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Sulfur Hexafluoride Transfer and Storage System of the Holifield Heavy Ion Research Facility: Some Thermodynamic Properties

W. T. Milner

OAK RIDGE NATIONAL LABORATORY

OPERATED BY UNION CARBIDE CORPORATION · FOR THE DEPARTMENT OF ENERGY

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Physics Division

SULFUR HEXAFLUORIDE
TRANSFER AND STORAGE SYSTEM
of the
HOLIFIELD HEAVY ION RESEARCH FACILITY:
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ABSTRACT

The transfer of SF_6 insulating gas from the 25 MV accelerator pressure vessel (volume = 80,000 ft^3) into liquid storage tanks (volume = 6,000 ft^3) is accomplished by means of two three-stage piston compressors operating in parallel. Gas from the first and second compression stages is passed through heat exchangers prior to injection into subsequent stages while gas from the third stage is cooled and/or liquified by a third exchanger/condenser. We have made use of tabulated thermodynamic data in calculating the SF_6 transfer rate, stage compression ratios and heat transfer rates as a function of the fraction of the total inventory of SF_6 (270,000 lbs) which has been transferred. The SF_6 temperature, pressure and liquid fraction are also calculated at various points throughout the system. Operating parameters and procedures are recommended which should eliminate the possibility that liquid SF_6 is injected into a compressor cylinder and prevent the generation of excessive pressure in the storage tanks.

ACKNOWLEDGEMENTS

I wish to give special thanks to the following people who have contributed to this study or the preparation of the report

Person	Primary Contribution
Joyce Y. Foster	Typing of manuscript under adverse conditions and being sweet about it
Sherry W. Hawthorne	Preliminary editing of draft
C. M. Jones	Suggestion of study, discussions, some calculations and review of manuscript
M. V. Keigan	Discussions and some calculations
J. E. Vasgaard	General review of problem, discussions and some calculations

1. INTRODUCTION

This report is concerned with the transfer of SF_6 insulating gas from the 25-MV tandem accelerator pressure vessel to the liquid storage tanks by two Cooper-Penjax compressors. A simplified diagram of that part of the gas-transfer system with which we are concerned is shown in Fig. 1. Figure 2 gives a more detailed diagram of one of the compressors. Compression is done in three stages, with SF_6 gas from the first and second compression stages being passed through heat exchangers before injection into subsequent stages. Gas from the third compression stage is cooled and/or liquified by the third stage exchanger/condenser prior to its transfer into the storage tanks. Heat is removed from each exchanger by a circulating bath of cooling water.

1.1 Motivation for the Study

During the review of the system design submitted by National Electrostatics Corporation (NEC), the question was posed as to whether liquid SF_6 could be taken into the second or third compression stages under certain conditions. The intake of a significant amount of liquid into a compression cylinder could damage or destroy the compressor. Some hand calculations were done by Nambar (of NEC) which indicated that for a cooling-water temperature of 85°F (the temperature specified by ORNL Specification XSP-439 and ORNL-TM-4942), interstage condensate would not develop. ORNL reviewed Nambar's calculations and agreed with his conclusion. We became concerned, however, that the use of 85°F cooling water at the flow rates recommended might lead to an excessive storage-tank pressure. In addition, we have no facilities to automatically or precisely control the cooling-water temperature. In fact, 85°F should be considered as an upper limit of the cooling-water temperature, which even during warm summer days has been found to be as low as 65°F. We decided that we needed to develop the capability to calculate realistically the performance of the system for a wide range of operating parameters in order to more accurately predict the behavior of the gas-transfer system.

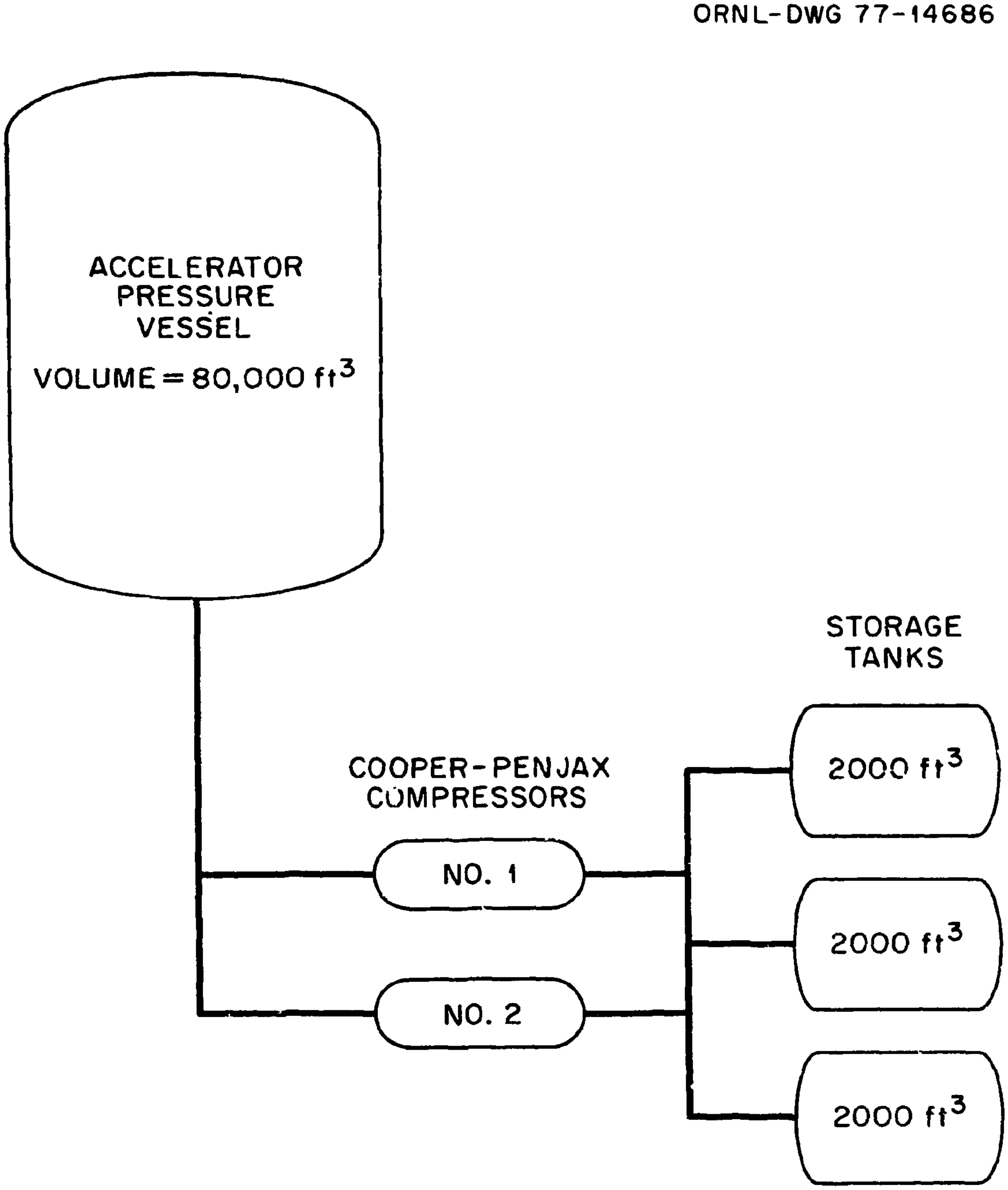

Fig. 1 Simplified diagram of some major components of the SF_6 transfer and containment system

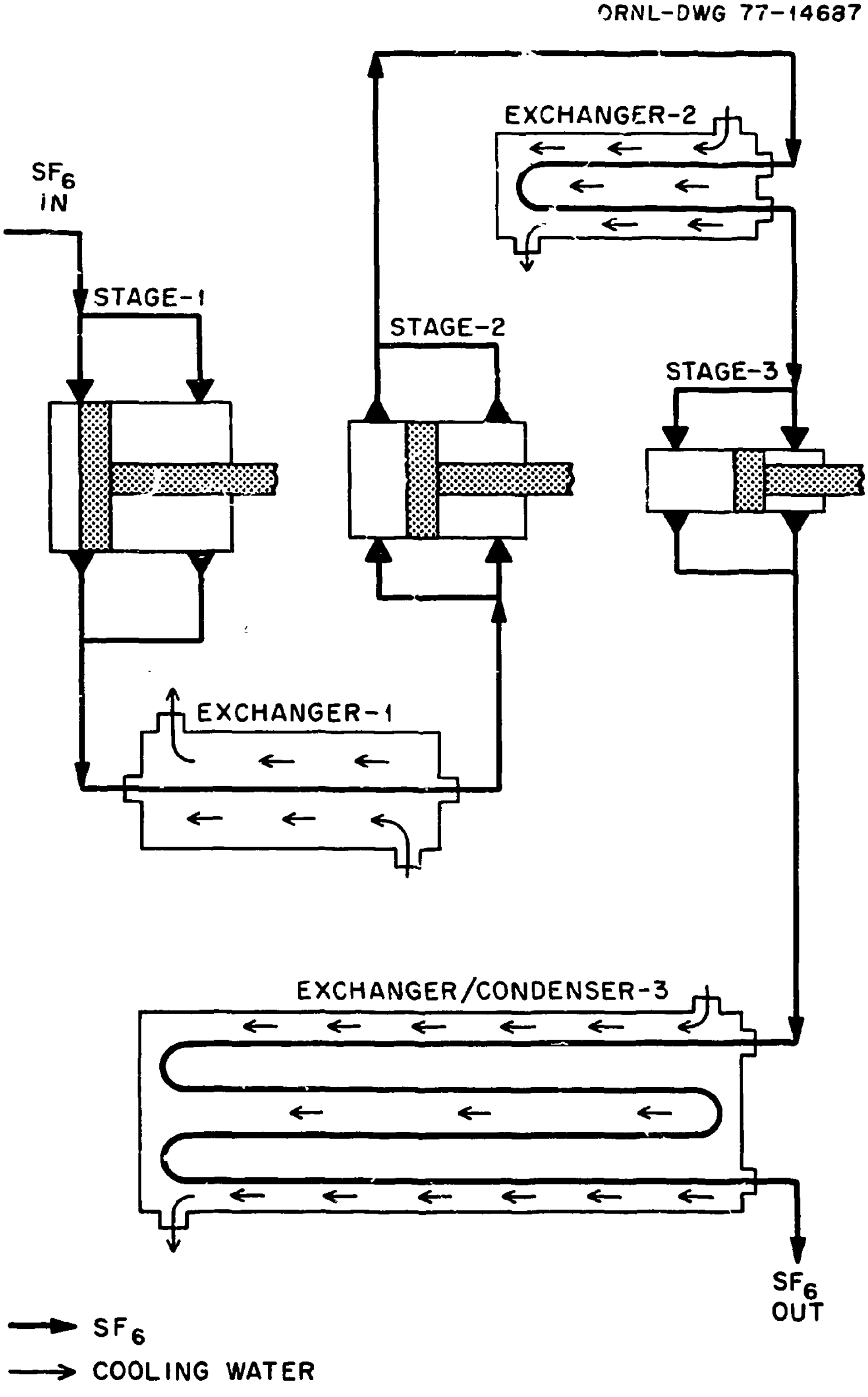

Fig. 2 Diagram of one Cooper-Penjax compressor showing the three compression stages and the heat exchangers for cooling the SF_6

1.2 Objectives of the Study

The most important objectives of this study were:

(1) To determine what conditions can lead to interstage condensation of the SF_6.

(2) To determine the temperature and pressure of the SF_6 in storage during and following a complete gas-transfer operation, under any chosen set of operating parameters.

(3) To specify operating procedures and/or hardware modifications required to avoid interstage condensation and storage-tank overpressure.

1.3 Approach to the Problem

We have developed the capability to calculate the temperature, pressure, and liquid fraction at any place in the system at any time during the gas transfer for any reasonable set of operating parameters. Tabulated data[1,2] are used for the SF_6 thermodynamic quantities enthalpy, entropy, specific heat, density, etc. Compression ratios and SF_6 flow rates are calculated from "first principles" and compared to a few values we have obtained from Cooper-Penjax[3] (see Section 7). The heat transfer in exchangers 1 and 2[4] is calculated using standard engineering formulas[5] (see Section 5), and the third-stage exchanger/condenser is treated numerically (see Section 6).

1.4 Assumptions

Our approach assumes that the compression is isentropic[6]; the storage tanks are always at a uniform temperature; all storage tanks involved in the transfer contain the same mass and thermal energy; and no heat is transferred into the system from the outside environment with the exception that approximately 200,000 Btu/h is transferred to the SF_6 in the accelerator vessel (see Section 3.1).

1.5 Definitions

TSUP - degrees of superheat, i.e., the temperature of the vapor under consideration minus the temperature of saturated vapor at the same pressure.

FMX - fraction of mass transferred, i.e., that fraction of the total SF_6 mass which is contained in the storage tanks.
GPM - gallons per minute
CW - cooling water
CWT - cooling-water temperature
CWF - cooling-water flow
HEX - heat exchanger
PSTOR - storage-tank pressure
TSTOR - storage-tank temperature
U - heat-transfer coefficient
TSI - temperature of SF_6 at compressor intake
TSV - temperature of SF_6 in the accelerator vessel
SMF - SF_6 mass flow rate (lbs/h)
UNORM - a normalization factor by which all heat-transfer coefficients in the system have been multiplied.

The suffixes 1, 2, and 3 are used in conjunction with the abbreviations TSUP, CWF, and HEX to denote reference to Stages 1, 2, and 3. For example, HEX2 denotes heat exchanger #2, CWF1 denotes the cooling-water flow through HEX1, etc.

1.6 Nominal System Parameters and Data

Accelerator pressure vessel volume = 80,000 ft.^{3}
Storage tank volume = 6000 ft.^{3} (three 2000 ft.^{3} tanks)
SF_6 inventory = 270,000 lbs.
TSI ≈ 85°F initially*

1.6.1 Compressor Data[3]

Stage	Piston Diameter	Piston Displacement	Min. Cylinder Vol. (% of Displacement)
1	10.5"	445 ft.^{3}/min	8.51
2	6.75"	178 ft.^{3}/min	13.51
3	4.5"	83 ft.^{3}/min	13.11

The compressor is double acting, has a 9.0" stroke, a 2.25" rod diameter, and makes 505 strokes per minute.

*The SF_6 temperature at the input to the compressor is not very well known. The accelerator vessel temperature will initially be at 75 to 85°F, but as gas is removed the contents will cool due to the expansion of the gas (see Section 3.1).

1.6.2 Heat-Exchanger Data[4]

	Stage 1	Stage 2	Stage 3
Heat-transfer area (ft.^{2})	161.3	111.0	479
Heat-transfer coefficient $\left(\frac{\text{Btu}}{\text{h. ft. °F}}\right)$	113.5	115.2	117.0*
Nominal CW flow (GPM)	56	29	105**
CW pressure drop at nominal flow (psi)	1.9	1.14	<5

* The heat-transfer coefficients given here are values calculated by the manufacturers (Bos-Hatten)[4] for a fixed SF_6 flow rate of approximately 50,000 lbs/h. The actual values will depend on a number of parameters, one of the most important being the SF_6 flow rate (see Section 8).

** These CW flow rates are those suggested by Bos-Hatten; however, we are not strictly bound by these rates.

1.7 Heat-Exchanger Cooling-Water Flow Configuration

Figure 3a illustrates what we refer to as the PARALLEL cooling-water flow configuration. In this case all exchangers are supplied with CW from a common source, and the flow through each exchanger can be adjusted independently. This is the usual arrangement.

Figure 3b illustrates an arrangement in which the CW discharged from HEX1 is used as input to HEX2. The advantage of this SERIES configuration will become evident in subsequent discussions.

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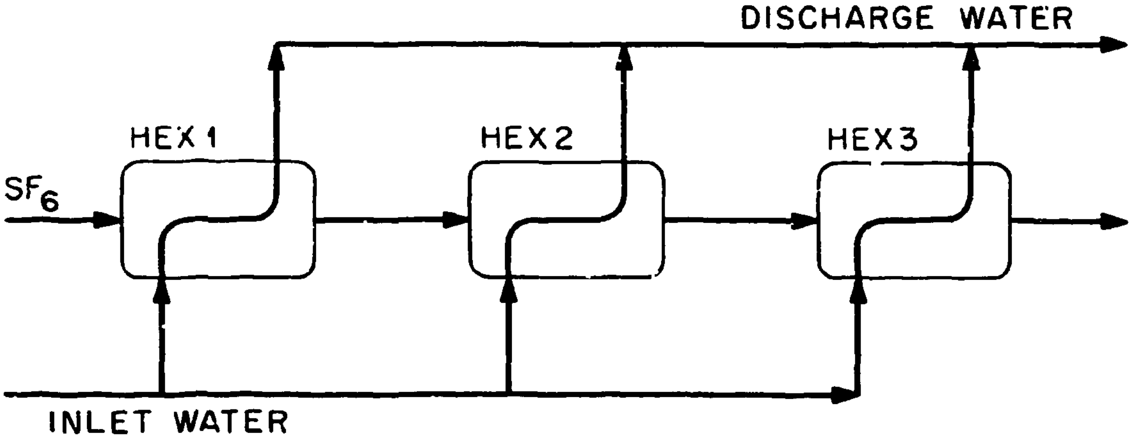
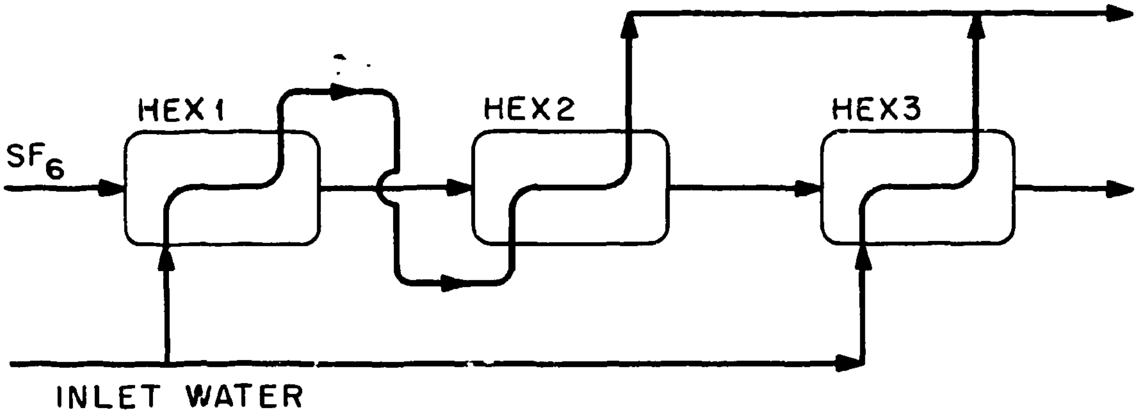


PARALLEL COOLING WATER FLOW

(*a*)



SERIES COOLING WATER FLOW

(*b*)

Fig. 3 Diagram showing the cooling water flow geometry for the PARALLEL and SERIES configurations

2.0 DISCUSSION OF THE PRINCIPAL RESULTS

This section deals primarily with the final results of the study. Section 3 will present a discussion of the intermediate results or system dynamics, and Sections 4 through 8 are concerned with computational details and procedures.

