1. Calculated and actual heat-transfer rates
2. Film resistances
3. MTD correction factor
4. Shell-side velocity and pressure drops
5. Physical properties used.

Other output shows summarized data on the different shell sizes tried by the computer. This is quite helpful in cases where the computer just barely missed meeting pressure drop and/or heat transfer. Sometimes it will just barely miss in one size down from the final output. Then, upon the judgment of the designer, he may decide to use the smaller size.

Air cooled

Once a computer program is developed for shell-and-tube exchangers, then there are certain characteristics that can be utilized in the air-cooled program. The same type of logic can be used which optimizes both the heat-transfer surface and tube length. In addition the tube-side equations are the same.

Some of the difficult areas of putting air-cooled rating on a computer are:

1. MTD correction factor.
2. Fan curves.

Charts of MTD correction factors can be found on page 85 of Natural Gas Process Supply Association, (NGPSA) 1966 manual[9], Perry[10], Kern[11], etc. We did not have the basic equations for these charts, so we curve fitted the correction factors.

Another area of difficulty is determining accurately the horsepower required. The fan curves, even for a

Fig. 3—

Flange-design output

```
ITEM   6756811

DESIGN PRESSURE=  1100.

BOLT DIAM.= 1.625           NO. OF BOLTS=  28.

BOLT CIRCLE= 35.500

GASKET G= 29.375            GASKET WIDTH= 0.500      GASKET O.D.= 29.875

HUB BASE= 1.312             GO= 0.937
HUB LENGTH= 1.625

FLANGE I.D.= 28.625         FLANGE O.D.= 38.750
FLANGE THICKNESS=  5.250

R=  2.125      E=  1.625


FLANGE ALLOWABLE STRESS
    DESIGN TEMP 17500.
    ATMOS. TEMP 17500.
BOLT ALLOWABLE STRESS
    DESIGN TEMP 20000.
    ATMOS. TEMP 20000.
GASKET DATA
    N=  0.500        BR=0.156
    Y= 18000.      GY=5.500

BOLTING
    WM2= 259550.      AM=     45.998
    HP=  174475.      AB=     47.040
    H=   745484.       W=    930379.
    WM1= 919960.      NMIN=    0.283


LOAD                  LEVER      MOMENT
 HD=    707903.          2.781     MD=  1968856.
 HP=    174475.          3.062     MG=   534330.
 HT=     37581.          3.250     MT=   122139.

 TOTAL OPERATING MOMENT=  2625326.
 GASKET SEATING MOMENT=,  7849288.

OPERATING STRESS              GASKET SEATING STRESS
 LONG HUB          19278.       20923.
 RADIAL FLG.        2631.        2856.
 TANG. FLG.        12901.       14002.
 GREATER AVG.      16090.       17462.


ITEM   6756811


                 SHAPE CONSTANTS

K            1.353          H/HO       0.313
T            1.774          F          0.876
Z            3.407          V          0.385
Y            6.568          FF         1.000
U            7.217          F/HO       0.169
GI/GO        1.400
HO           5.180          D         85.252

STRESS FORMULA FACTORS

ALPHA        1.888
BETA         7.184
GAMMA        1.064
SIGMA        1.697
LAMBDA       7.761
OPERATING M=       91714.
GASKET SEAT M=     99538.


A CHECK=     38.750  A=     38.750
```

given manufacturer, are quite varied and complex. Therefore our program is set up to give an approximate horsepower.

It is accurate enough to evaluate different alternates. Then when the final selection is decided upon the horsepower can be checked accurately by hand. The computer uses a constant fan efficiency of 65%.

A computer is particularly useful in designing air-cooled equipment. This is because there are so many possible solutions to a given problem. You can either have low horsepower and lots of surface or high horsepower and relatively small surface.

The number of tube rows also varies with the application. Sometimes it is necessary to look at many tube-row alternates. Fintube height and tube spacing are other variables that can be investigated on any given rating.

Another area where a computer run is particularly desirable is putting more than one service in a bay. Here it is necessary to run through each service to determine its requirements. Then adjustments have to go back through the design of all services several times before the final solution is achieved.

In Fig. 2 you will see a primary output of an air-cooled heat exchanger. This is also in specification-sheet form. The first part of the output shows the basic characteristics of the air cooler. The number of sections is defined as an individual unit of air-finned tubes. There may be a number of these over one fan drive.

For this selection there are two sections in each bay. Each bay will be approximately 16 ft wide by 40 ft long with two fans. Next is the summarized heat-transfer information. Both fin-tube and bare tube rates are given. Both total fin surface and bare-tube surface are given. The middle portion is process information.

The bottom section has to do with the air side and the mechanical construction. The supplementary output which is not shown here is similar in principle to the shell and tube.

The second page of the output gives the physical properties and other calculated variables. Just as in the shell and tube, summarized intermediate sizes are shown.

Mechanical design of shell and tube

The most important program here is one for the design of the body flanges. Each flange design is a complex trial-and-error solution to determine the optimum proportions of the flanges.

The flange dimensions are designed so that they will stay within the restrictions of American Society of Mechanical Engineers (ASME) Code and good fabricating practice. The most economical proportions will depend upon whether the manufacturer machines the flange from a ring or a forging.

Korelitz[12] has information on the programming logic and optimum gasket location. The determination of the optimum gasket location can be achieved by trial-and-error methods also.

Fig. 3 shows the output for the flange program. This output looks quite similar to the standard Taylor-Forge design form.

Other programs for mechanical design are:

1. Floating-head closure.
2. Channel-cover thickness.
3. Tube-sheet thickness.
4. Miscellaneous shell, channel, and ellipsoidal head thicknesses.
5. Reinforcing-pad design.