Most of the results discussed in this report involve a full transfer calculation as described in Section 4. Briefly stated, the calculation starts with a given inventory of SF_6 at equilibrium in the combined volumes of the accelerator pressure vessel and the storage tanks. The compressors are started, and SF_6 is transferred into the storage tanks until the accelerator vessel pressure decreases to 15 psia. At this point the intake pressure of the compressor is maintained at 15 psia (by high-speed vacuum pumps) until essentially all of the SF_6 has been pumped into storage.

2.1 Specification of the Cases Discussed

Although many different sets of operating parameters have been investigated in the course of this study, we will generally restrict the scope of this report to a discussion of the effect of the four variables listed below.

(1) Cooling-water flow configuration

PAR - PARALLEL (as defined in Fig. 3a)

SER - SERIES (as defined in Fig. 3b)

(2) CWT - cooling-water temperature

(3) CWF3 - cooling-water flow through HEX3

(4) UNORM - heat-transfer coefficient normalization factor (applied equally to all coefficients in the system)

2.1.1 Conditions Which are the Same for all Cases

In addition to the system parameters given in Section 1.6, we have assumed that the temperature of the SF_6 in the accelerator vessel decreases linearly with FMX from an initial value of 85°F to a final value of 30°F. For practical purposes, this variation approximates the situation in which 200,000 Btu/h is supplied to the SF_6 in the accelerator vessel during the transfer into storage (see Section 3.1).

We have used the heat-transfer coefficients (modified by UNORM in some instances) described in Section 8.

2.1.2 Parameters Implied by the Configuration

In all cases where the configuration is PAR, we also imply

CWF1 = 56 GPM,
CWF2 = 29 GPM.

In all cases where the configuration is SER, we also imply

CWF1 = 40 GPM,
CWF2 = 40 GPM.

2.1.3 Specification Format

The following abbreviated format is used in order to conveniently specify the four system parameters defined in section 2.1: CONFIGURATION (CWT, CWF3,UNORM), where CONFIGURATION is either PAR or SER, CWT is given in °F, CWF3 is in GPM, and UNORM is a fraction less than or equal to unity. For example, PAR(75,105,0.8) specifies PAR configuration with:

CWT = 75°F
CWF1 = 40 GPM (implicit)
CWF2 = 29 GPM (implicit)
CWF3 = 105 GPM
UNORM = 0.8.

A specification of the form SER(X, 150, 1.0) indicates that in the results being discussed or labeled, CWT is a variable. The words PARALLEL and SERIES are used, without arguments, in general references to these configurations.

2.1.4 The INFINITE Case

This case, so named because it corresponds to a situation where all heat exchangers have infinite capacity, is not intended to be an approximation to reality and has been included for comparison only (i.e. the only variable remaining is CWT). Its parameters are:

CWF1 = CWF2 = CWF3 = U1 = U2 = U3 = ∞
CWT1 = 120°F (unimportant in this case)
CWT2 = 120°F (unimportant in this case)
CWT3 = 35 -95°F as specified

2.2 Interstage Superheat

Figure 4 displays the minimum value of TSUP1 and TSUP2 (the degrees of superheat) which occurred in the transfer, as a function of CWT, for the PAR (X, 105, 1.0) and SER (X, 150, 1.0) cases. From these results, it appears that the use of the SERIES configuration will decrease by about 20°F, the CWT at which it is safe to operate. Since the minimum value of TSUP2 depends strongly upon how the compression is distributed between the three stages (see Section 3.4, Fig. 15), it would seem unwise to plan to operate with values of TSUP2 below about 10°F. If this criterion is accepted, operation with the PARALLEL and SERIES configurations should be restricted to CWT above 40 and 60°F, respectively. See Section 3.4 for more detailed results and discussion.

Since the pressure in HEX1 is generally less than the pressure in HEX2, TSUP1 is always greater than TSUP2 in normal operation.

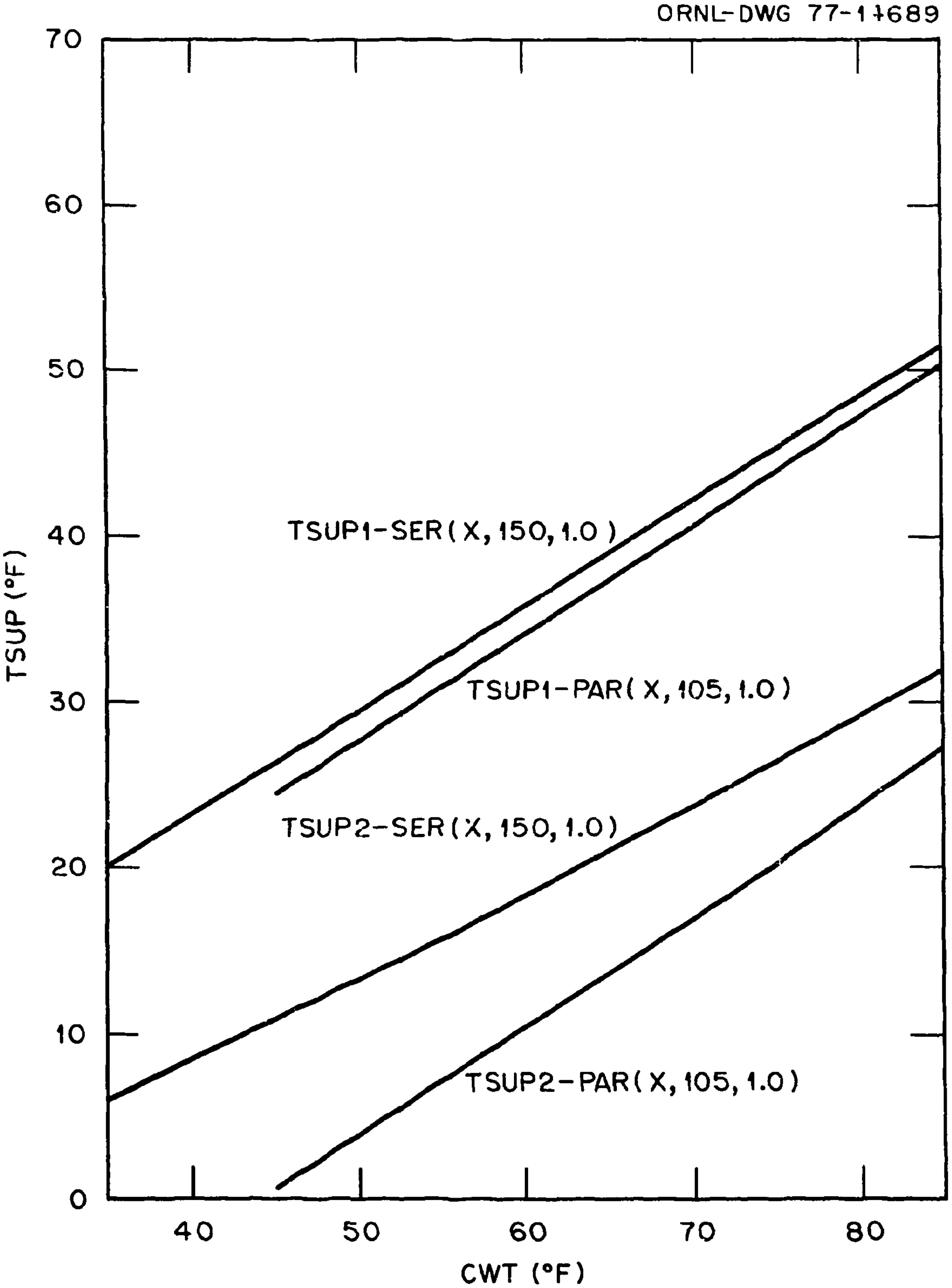


Fig. 4 Graph showing the minimum values of SF_6 "superheat" that developed in exchangers 1 and 2 as a function of the cooling water temperature CWT

2.3 Storage-Tank Temperature as a Function of CWT

The final temperature in the storage tanks, TSTOR, is presented as a function of CWT in Fig. 5. There is only a modest difference between the PAR (X, 105, 1.0) and SER (X, 150, 1.0) cases. The SER (X, 150, 1.0) case is slightly better than the PAR (X, 105, 1.0) case because more cooling water is applied to HEX3, where it can be utilized more effectively.

2.4 Storage-Tank Pressure as a Function of CWT

The final storage-tank pressure, PSTOR, is presented as a function of CWT in Fig. 6. Since the pressure is a function of the temperature, the same arguments apply here as in the preceding section.

Observe that the performance of an "INFINITE" exchanger operating with CWT = 85°F can be achieved with "realistic" exchangers (PARALLEL or SERIES) operating with CWT = 68 - 72°F. Results for the PARALLEL case are limited to CWT = 45 - 80°F because interstage condensation occurs below 45°F and a computational difficulty arises above 80°F which makes the results less reliable. Basically, the problem is this: the program has not been provided with sufficiently detailed thermodynamic data in the region above the critical temperature and pressure to allow it to interpolate in the pressure - temperature space with the desired degree of accuracy.

The SF_6 storage tanks are designed to operate at a maximum pressure of 600 psig; therefore it is desirable to keep PSTOR (the partial pressure of SF_6 as we have defined it) significantly below this value in order to prevent SF_6 loss via safety-valve operation. Also, a certain amount of air is expected to build up in the SF_6 inventory as a result of imperfect accelerator vessel evacuation subsequent to an entry. The system is specified to operate with up to 10,000 standard cubic feet of air in addition to the 270,000 lb. inventory of SF_6. This quantity of air could lead to an additional partial pressure of 50 psia. Therefore, the partial

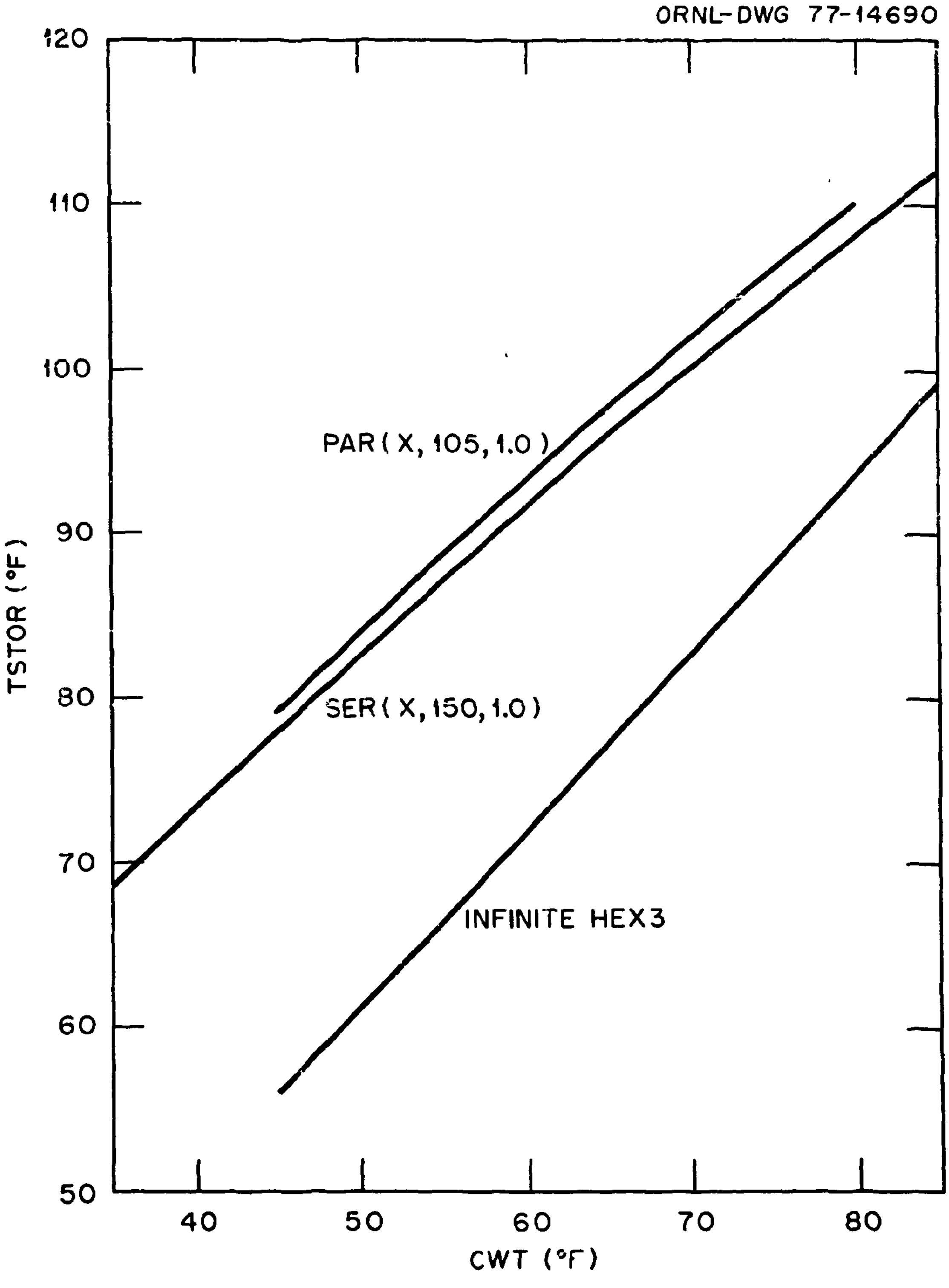


Fig. 5 Temperature of SF_6 in storage, TSTOR (following a complete transfer), as a function of CWT

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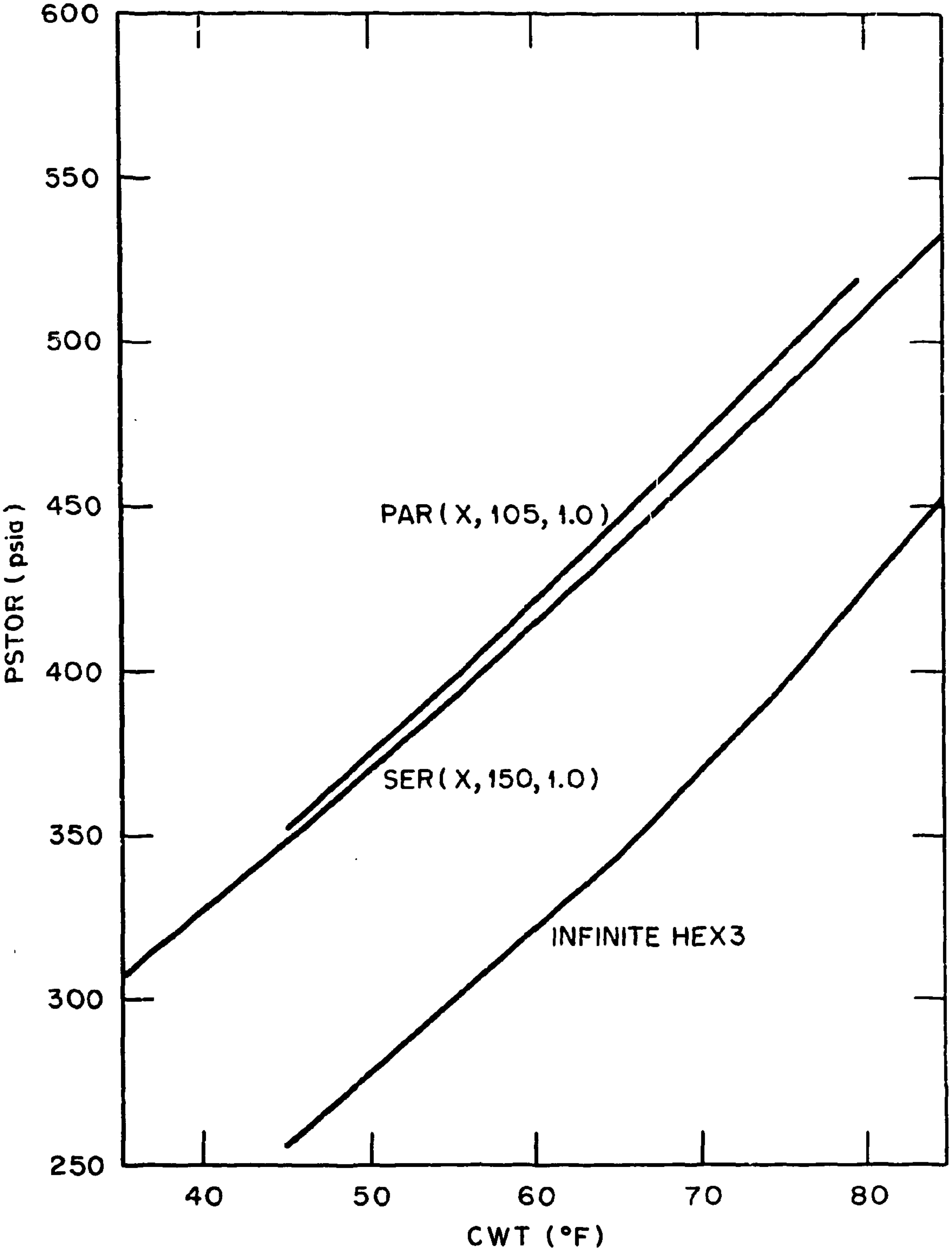

Fig. 6 Pressure of SF_6 in storage, PSTOR (following a complete transfer), as a function of CWT

pressure of SF_6 should be kept below 525 psia if at all practical. This implies that a CWT above 75°F should be used with caution and only if absolutely necessary.

2.5 Storage-Tank Pressure as a Function of U3

The final value of PSTOR depends strongly on the parameters that define the performance of HEX3 since this exchanger is subjected to the large heat load associated with liquefaction of the SF_6. Two questions arise with regard to this exchanger/condenser: How accurately is the effective transfer coefficient, U3, of HEX3 actually known, and how does PSTOR vary with U3?

Since it does not seem to be possible to calculate the heat-transfer coefficients (especially U3) with a great degree of confidence (see Section 8), it seems worthwhile to determine to what extent the system performance is limited by the transfer coefficients (primarily U3). In other words (since the heat transfer $\propto$ U • Area), is the area of HEX3 sized conservatively?

In Fig. 7 the final storage-tank pressure, PSTOR, is displayed as a function of the heat-transfer coefficient normalization factor, UNORM, for the SERIES configuration. These results indicate that the system should perform satisfactorily with CWT as large as 75°F and with UNORM as small as 0.6 (i.e., up to 40° degradation in the heat-transfer coefficients of all three stages).

2.6 Maximum Stage Pressure as a Function of CWT

The SF_6 transfer and containment system is provided, throughout, with a number of pressure relief (safety) valves. The maximum stage pressures that develop during the SF_6 transfer have been examined for all probable operating conditions in order to determine the pressure rating and setting of relief valves associated with each of the three compression stages.

Figure 8 displays the maximum pressure developed in Stages 1, 2 and 3 for CWT = 35 - 80°F. The configuration is SER (X, 150, 1.0).

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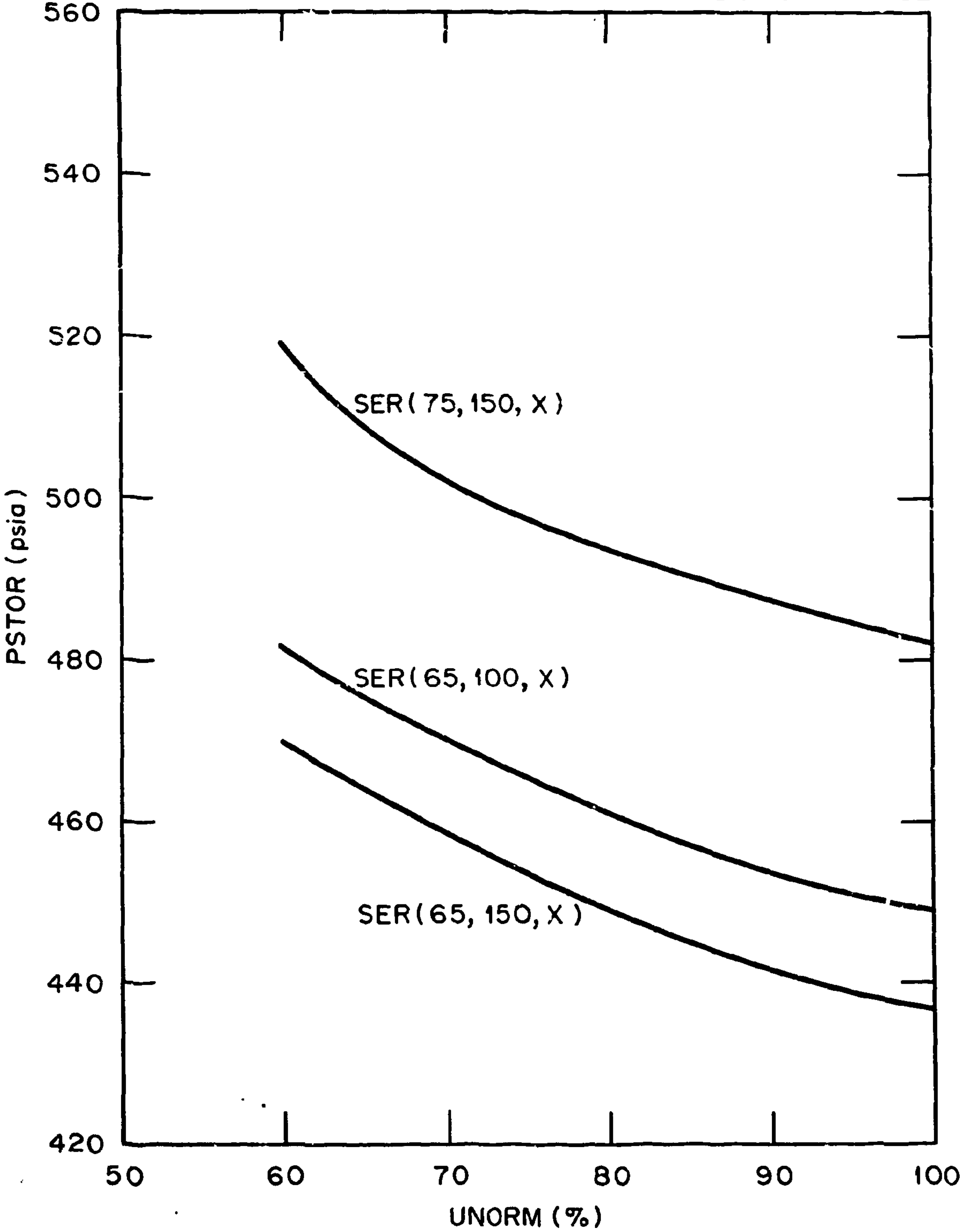

Fig. 7 Pressure of SF_6 in storage, PSTOR (following a complete transfer), as a function of the heat-transfer coefficient normalization factor, UNORM

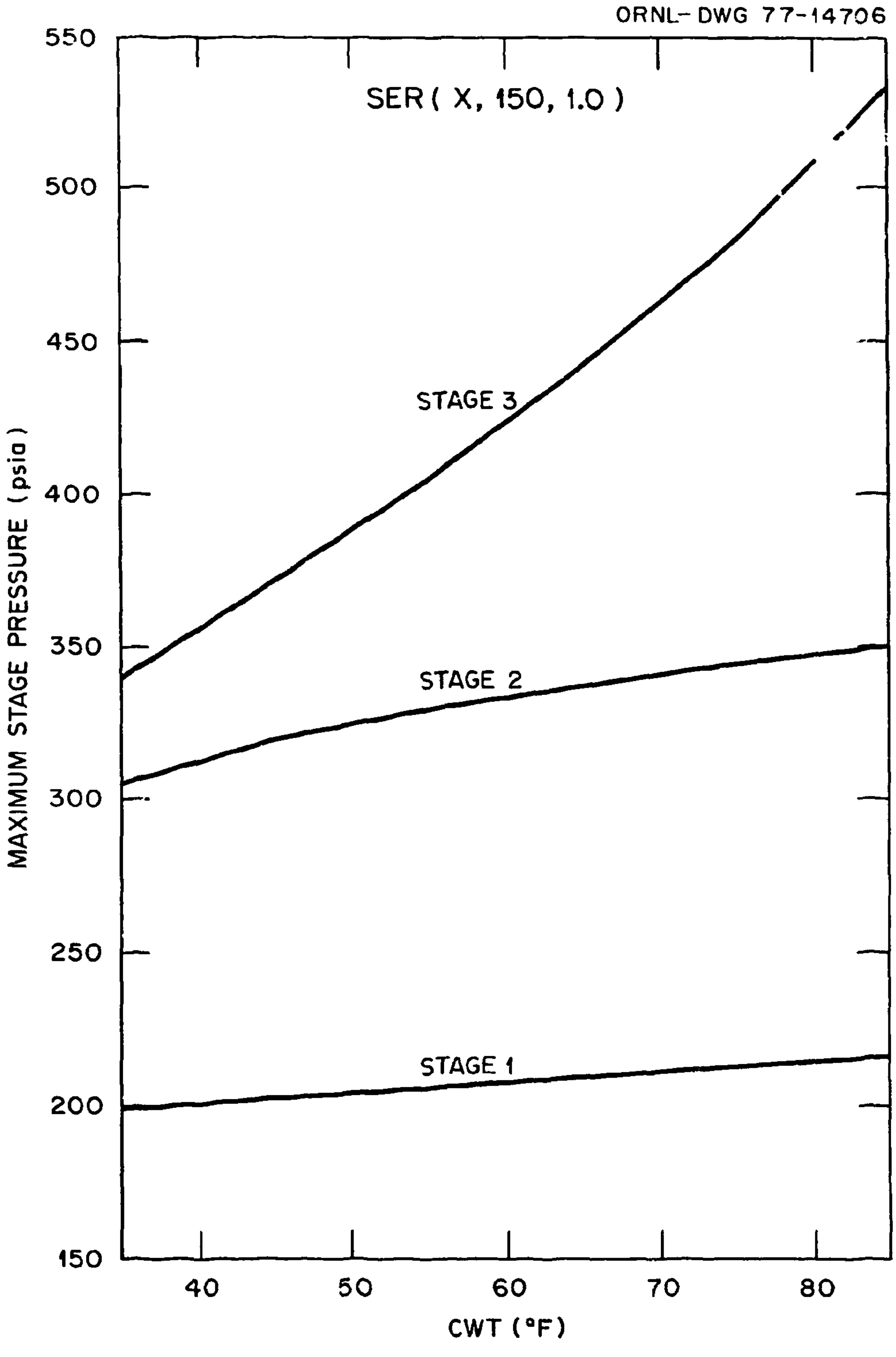

Fig. 8 Maximum pressure that developed during a complete SF_6 transfer (in compression Stages 1, 2, and 3) as a function of CWT.

3.0 DISCUSSION OF SYSTEM DYNAMICS

The gas-transfer system does not operate in a steady-state environment since the accelerator vessel pressure continuously decreases while the storage-tank pressure generally increases, but in a non-linear fashion. In this section we will see how the SF_6 temperature, pressure, and liquid fraction (at various points in the system), as well as the SF_6 flow rate, change as the transfer proceeds. Finally, we will see how this is related to the interstage condensation problem and to the final storage-tank pressure and temperature.

Many of the results presented in this section are for the SERIES configuration with CWT = 75°F. This is considered to be the most desirable configuration and the "maximum likely" cooling-water temperature. The system will behave the same qualitatively whether operated in the PARALLEL or SERIES configuration with any value of CWT between 45 and 85°F.

3.1 SF_6 Temperature in the Accelerator Vessel

As SF_6 gas is transferred from the accelerator vessel into storage, the SF_6 remaining in the vessel will expand and become cooler. The curve in Fig. 9 labeled "isentropic expansion" shows how the temperature of the SF_6 in the vessel would vary with FMX if no heat were transferred into the gas. In actual practice, there are several sources of heat which will increase this temperature (TSV). Although the value of TSV will have some impact on the performance of the gas-transfer system, this turns out to be of secondary importance. A more important concern is whether components located within the accelerator vessel will get so cold that condensate will be formed when the vessel is filled with humid air. Of course, it is not expected that the temperature of the solid components within the vessel will follow the gas temperature completely; however, it should be noted that components located along the column are more or less thermally isolated from most heat sources, with the possible exception of one aluminum bulkhead.

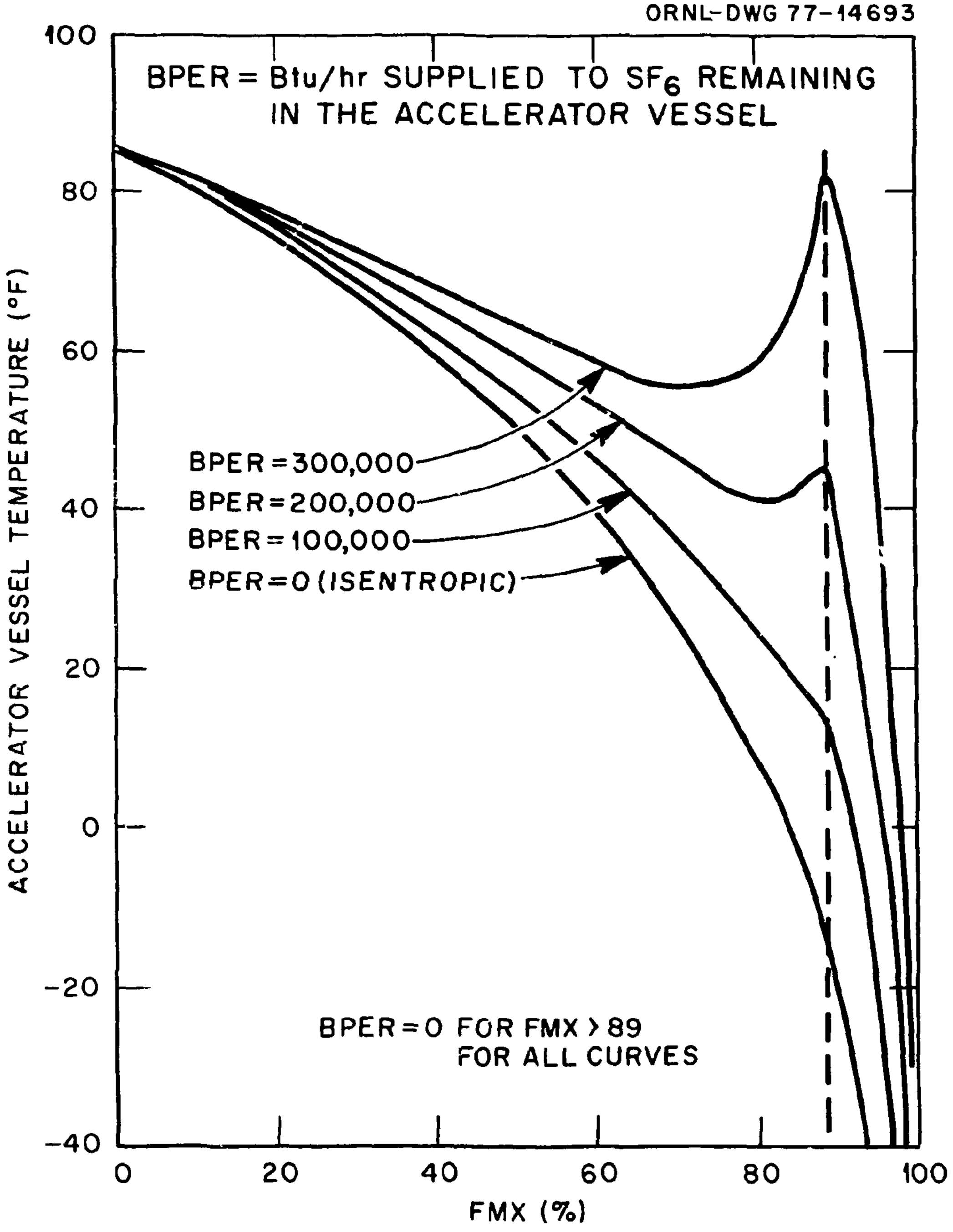

Fig. 9 Temperature of SF_6 in the accelerator vessel as a function of the fraction of the total inventory removed, FMX. Curves are labeled according to the external heat being supplied (at constant rate).

The total amount of heat required to maintain the accelerator vessel contents at an approximately constant temperature (say, no lower than 75°F) is about 2 million Btu's. The accelerator vessel and its solid contents contain about 1 million lbs. of steel, aluminum, etc. If we use a specific heat of 0.1 Btu/lb. °F (the approximate value for steel), we find that heat capacity of all solids associated with the vessel is about 10^5 Btu/°F. Thus, a 10°F reduction in the vessel temperature would yield 1 million Btu's. The actual amount of heat that can be transferred from the vessel will depend upon the rate of gas circulation that can be maintained by thermal convections as well as by the external recirculating system.

It should also be possible to heat the SF_6 by operating the recirculating system (normally used for cooling and drying the SF_6) with warm water supplied to the external heat exchanger. This exchanger can normally supply about 3×10^5 Btu/h when operated with water that is $\approx$ 40°F cooler than the SF_6. The SF_6 flow through the external heat exchanger and dryer is normally maintained at 350 ICFM by one of two (one spare) turbo-compressors. It should be possible to run both compressors in parallel during gas-transfer operations to achieve a larger SF_6 flow rate as well as a somewhat higher heat-transfer rate in the external exchanger (see Section 8). If an SF_6 flow rate of $\sim$ 600 ICFM could be maintained by this means, the vessel contents could be recirculated at a rate of once in two hours. However, it may not be possible to operate the recirculation system at vessel pressures below around 15 psia.

The curves in Fig. 9 labeled 1×10^5, 2×10^5 and 3×10^5 Btu/h present TSV vs. FMX to be expected when heat is supplied to the gas at the constant rates indicated. The heat source has been cut off at approximately 15 psia.

When the vessel pressure decreases to $\leq$ 15 psia (at FMX $\approx$ 88%) high-speed vacuum pumps (Kinney pumps) will start and maintain the intake pressure of the Cooper-Penjax compressors at about 15 psia. The SF_6

temperature at the Cooper-Penjax intake should then remain fairly constant for the duration of the transfer.

Finally, there are about 290 feet of steel pipe (diam. = 8" to 12") between the accelerator vessel and the Cooper-Penjax compressors. The total surface area of the pipe is $\sim$ 1000 ft^2. Although the flow through this pipe will be more or less laminar, resulting in a very small heat-transfer coefficient, some additional heat will be transferred by this pipe.