The procedure for calculating the tube-sheet thickness for fixed-tube-sheet exchangers has been expanded in the 1968 TEMA standards. Because of the new trial-and-error procedure, the calculation is much better suited for computers than it was previously. These effective design pressures take into consideration factors that were not used previously in TEMA.

Another computer program that can be in conjunction with the one for calculating tube-sheet thicknesses in fixed-tube-sheet exchangers is one for analyzing the stresses. The output from the tube-sheet program can be tied directly into one for analyzing the stresses.

In the future it is anticipated that plotters will be used to make the fabrication drawings. The mechanical information will be fed into the computer and the calculation of all thicknesses will be determined. From this information plotters will make the drawings.

References

1. Fabregas, J. A., "Electronic Computer Rates Heat Exchangers," Alco Products Review, 5, No. 4, Fall 1956.
2. Higgins, E. J., Kellett, J. W., Ung, L. T., "Engineering Design on a Computer," Ind. and Engr. Chem., Vol. 50, No. 5, May 1958, pp. 712-718.
3. Taborek, J. J., "Organizing Heat Exchanger Programs on Digital Computers," Chemical Engineering Progress, Vol. 55, No. 10, October 1959, pp. 45-48.
4. Gulley, D. L., "Use Computers to Select Exchangers," Petroleum Refiner, 39, 1960, pp. 149-156.
5. Whitley, Donald L., Ludwig, Ernest E., Petrolenm Refiner, January 1961, pp. 147-156.
6. Sommerfeld, Jude T., Perry, Gerald L., "Compressibility Factors by Computer," Hydrocarbon Processing, Vol. 47, No. 10, October 1968, pp. 109-110.
7. Souchaud, M. M., "Heat Exchanger Design by Computer," Petroleum Times, Aug. 21, 1964, pp. 428-430.
8. Standards of TEMA, Fifth Edition, 1968.
8. Natural Gas Processors Suppliers Assoc. Data Book, 1966.
10. Perry's Chemical Engineerings Handbook, Fourth Edition, McGraw-Hill, 1963.
11. Kern, D. Q., "Process Heat Transfer," McGraw-Hill, 1950.
12. Korelitz, T. H., "Cut Vessel Flange Costs by Computer," Hydrocarbon Processing Vol. 43, No. 7, July 1964, pp. 125-129.

HEAT-TRANSFER REPORT

Exchanger corrosion and fouling torment exchangers

ROBERT C. EWING
Gulf Coast Refining Editor

ON-STREAM service factors of refinery heat exchangers are going up as refiners continue to increase on-stream times of major units. This means a constant effort to design into exchangers those qualities which allow them to run for longer periods.

Designers maintain a vigorous search for new products and new concepts in the fields of metals, inhibitors, and antifoulants. Exotic metals provide a partial answer, but often must be aided by chemical neutralizers, inhibitors, and antifoulants.

In situ cleaning has in many cases facilitated the handling of the fouling problem, particularly on the tube side. But shell-side in-place operations continue to be a problem.

Oxygen problems. Oxygen, picked up from stocks in storage tanks open to the air, always causes either a corrosion or a fouling problem, especially where moisture is present. Corrosion of carbon steel under these conditions can be very serious.

Polymers and fouling polyps, which foul the equipment seriously, can form. And it's thought that sulfides polymerize olefinic trace material.

To overcome this problem, it might be ideal to run the stock from unit to unit with no intermediate storage. But if the refinery has developed and built separate units from time to time, or if incoming feed is not balanced with intermediate unit consumption, stocks must go to intermediate storage.

Then gas blanketing is the most obvious way to keep air from contracting the stock, and it is the most used tool. Where levels fluctuate in cone-roof tanks, modern standards dictate the use of breather valves.

Repressuring of a tank is a simple matter of bringing gas in under control. Very sensitive regulators detect the variations in pressure, from 0.2 or 0.3 oz maximum vacuum to 0.75 or 0.80 oz maximum.

Salt problem. A crude that comes well desalted or which has very little salt content, say 1 to 10,000 ppm of salt per 1,000 bbl, will create a lot fewer corrosion and fouling problems.

Salt is either decomposed or hydrolized during distillation. Magnesium chloride hydrolizes to give free hydrochloric acid gas, which distills over. If HCl is dry it causes no particular problem, but if it contacts moisture it becomes a very corrosive material.

If HCl reaches heat exchangers, corrosion-mitigation measures such as alloys and linings, inhibitors, and/or injection of ammonia must be used. Ammonia combines with the HCl to form ammonium chloride which is a far less corrosive material.

Inhibitors will usually be one of the filming types, usually amine or amide nitrogen-based materials, which form films on the metal and tend to protect corrosion.

Some plants use only the ammonia, controlled to give a certain pH value, usually ranging from 7 to 8. A higher pH takes more ammonia, and if admiralty tubes are being used, they are attacked by free ammonia.

If the crude contains a lot of salt and is also very sour, where the sulfur will protect the admiralty from the ammonia, a slightly higher value might be used. But it is usually not desirable to go above 8 or 8.5 in any case.

Acids. Mercaptans and organic acids also cause exchanger corrosion. These acids may be formed because of air or CO_2 in the crude, or some may be in the oil to begin with, but acids will show up in varying amounts depending upon the crude stocks and the actual operation.

Phenol-extraction units experience a mild form of corrosion which seems to tie in with air pickup. Phenol is rather easily oxidized to produce organic acids. So 18-8 stainless steels