3.2 SF_6 Mass Flow Rate and Stage Compression Ratios

The SF_6 mass flow rate provided by a single 3-stage compressor is displayed as a function of FMX in Fig. 10a. The nearly linear decrease in flow rate is due simply to the nearly linear decrease in the accelerator vessel pressure (SF_6 in the accelerator vessel is almost an ideal gas). There is some effect due to the change in the volumetric efficiency of Stage-1 (see Section 7) with compression ratio, which gives rise to the non-linear region at FMX = 8 - 20%. Here the Stage-1 compression ratio increases rapidly from 1.0 to 2.5 and the Stage-1 volumetric efficiency decreases accordingly. For FMX > 20%, the Stage-1 compression ratio changes much more slowly. The constant flow (8000 lb./h) in the region FMX = 90-100% is due to the maintenance of a constant fore-pressure ($\sim$ 15 psia) by high-speed vacuum pumps.

Figure 10b shows how the stage compression ratios vary as the transfer proceeds (see Section 7.0 for computational details). Initially, Stage 1 provides all of the compression with Stages 2 and 3 simply allowing gas to flow through. At FMX $\approx$ 16% and 27%, Stage 2 and Stage 3, respectively, start to contribute. In the final portion of the transfer, Stages 1 and 2 contribute approximately equal amounts; Stage 3 contributes a much larger fraction of the total compression.

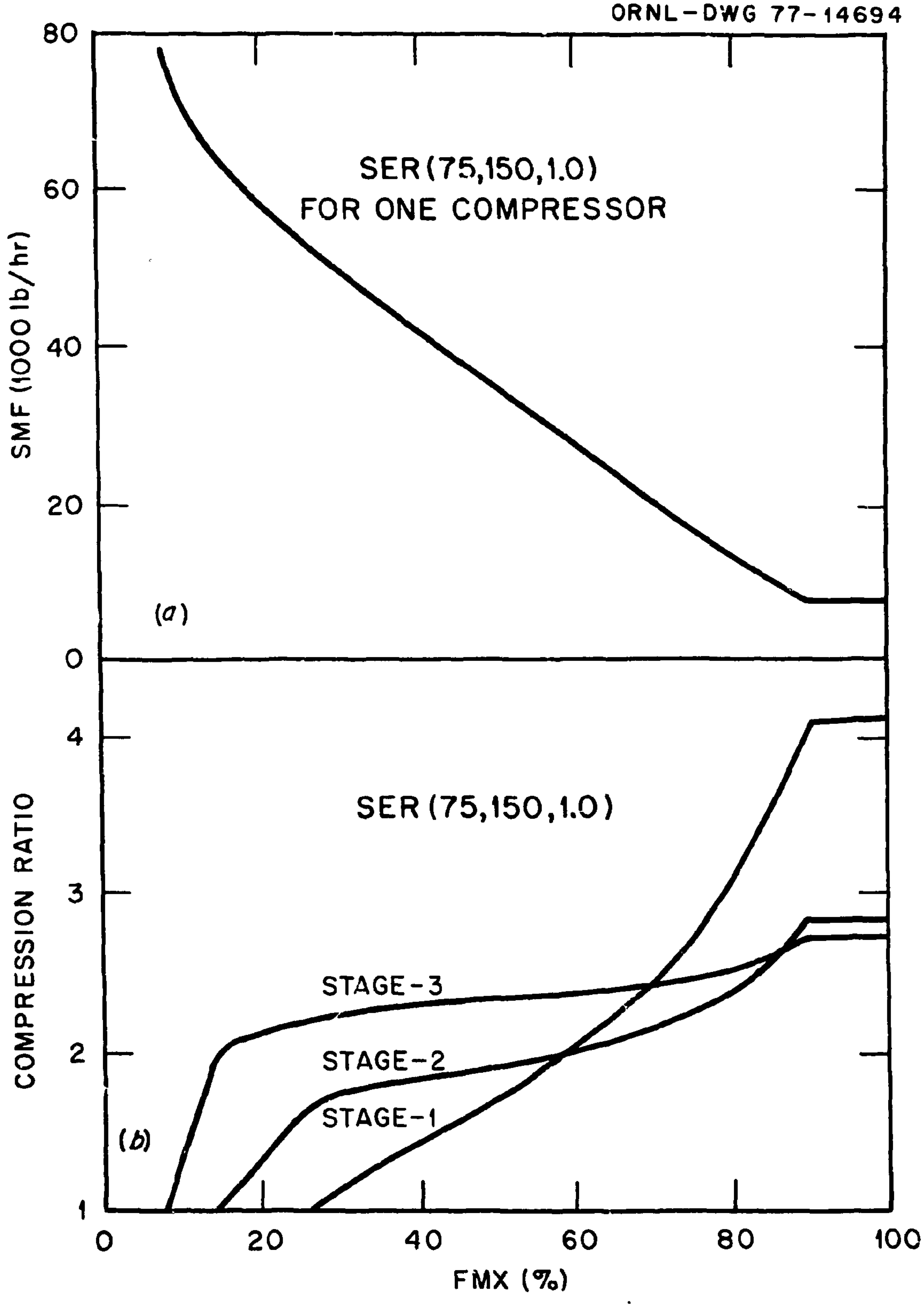

Fig. 10 SF_6 mass flow rate (for one compressor) and stage-compression ratios as functions of FMX.

3.3 Compression-Stage Discharge Temperatures

The discharge temperatures from compression stages 1 through 3 are displayed as a function of FMX for PAR (75, 105, 1.0) and SER (75, 150, 1.0) in Figs. 11 and 12, respectively. Both the Stage-1 and Stage-2 temperatures rise rapidly to about 120°F. The Stage-2 temperature then remains fairly constant throughout the transfer while the Stage-1 temperature decreases due to the decreasing input temperature (TSI). The Stage-3 discharge temperature, however, increases in a roughly linear fashion, due to an increasing compression ratio, to a maximum value of approximately 170°F at FMX = 90 - 100%.

3.4 Interstage Superheat as a Function of FMX

Figures 13 and 14 show how the amount of SF_6 superheat (TSUP2) at the discharge of HEX2 changes throughout the gas-transfer operation for the PAR (X, 105, 1.0) and SER (X, 150, 1.0) cases, respectively. Curves corresponding to CWT = 35 - 85°F are shown where appropriate. The SERIES curves are somewhat flatter and lie 5 to 10°F higher than the corresponding PARALLEL curves (see also Fig. 4).

Figure 15 presents TSUP2 and the stage pressures (i.e., the pressure in HEX1, HEX2 and HEX3) as a function of FMX for the case SER (55, 150, 1.0). As the Stage-2 pressure increases, TSUP2 decreases as one would expect (recall that the Stage-2 temperature tends to be fairly constant for FMX > 20%). Beyond FMX ≃ 28% (the point at which Stage 3 starts to contribute to the compression), the Stage-2 pressure decreases and TSUP2 increases. The changing heat loads on HEX1 and HEX2 also influence the shape of the TSUP2 curve, but its main features can be understood in terms of pressure, for a given value of CWT.

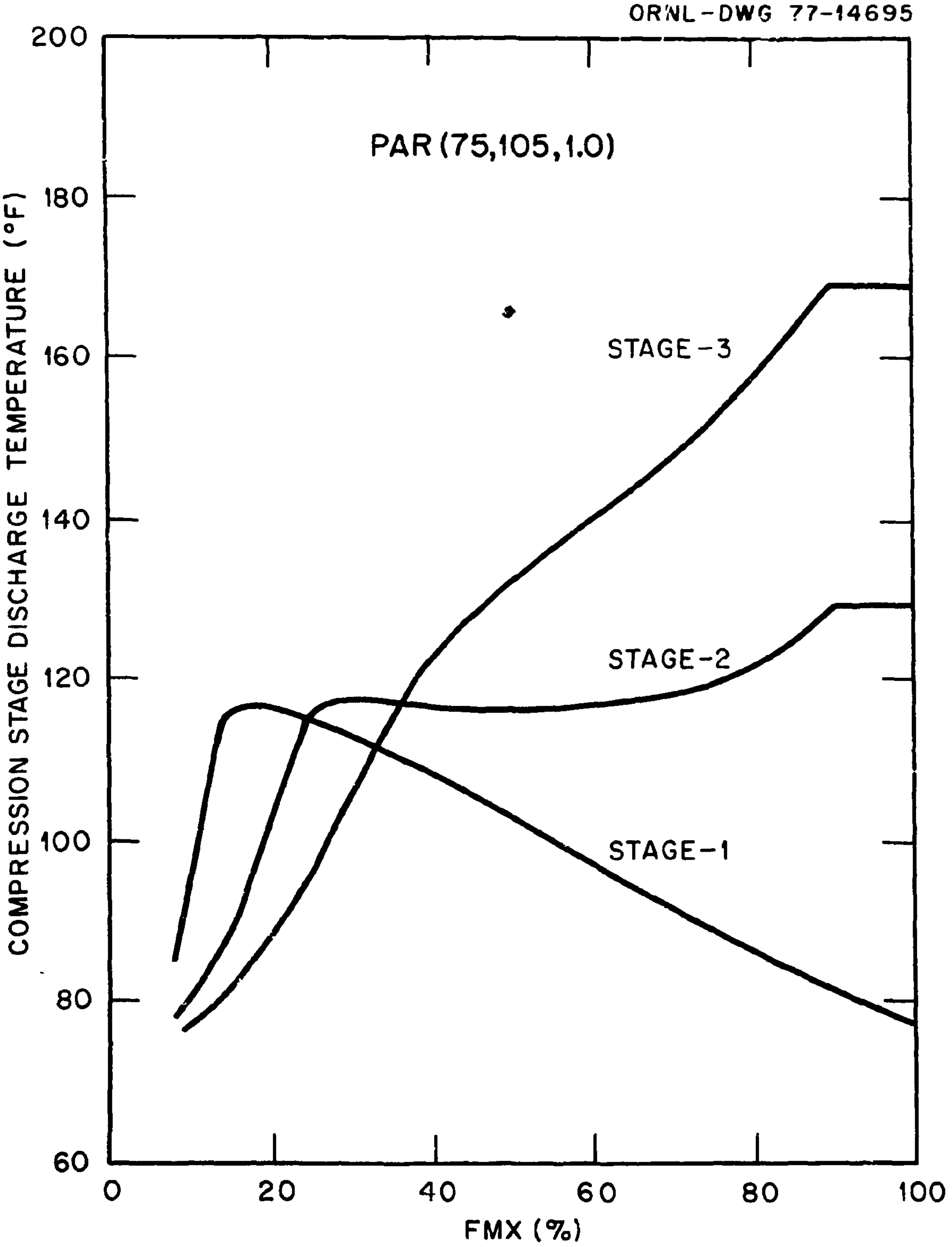

Fig. 11 Compression-stage discharge temperatures as functions of FMX.

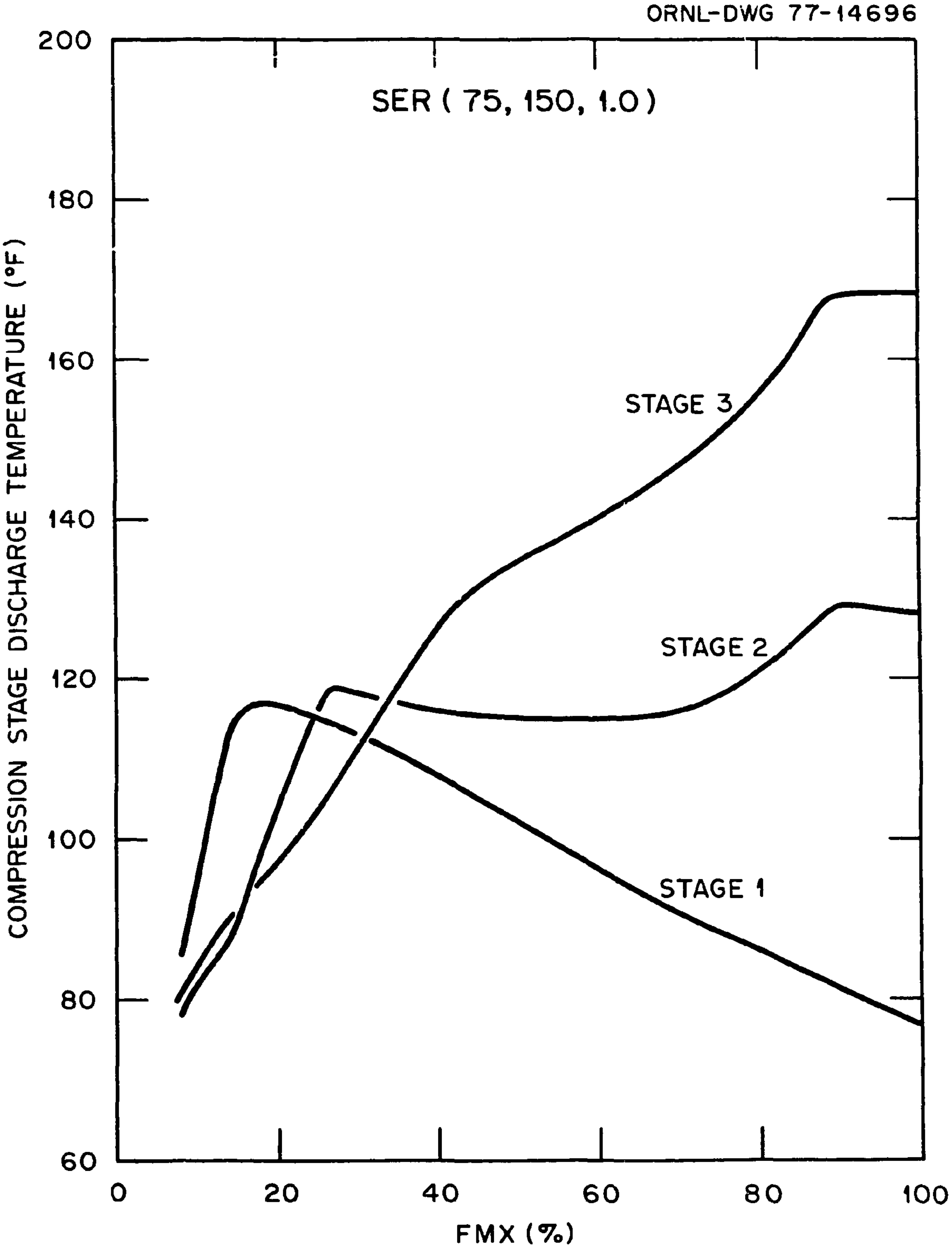

Fig. 12 Compression-stage discharge temperatures as functions of FMX.

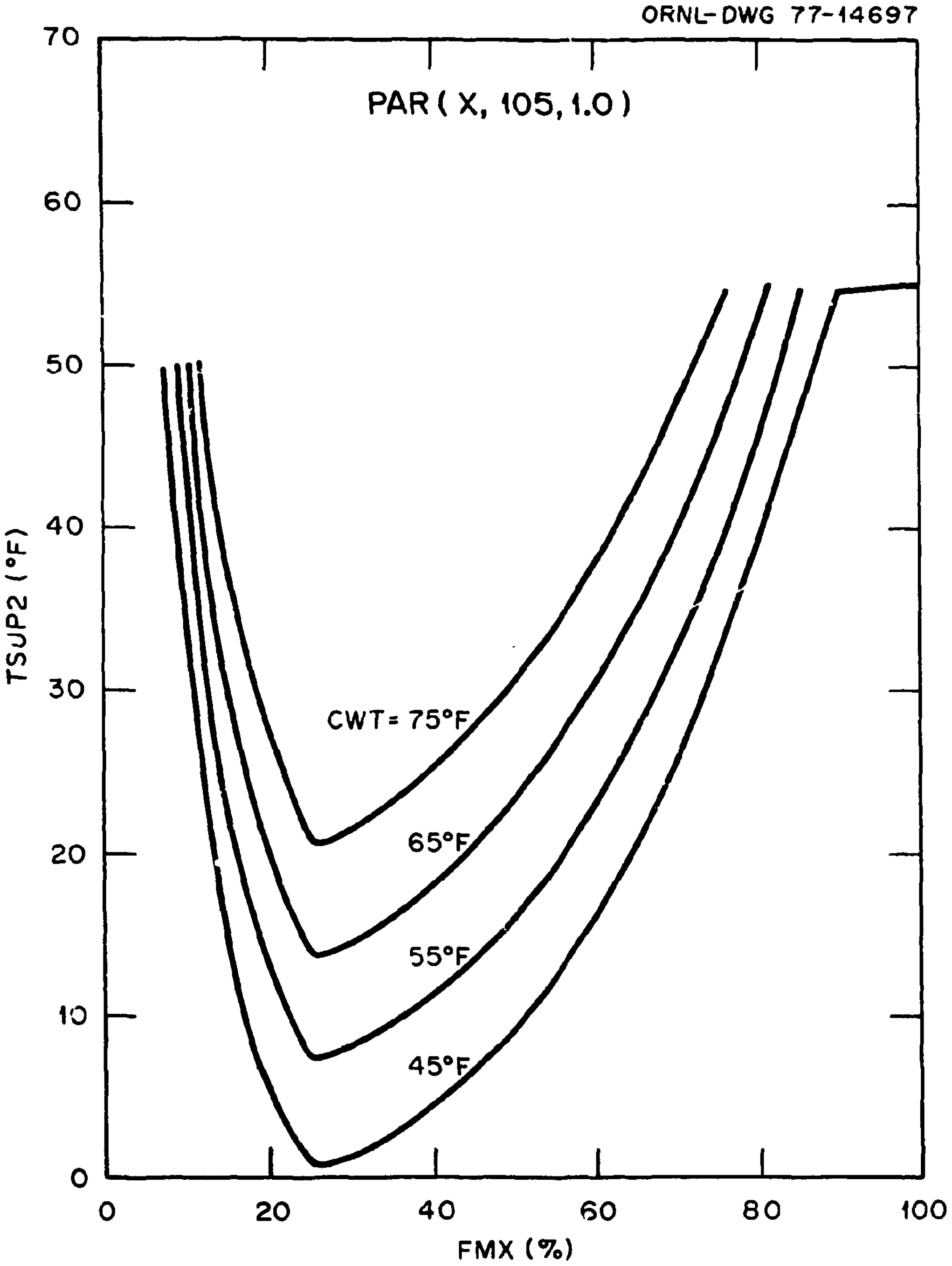

Fig. 13 The amount of "superheat", TSUP2, for SF_6 discharged from HEX2 is presented as a function of FMX and CWT for the PAR (X, 105, 1.0) configuration.

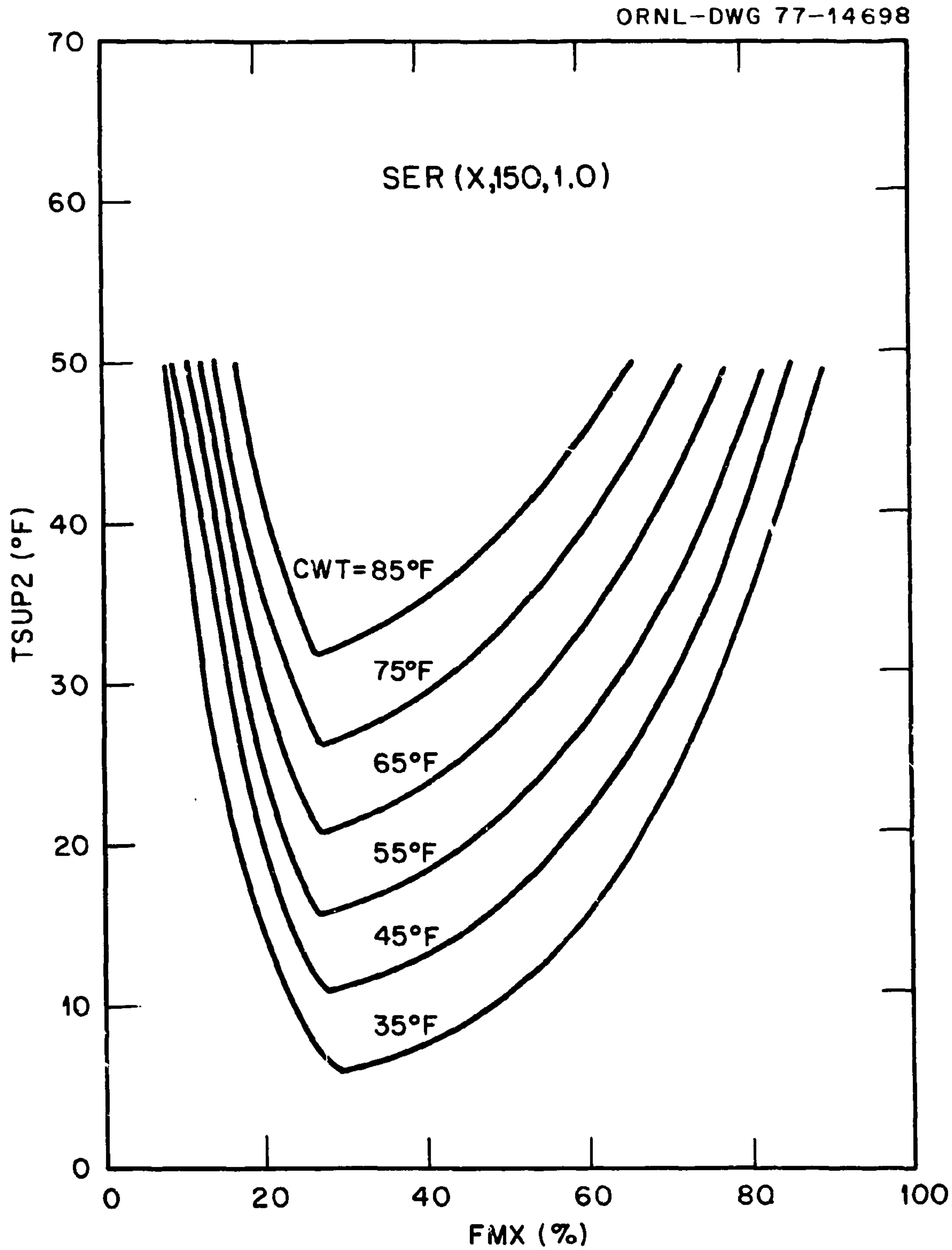

Fig. 14 The amount of "superheat", TSUP2, for SF_6 discharged from HEX2 is presented as a function of FMX and CWT for the SER (X, 105, 1.0) configuration.

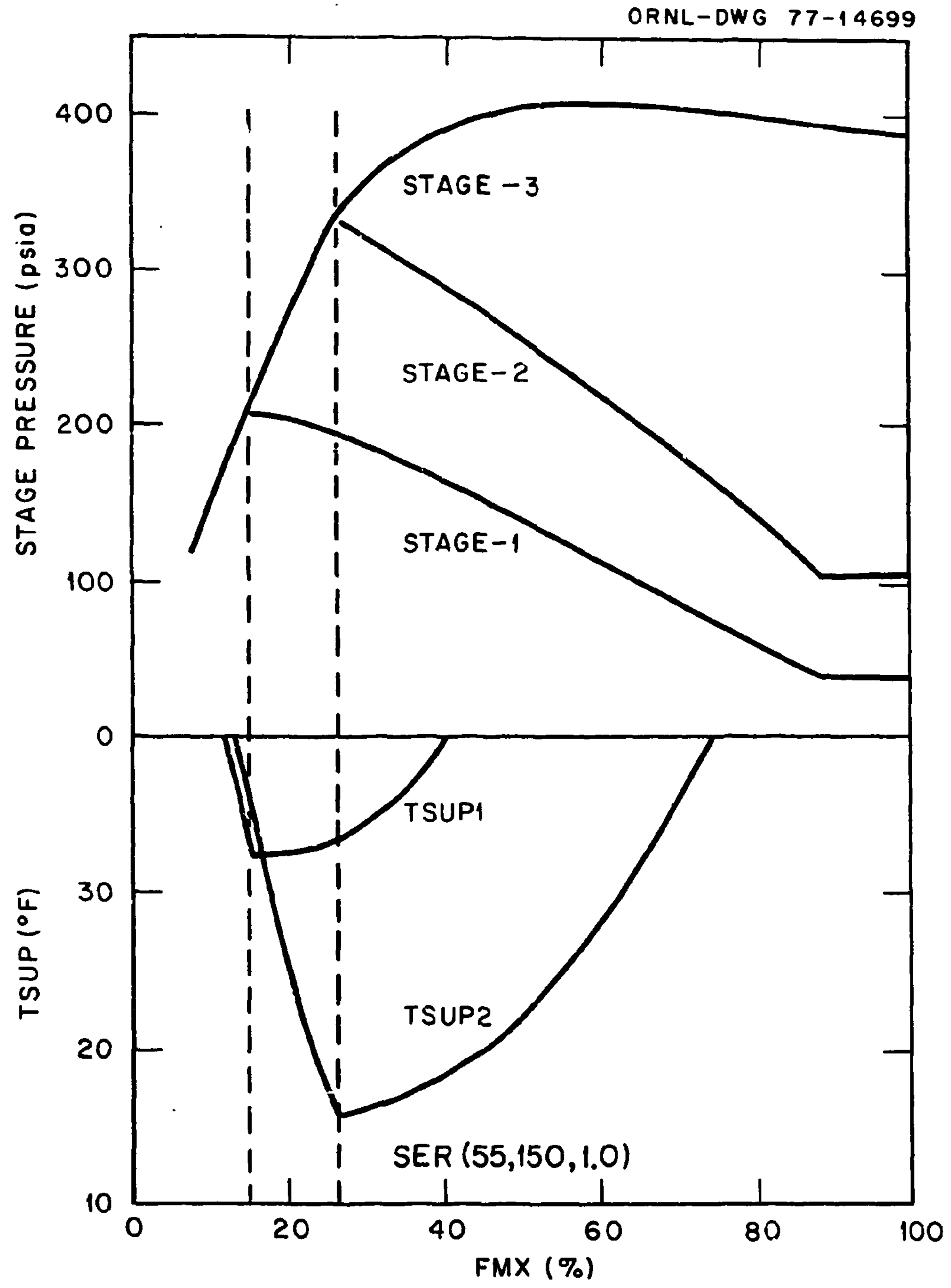

Fig. 15 The discharge pressures of SF_6 from Stages 1, 2, and 3 and the amounts of "superheat" for SF_6 discharged from HEX1 and HEX2 (TSUP1 and TSUP2) are displayed as functions of FMX. The configuration is SER (55, 150, 1.0).

Figure 15 also illustrates why TUSP1 is always larger than TSUP2 in the region where TSUP is critical. Observe that the maximum Stage-1 pressure is about 120 psi less than the maximum Stage-2 pressure. Although the discharge temperature from HEX1 is usually less than that from HEX2 (see Section 3.5), the stage pressure is the dominant factor.

Figure 16 illustrates how the water temperature (CWT2) at the inlet of HEX2 "tracks" with the requirement for less cooling in the region, defined by FMX = 20 - 50%.

3.5 Various SF_6 Temperatures as a Function of FMX

The temperature of the SF_6 discharged from HEX1, HEX2 and HEX3, as well as the storage-tank temperature, is presented as a function of FMX in Fig. 17. The initial rise in the discharge temperatures of HEX1 and HEX2 is due to increasing compression ratios, whereas the subsequent decrease in these temperatures is a result of decreasing SF_6 flow rate. The discharge temperature from HEX3 is strongly influenced by the heat load imposed by SF_6 liquefaction and is more easily discussed in conjunction with results displayed in Fig. 18.

3.6 Storage-Tank Dynamics

In Fig. 18 curves are displayed which are intended to clarify the behavior of the storage-tank temperature (and pressure) throughout the transfer. Observe that the storage temperature (curve 1) increases rapidly, primarily due to compression, until SF_6 starts to condense in HEX3 (at FMX ≈ 28%, see curve 4). Initially, all of the liquid injected into storage evaporates (see curve 3) and slows the heating process. In the region FMX ≈ 35 - 55°, this evaporation results in a cooling of the SF_6 in storage. At the same time (in the region FMX = 28 - 48%) this condensation imposes a larger and larger heat load on HEX3 and causes the discharge temperature to rise (see curve 2). At FMX = 48%, HEX3 is producing 100% liquid. At FMX = 50%, liquid begins to form in storage.

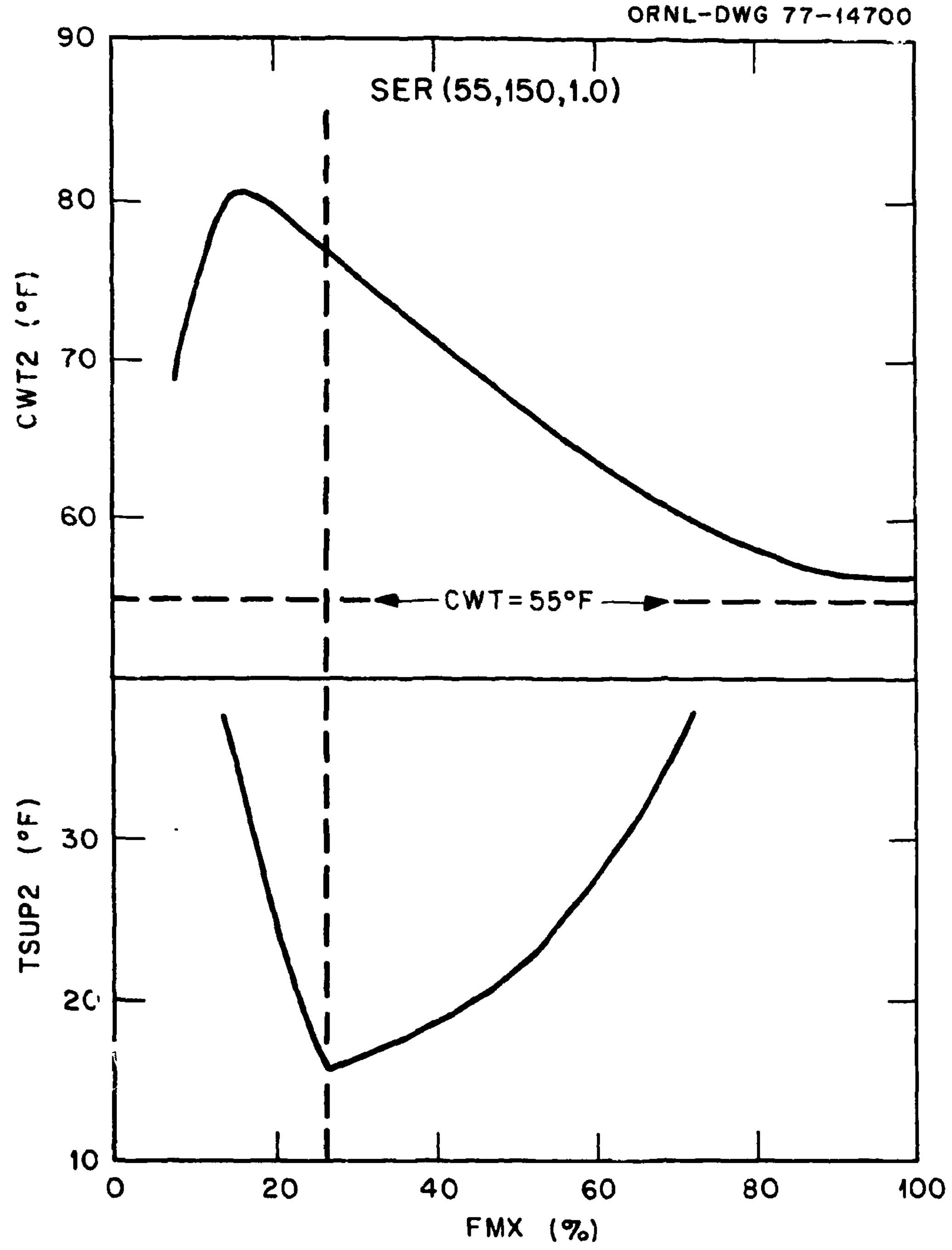

Fig. 16 The temperature of cooling water discharged from HEX1 (and injected into HEX2), CWT2, and the amount of "superheat" for SF_6 discharged from HEX2 are given as a function of FMX. CWT1 = CWT3 = 55°F.

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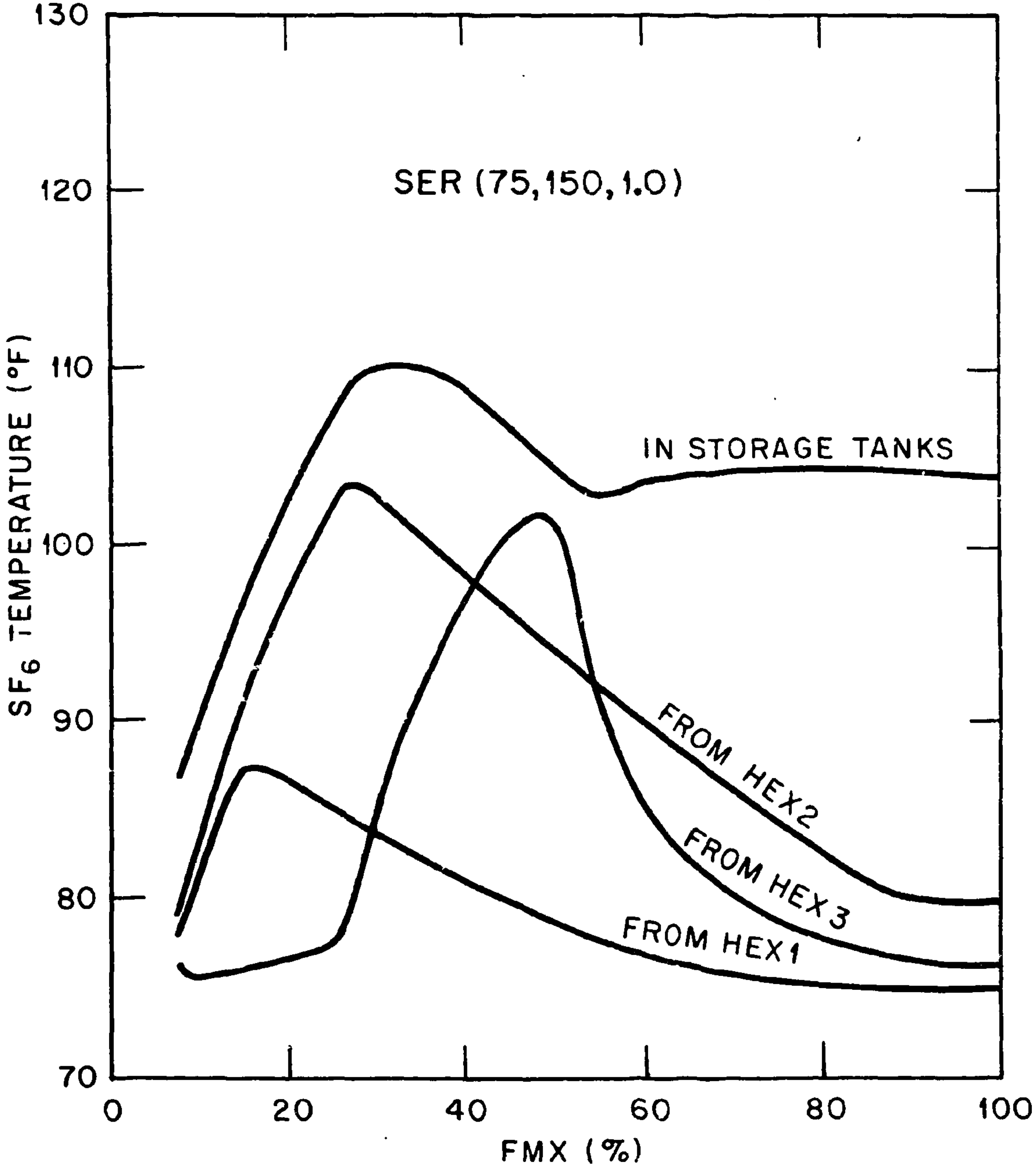

Fig. 17 Temperatures of SF_6 at various points in the system (at the discharge of HEX1, HEX2, and HEX3 and in the storage tanks) are given as functions of FMX for the configuration SER (75, 150, 1.0).

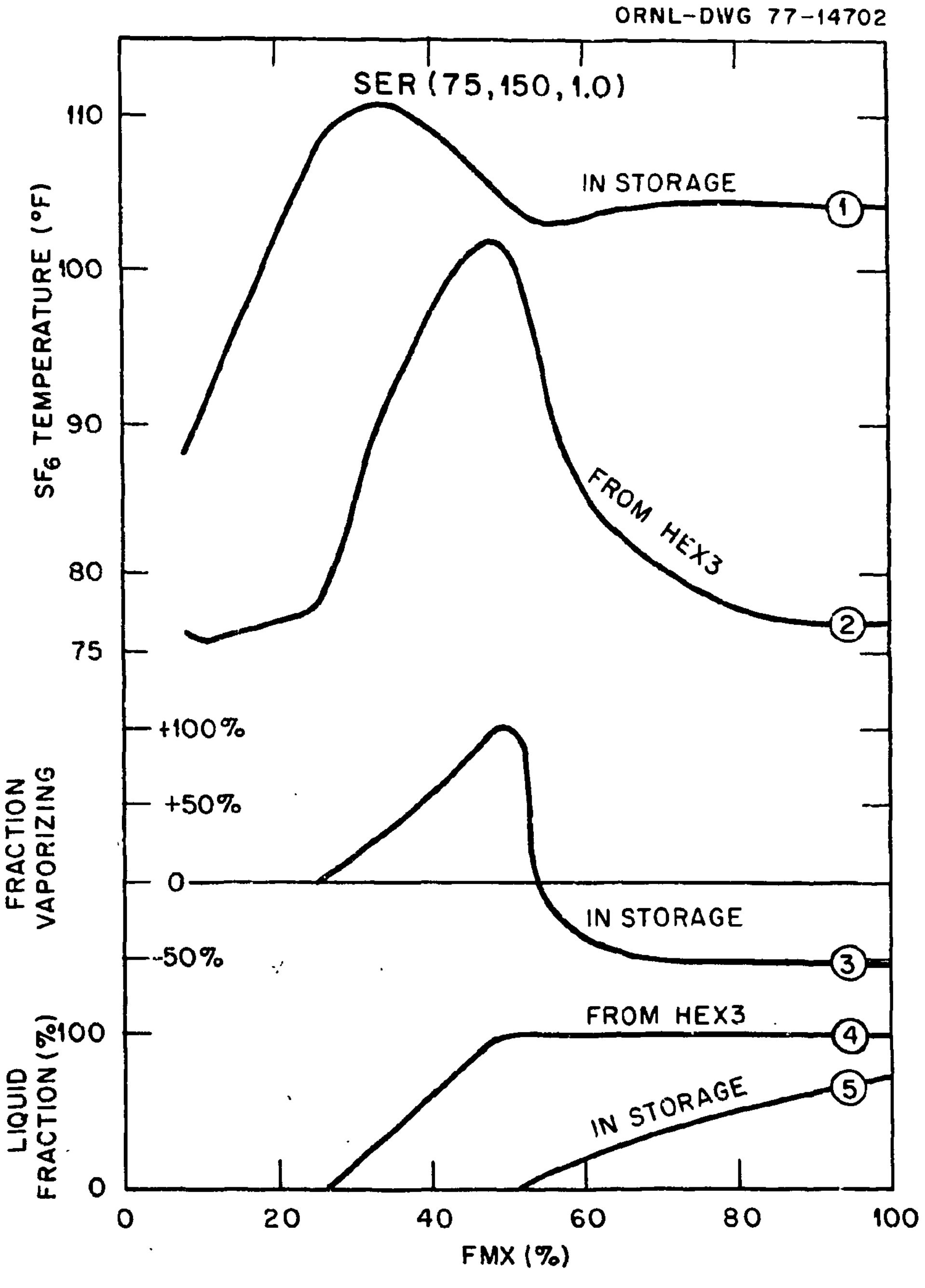

Fig. 18 The temperatures of SF_6 in storage and discharged from HEX3 are given as functions of FMX. The fraction of SF_6 being transferred to storage which vaporizes inside the storage tanks is displayed *vs.* FMX. The fraction of SF_6 discharged from HEX3 that is liquid and the fraction of the storage tank contents that is liquid are displayed *vs.* FMX. The configuration is SER (75, 150, 1.0).

At FMX ≈ 52%, more liquid is being formed in the sto[illegible]nks than is being injected, giving rise to an increase in the s[illegible] temperature. At the same time, however, the HEX3 discharge temperature is decreasing due to the reduction in the SF_6 flow rate. Of course, work is still being done on the SF_6 gas in storage. The sum of these effects leads to a rather constant storage temperature during the final stages of the transfer (FMX = 70 - 100%).

3.7 Heat Load on HEX3

The heat load on HEX3 is presented as a function of FMX in Fig. 19.

3.8 Storage-Tank Temperature and Pressure as a Function of FMX

Storage-tank temperatures for CWT = 35 - 85°F are shown as a function of FMX in Fig. 20. The intersections of dashed line (1) with the temperature curves define the points in the transfer at which liquid begins to form in HEX3. Dashed line (2) similarly indicates the points at which liquid SF_6 first appears in the storage tanks.

Figure 21 shows the storage-tank pressure (for CWT = 35 - 85°F) as a function of FMX.

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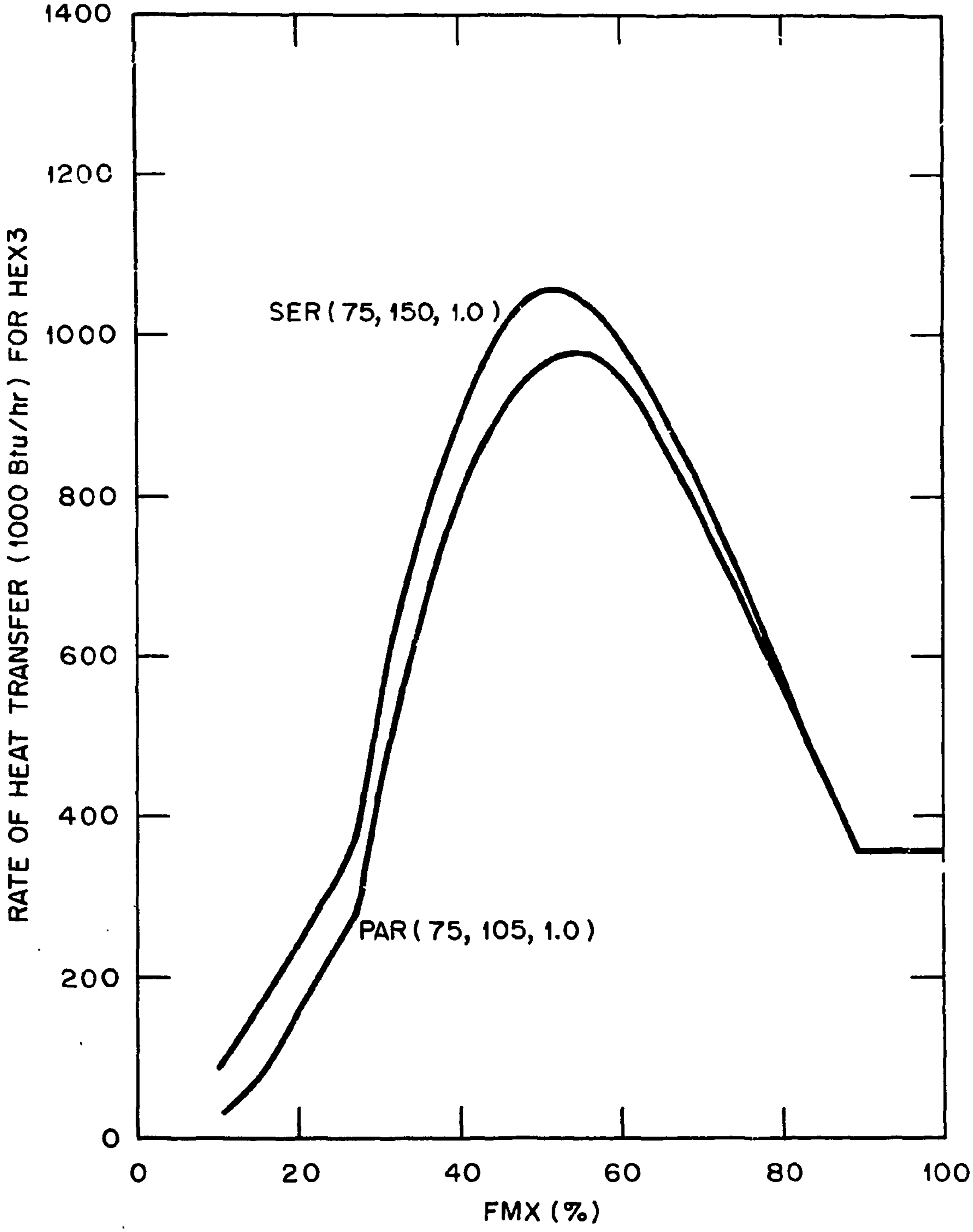

Fig. 19 The heat load on HEX3 is given as a function of FMX for the configurations SER (75, 150, 1.0) and PAR (75, 105, 1.0).

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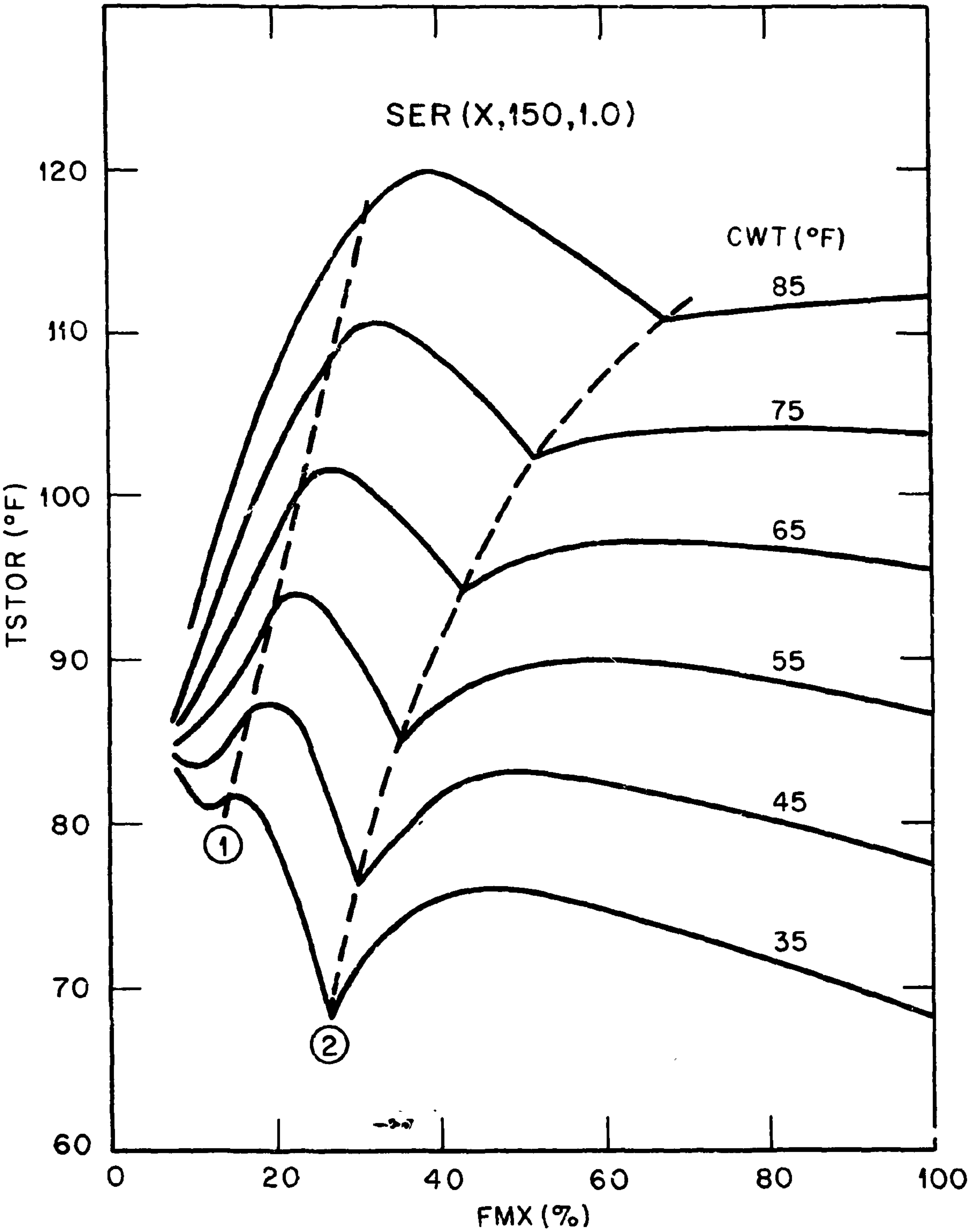

Fig. 20 The temperature of SF_6 in storage, TSTOR, is given as a function of FMX and CWT. See text for an explanation of the significance of the dashed curves (1) and (2).

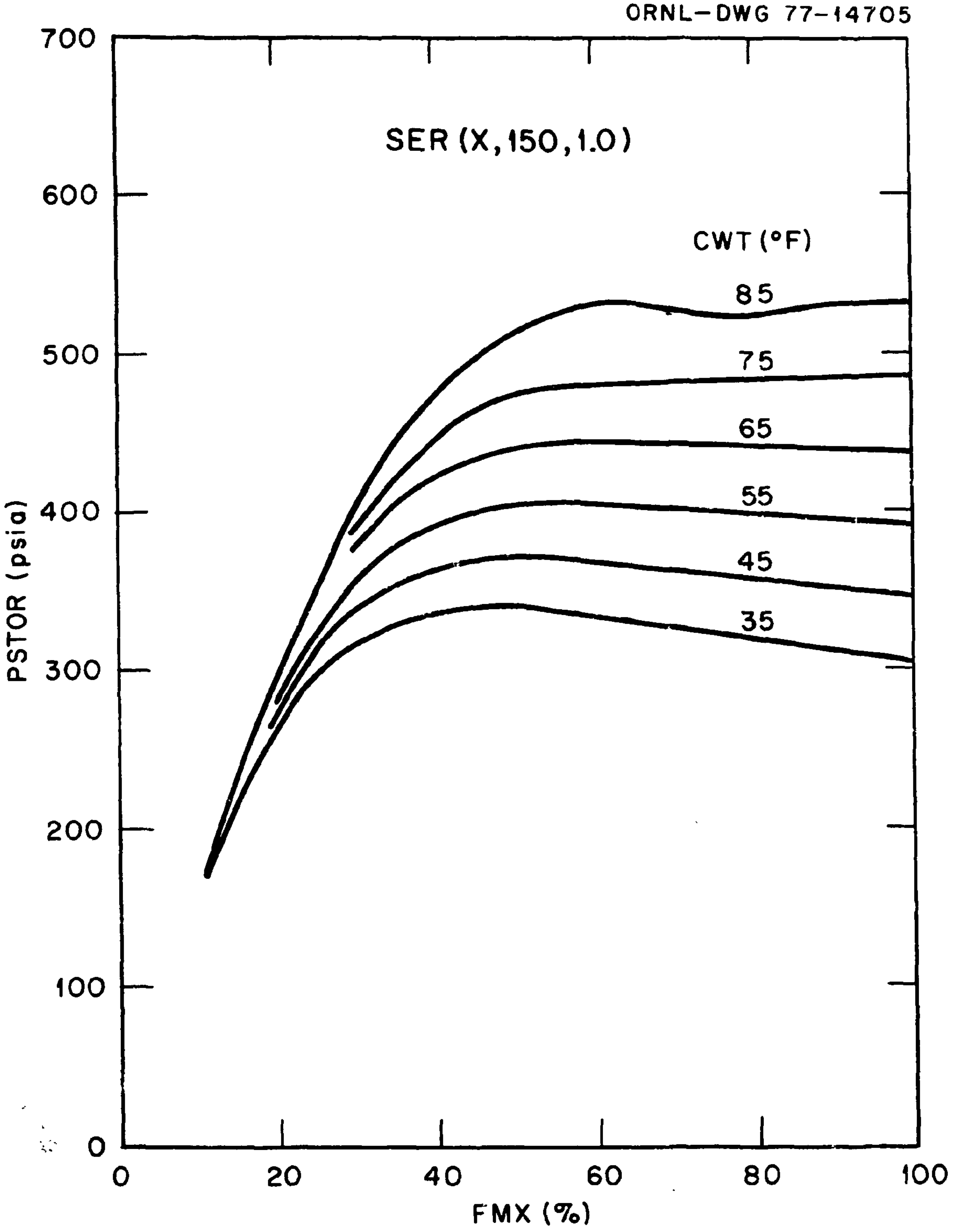

Fig. 21 The pressure of SF_6 in storage is given as a function of FMX and CWT.

4.0 GENERAL COMPUTATIONAL PROCEDURE FOR SF_6 TRANSFER

(1) Assume or read in

- (A) SF_6 inventory (270,000 lbs.)
- (B) Storage-tank volume (6000 ft^3)
- (C) Accelerator tank temperature
- (D) Cooling-water temperature
- (E) Cooling-water flow rates
- (F) Heat-exchanger configuration (series or parallel)
- (G) Heat-transfer coefficients
- (H) Integration step size, etc.

(2) Open the valve and let the accelerator vessel and storage tanks come to pressure equilibrium. Assume that the SF_6 temperature is everywhere equal to the value read in step (1C) above.

(3) Calculate the total energy, USTOR, in the storage tanks.

(4) Turn on the compressor.

(5) Calculate the SF_6 flow rate and stage compression ratios as described in Section 7.

(6) Calculate the heat transfer in HEX1, HEX2 (as described in Section 5) and HEX3 (as described in Section 6).

(7) Calculate the temperature and mass fraction liquified for the SF_6 being transferred into storage.

(8) Assume that a small fraction (∿ 0.5 to 1.0% of the total inventory) of the SF_6 is transferred to storage under fixed conditions (i.e., suction pressure, discharge pressure, SF_6 flow rate, temperature, liquid fraction, etc.)

(9) Calculate the energy content of the SF_6 added to storage and the work done on the SF_6 gas in storage.

(10) Calculate the total mass, total energy content, temperature, pressure, and liquid fraction of the storage-tank contents.

(11) Calculate the average pressure in storage during the transfer of this mass increment and recalculate the work done on the gas in storage. Repeat steps (9) and (10) in order to arrive at a more accurate energy content, temperature, pressure, etc.

(12) Recalculate the compressor fore- and back-pressures and go back to step (5).

Repeat steps (5) through (12) until the accelerator vessel pressure decreases to 15 psia. At this point assume that the high-speed vacuum pumps maintain the fore-pressure at 15 psia until all the SF_6 has been transferred.

5.0 HEAT-TRANSFER CALCULATIONS FOR EXCHANGER STAGES 1 AND 2

Heat exchanger-1 (HEX1) is a single-pass counter flow unit, as diagrammed in Fig. 22a. Heat exchanger-2 (HEX2) is a two-pass unit, as illustrated in Fig. 22b. Definitions pertinent to heat-transfer calculations are:

TW1 - Cooling-water inlet temperature
TW2 - Cooling-water outlet temperature
TS1 - SF_6 inlet temperature
TS2 - SF_6 outlet temperature
PS1 - SF_6 inlet pressure
PS2 - SF_6 outlet pressure
HS1 - Enthalpy of SF_6 at inlet
HS2 - Enthalpy of SF_6 at outlet
HV2 - Enthalpy of SF_6 saturated vapor at outlet
HL2 - Enthalpy of SF_6 liquid at outlet
FS - SF_6 flow rate
FW - Cooling-water flow rate
CPW - Specific heat of water ≈ 1 Btu/(lb. °F)
FLIQ - Mass fraction of SF_6 that liquifies in the exchanger
FVAP - Mass fraction of SF_6 that exits the exchanger as vapor
U - Overall heat-transfer coefficient of exchanger
A - Area of heat-transfer surface
DQ - Rate of heat transfer (Btu/h)
MTD - Mean temperature difference

For the SF_6 flow rates with which we are concerned, the SF_6 pressure drop across the exchanger is quite small (1-3 PSI). We have assumed that PS2 = PS1.

The rate of heat transfer in HEX1 and HEX2 may be expressed by an equation of the form

$$DQ = U \cdot A \cdot MTD\ (TW1,\ TW2,\ TS1,\ TS2), \qquad (5.1)$$

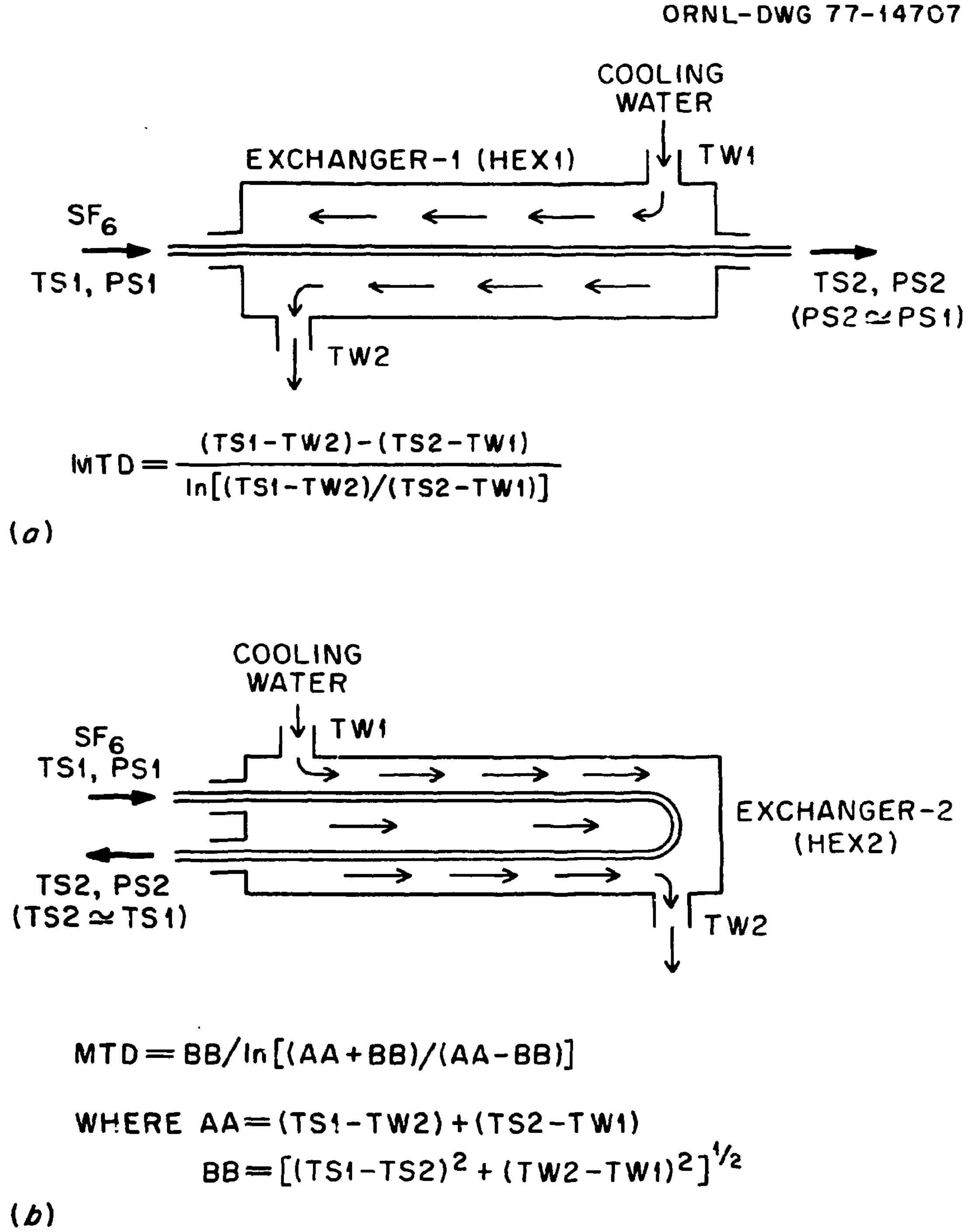

Fig. 22 Diagram of the cooling water and SF_6 flow patterns for HEX1 and HEX2 along with the expressions for the mean temperature difference MTD, to be associated with each.

provided that MTD is given the appropriate definitions shown in Fig. 22a and 22b.

Of course TW2 and TS2 are not known at the outset, and it is not possible to calculate DQ from equation (5.1) alone. However, there are two other heat-transfer equations which must be satisfied, namely:

$$\mathrm{DQS} = \mathrm{FS}\ (\mathrm{HS1} - \mathrm{HS2}) \tag{5.2a}$$

or

$$\mathrm{DQS} = \mathrm{FS}\ [\mathrm{HS1} - (\mathrm{FVAP} \cdot \mathrm{HV2} + \mathrm{FLIQ} \cdot \mathrm{HL2})] \tag{5.2b}$$

$$\mathrm{DQW} = \mathrm{FW} \cdot \mathrm{CPW} \cdot (\mathrm{TW2} - \mathrm{TW1}) \tag{5.3}$$

We have given different symbols to the heat transfer calculated from equations (5.2) and (5.3) for convenience in referencing, only. If the equations (5.1), (5.2) and (5.3) are to be consistent, then

$$\mathrm{DQS} = \mathrm{DQW} = \mathrm{DQ}. \tag{5.4}$$

An iterative procedure (usually called the heat balance method) is used to achieve a simultaneous solution to equations (5.1), (5.2) and (5.3). This procedure is outlined briefly below:

(1) Choose a value for TS2

(2) Calculate HS2 and DQS

(3) Calculate TW2 = TW1 + DQS/(FW • CPW)

(4) Calculate DQ = U • A • MTD (TW1, TW2, TS1, TS2)

(5) Adjust TS2 (or FLIQ) and iterate steps 1 through 5 to minimize QDIF = ABSF (DQ-DQS)

Note that if any significant fraction of the SF_6 is liquified in the exchanger, the expressions for MTD given in Fig. 22 are no longer valid. Therefore, we have used the above procedure for HEX1 and HEX2 only. A description of the method used for HEX3 is given in the next section.

6.0 HEAT-TRANSFER CALCULATIONS FOR EXCHANGER/CONDENSER-3

Since this treatment is probably unconventional, we will describe it in some detail. The problem is to determine the behavior of a heat exchanger/condenser whose geometry is similar to that diagrammed in Fig. 23. The SF_6 flows through a bundle of small tubes which make four passes through a circulating bath of cooling water. There are baffels in the water flow path which force it to meander across the SF_6 tube bundles about two dozen times as it flows from one end of the exchanger to the other. The principal objective is to calculate the temperature and liquid fraction of the SF_6 discharged from the exchanger. We have some (but not much) interest in temperature of the discharged cooling water.

6.1 List of Definitions and Identification of "Given" and "to Determine" Quantities

TSI	- Temperature of inlet SF_6	Given
PSI	- Pressure of inlet SF_6	Given
PSO	- Pressure of discharged SF_6 = PSI	Given
TWI	- Temperature of inlet cooling water	Given
SMF	- SF_6 mass flow rate	Given
WMF	- Cooling-water mass flow rate	Given
AEX	- Total area of exchanger surface	Given
UG	- Heat-transfer coefficient in "gas" region	Given
UC	- Heat-transfer coefficient in "condensing" region	Given
UL	- Heat-transfer coefficient in "liquid" region	Given
TSO	- Temperature of discharged SF_6	To determine
TWO	- Temperature of discharged cooling water	To determine
XCON	- Fraction of SF_6 condensed	To determine
T	- Temperature	
P	- Pressure	
VS	- Specific volume (for SF_6)	
CPGAS	= Specific heat of superheated "gaseous" SF_6 (Btu/(lb. °F) Values of CPGAS(T,P) are obtained by three-point bi-directional interpolation of the values given in Section 9, Table II.	

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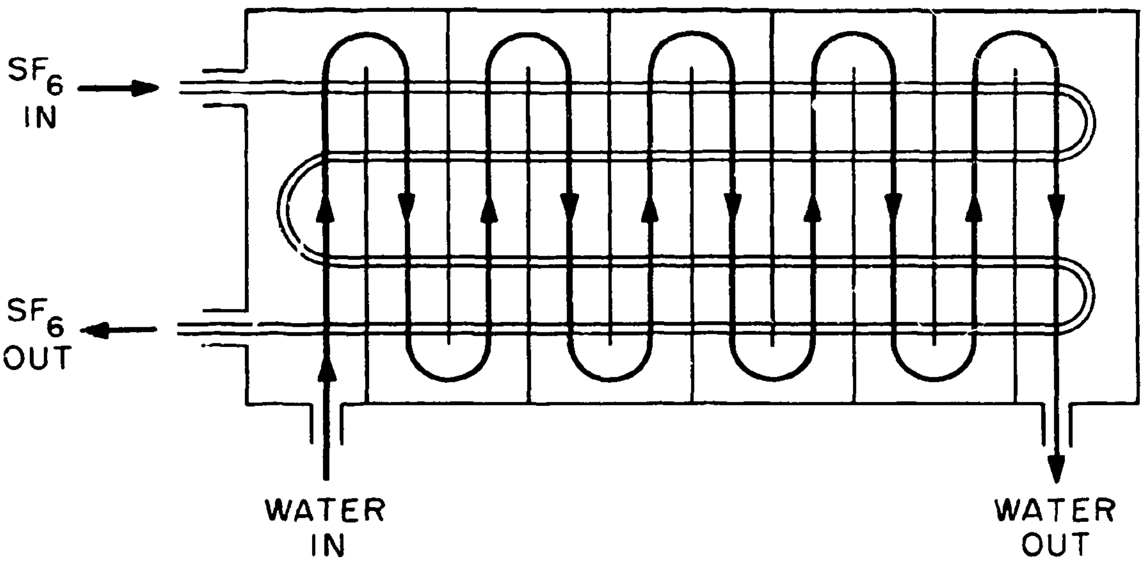

Fig. 23 Diagram of the cooling water and SF_6 flow patterns for HEX3.

CPLAT = Latent heat of vaporization of SF_6 (Btu/lb). Values of CPLAT (T) are obtained by interpolation of the thermodynamic data for gas and liquid in equilibrium. A representative sample of this data is given in Section 9, Table I.

CPLIQ = Specific heat of liquid SF_6 (Btu/lb. °F). Values of CPLIQ (T) are calculated by taking the derivative of the internal energy function, ULIP (T,P). That is

$$\mathrm{CPLIQ(T)} \simeq \frac{\Delta\ \mathrm{ULIQ(T,P)}}{\Delta \mathrm{T}},$$

where $\mathrm{ULIQ(T,P)} = \mathrm{HLIQ(T,P)} - \mathrm{P} \cdot \mathrm{VS}$, and $\mathrm{HLIQ(T,P)}$ = the enthalpy of liquid SF_6 at temperature, T, and pressure, P.

6.2 Computational Procedure

The computational procedure (a relaxation technique) used to solve this problem is outlined below.

1. The exchanger volume is divided into N elements (x-elements) in the x-direction as defined in Fig. 24.

2. Since the SF_6 tubes make four passes through the water bath, there will be 4N elements (s-elements) of path length along the SF_6 tubes.

3. We assume that there is no significant coolant temperature gradient in the y-direction, as defined in Fig. 24.

4. Initially, a linear water-temperature profile is set up in the exchanger such that:

$$\mathrm{TW(1)} = \mathrm{TW(2N+1)} = \mathrm{TW(4N+1)} = \mathrm{TWI}$$
$$\mathrm{TW(N+1)} = \mathrm{TW(3N+1)} = \mathrm{TSI}$$

   Note - TW(I) is the water temperature at the beginning of the Ith s-element and TW(I+1) is the temperature at the end of the Ith s-element.

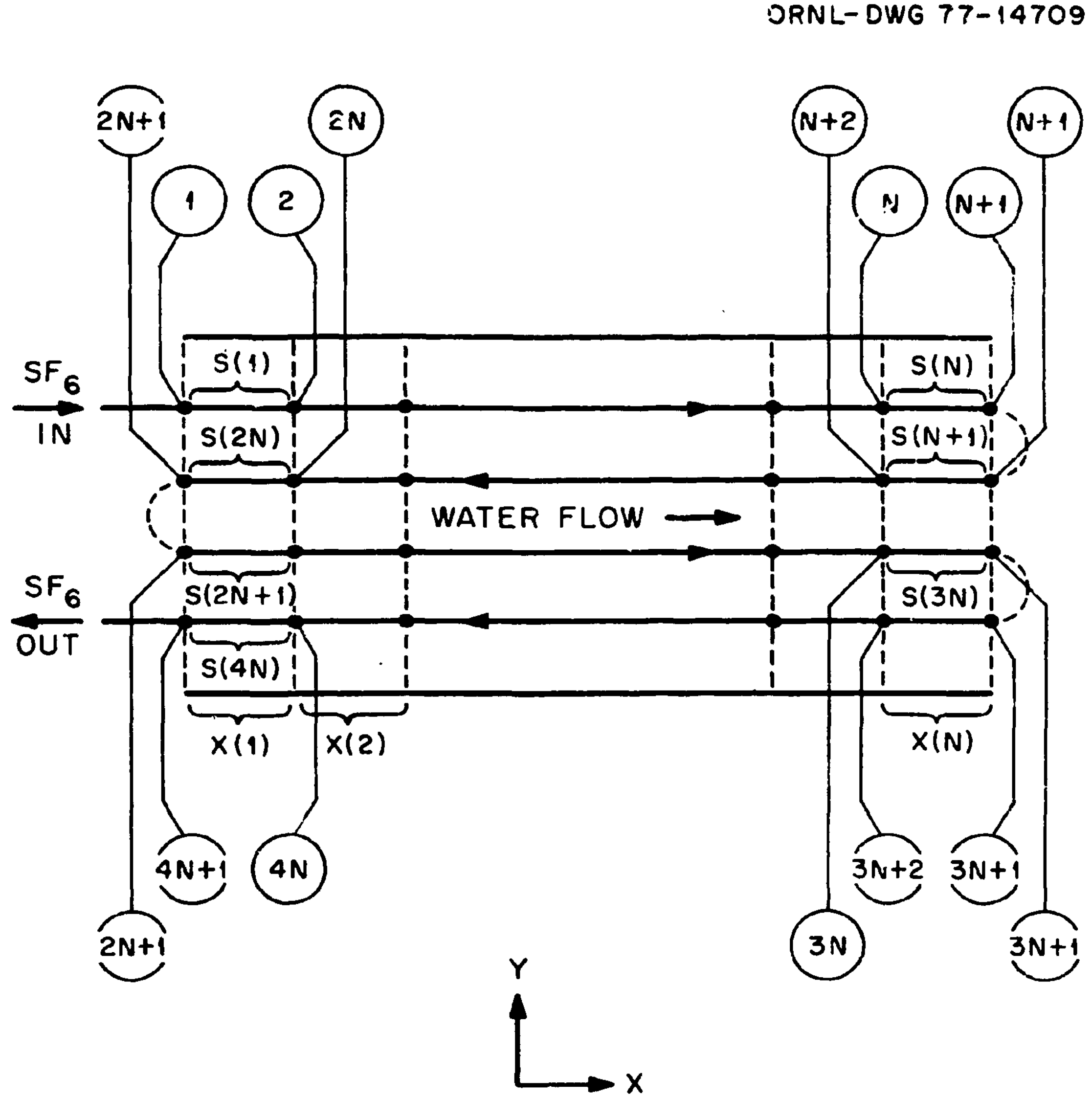

Fig. 24 Numbering scheme for points, x-elements, and s-elements used in the numerical treatment of the third stage exchanger/condenser, HEX3.

5. If we take the water-temperature profile as correct (for the moment) we can calculate the heat transferred across each s-element in succession. For the first s-element, the heat transferred per unit time, DQ(1), is given by,

$$DQ(1) = U(1) \cdot DA \cdot (TS(1) - TW(1))$$

and for the Ith s-element,

5a. $DQ(I) = U(I) \cdot DA \cdot (TS(I) - TW(I))$.

TS(I) and TS(I+1) are the SF_6 temperatures at the beginning and end of the Ith s-element, respectively. U(I) is the heat-transfer coefficient to be applied to the Ith s-element. DA, the s-element area, is given by DA = AEX/(4N).

After a temperature profile has been established for both the coolant and the SF_6, the heat transferred in the Ith s-element is calculated as follows:

5b. $DQ(I) = U(I) \cdot DA \cdot (TSAV - TWAV)$

where

$$TSAV = 0.5 \cdot [TS(I) + TS\ (I+1)]$$
$$TWAV = 0.5 \cdot [TW(I) + TW(I+1)]$$

That is, we use the average temperatures at the Ith s-element.

6. Next, by using the SF_6 flow rate (SMF) and the appropriate SF_6 specific heat (CPGAS or CPLIQ) or latent heat (CPLAT), we can calculate the temperature change DT(I) (or the mass condensed DM(I)) as the SF_6 passes over the Ith s-element.

$$DT(I) = DQ(I)/(SMF \cdot CPGAS)$$

or

$$DT(I) = DQ(I)/(SMF \cdot CPLIQ)$$

or

$$DM(I) = DQ(I)/CPLAT$$

7. If the SF_6 temperature is above the dew point or all of the SF_6 is liquified, the temperature of the SF_6 entering the next s-element

is given by

$$TS(I+1) = TS(I) + DT(I).$$

7a. If the SF_6 temperature drops to the dew point, cooling is discontinued and condensing takes over until all s-elements are considered or until all SF_6 has been liquified. If condensation is incomplete we have,

$$TS(I+1) = TS(I)$$

and

$$XCON(I+1) = XCON(I) + DM(I)$$

Steps 5a through 7 or 7a are continued until all s-elements have been considered at least once. Steps 5b through 7 or 7a are executed in subsequent iterations.

At this point we compare the sum of the DQ(I) (the heat transferred across all elements) to the heat gained by the coolant.

$$QS = SUM\ (DQ(I),\ I = 1,\ 4N)$$

$$QW = FW\ (TWO - TWI)$$

$$XQ = ABSF\ ((QS - QW)/QS)$$

If XQ is less than .005 the heat balance is considered to be adequate and the iteration is terminated. Otherwise, the coolant temperature profile is adjusted as follows:

a) Sum all DQ's corresponding to x-element #1. (DQ(1), DQ(2N), DQ(2N+1), DQ(4N)).
b) Remove this amount of heat from the coolant and calculate the coolant temperature at the beginning of x-element #2.
c) Continue this process until all x-elements have been included.
d) Go back to step 5b and proceed with the next iteration.

Although the initial cooling-water temperature profile that we assumed was rather poor, realistic profiles develop rapidly. In addition, the parameters that determine the SF_6 and cooling-water temperature profiles change only slightly from one step in the SF_6 transfer calculation to the next. That is, once developed, the profiles change slowly, and only 2 or 3 iterations are required for each successive step in the transfer.

Figures 25 through 27 present examples of cooling water and SF_6 temperature profiles at different stages of the SF_6 transfer. The results shown in Fig. 25 correspond to an early stage of the transfer (SMF = 61,000 lb/h and PSTOR = 255 psia) where the SF_6 temperature at the input HEX3 is low (97°F) and no condensation takes place. Figure 26 illustrates a case where condensation starts at about one third of the distance through the exchanger but not all of the SF_6 is condensed. Observe that the SF_6 temperature is constant throughout the last two-thirds of its path (Pass 2 through Pass 4). The water temperature profile is more pronounced than in the previous example because of the larger heat load (910,000 versus 230,000 Btu/h). Figure 27 illustrates a case where the SF_6 is cooled to the dew point in about one-half pass and is fully condensed two-thirds of the way through the second pass. The liquid SF_6 is then further cooled during the final two and one-third passes. Notice that the SF_6 temperature tends to "track" the cooling-water temperature for the final one and one-half passes.

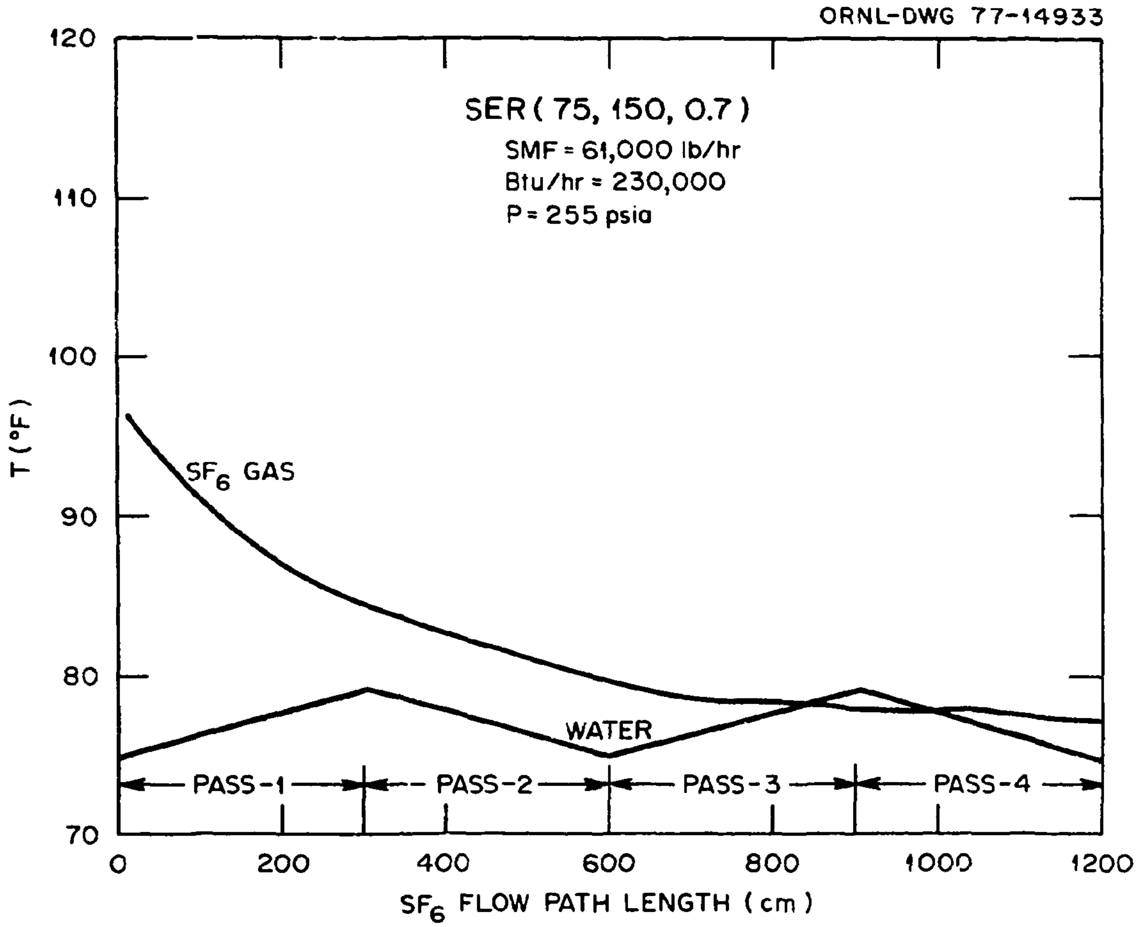

Fig. 25 Cooling water and SF_6 temperature profiles in HEX3 are given as functions of the SF_6 flow path length. This figure illustrates an early stage in the SF_6 transfer where no condensation occurs.

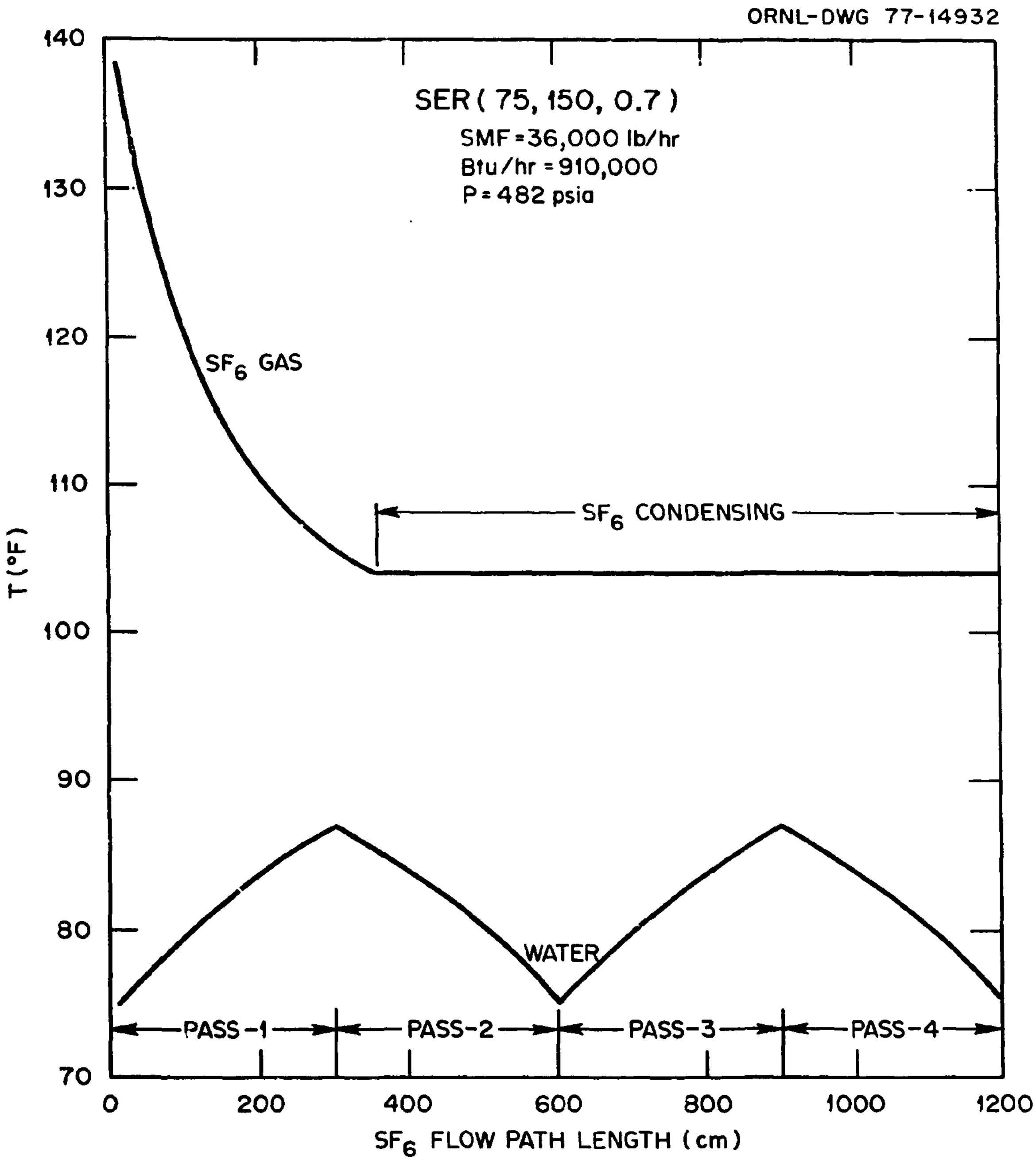

Fig. 26 Cooling water and SF_6 temperature profiles in HEX3 are given as functions of the SF_6 flow path length. This figure illustrates the situation where some, but not all, of the SF_6 is being condensed.

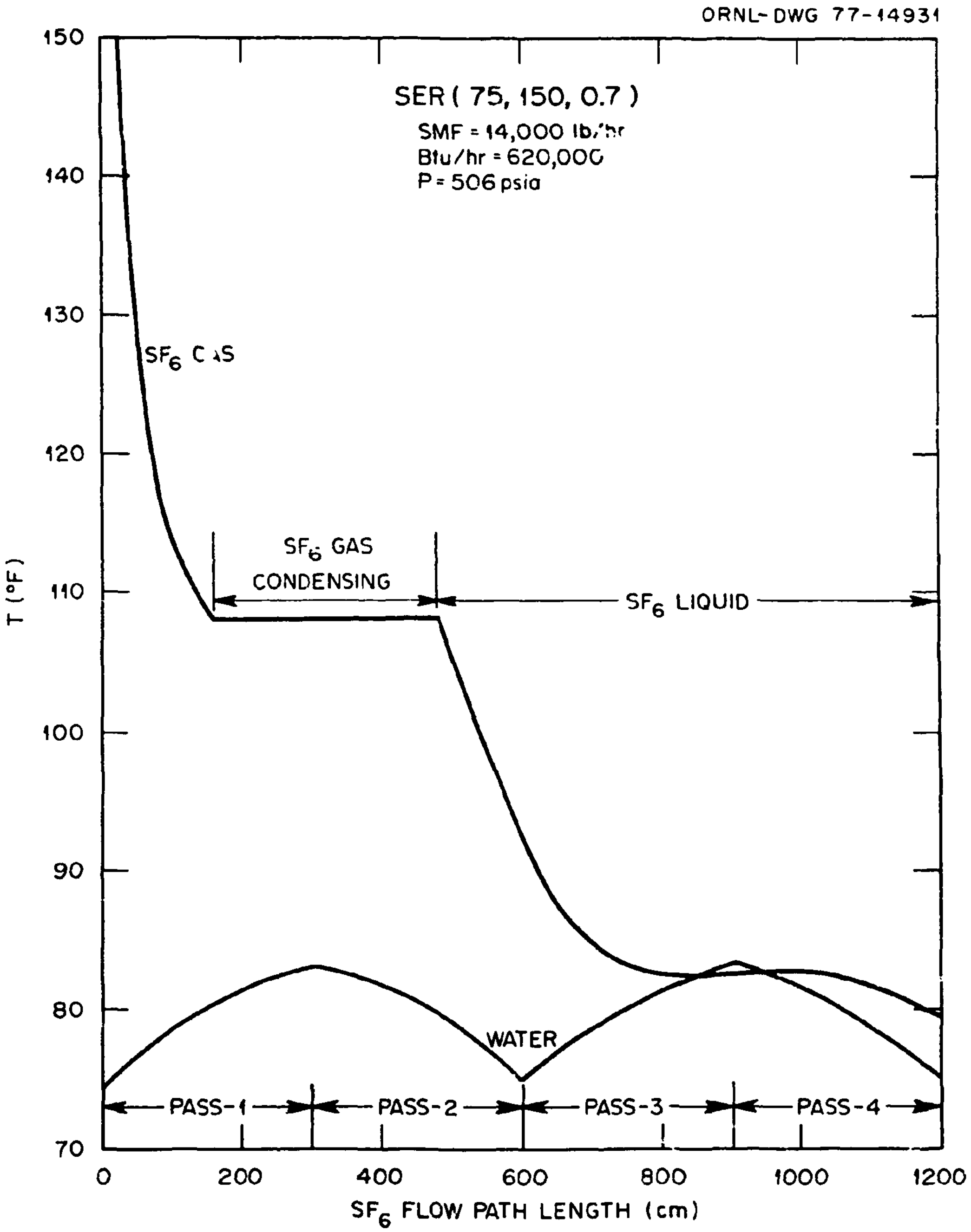

Fig. 27 Cooling water and SF_6 temperature profiles in HEX3 are given as functions of the SF_6 flow path length. This figure illustrates a latter stage of the transfer: the SF_6 flow rate is small (14,000 lb/h); all of the SF_6 is being condensed and supercooled.

7.0 CALCULATION OF SF_6 FLOW RATES AND STAGE COMPRESSION RATIOS

Figure 28 shows a diagram of one compression stage. The valves at either end of this double-acting compressor are pressure operated and are designed to allow gas to flow freely from the intake to the discharge manifold as long as P1 > P2. The reverse flow is not permitted. Normally (when P2 > P1), the compressor operates (considering one side of the piston only) as follows:

(1) Starting with the piston in an extreme left position, the cylinder with volume equal to VMAX is filled with gas at pressure P1 at temperature T1.

(2) The piston then compresses this volume of gas to pressure P2 and temperature T2 (see Section 7.2).

(3) Gas is then exhausted at constant pressure and temperature while the piston moves to the extreme righthand position. A volume of gas, VMIN (at P2 and T2), remains in the cylinder when the piston starts its return motion.

(4) As the piston returns, the remaining gas expands isentropically to P1 and T1, at which time gas from the intake manifold begins to flow into the cylinder.

The other side of the piston produces the same kind of action, with a flow rate which is slightly reduced due to the volume occupied by the piston rod.

Definitions pertinent to the calculation of flow rates and compression ratios:

VSTR - Volume of one piston stroke
VMIN - Minimum piston volume (head volume)
VMAX - VMIN + VSTR
PD - Piston displacement per minute

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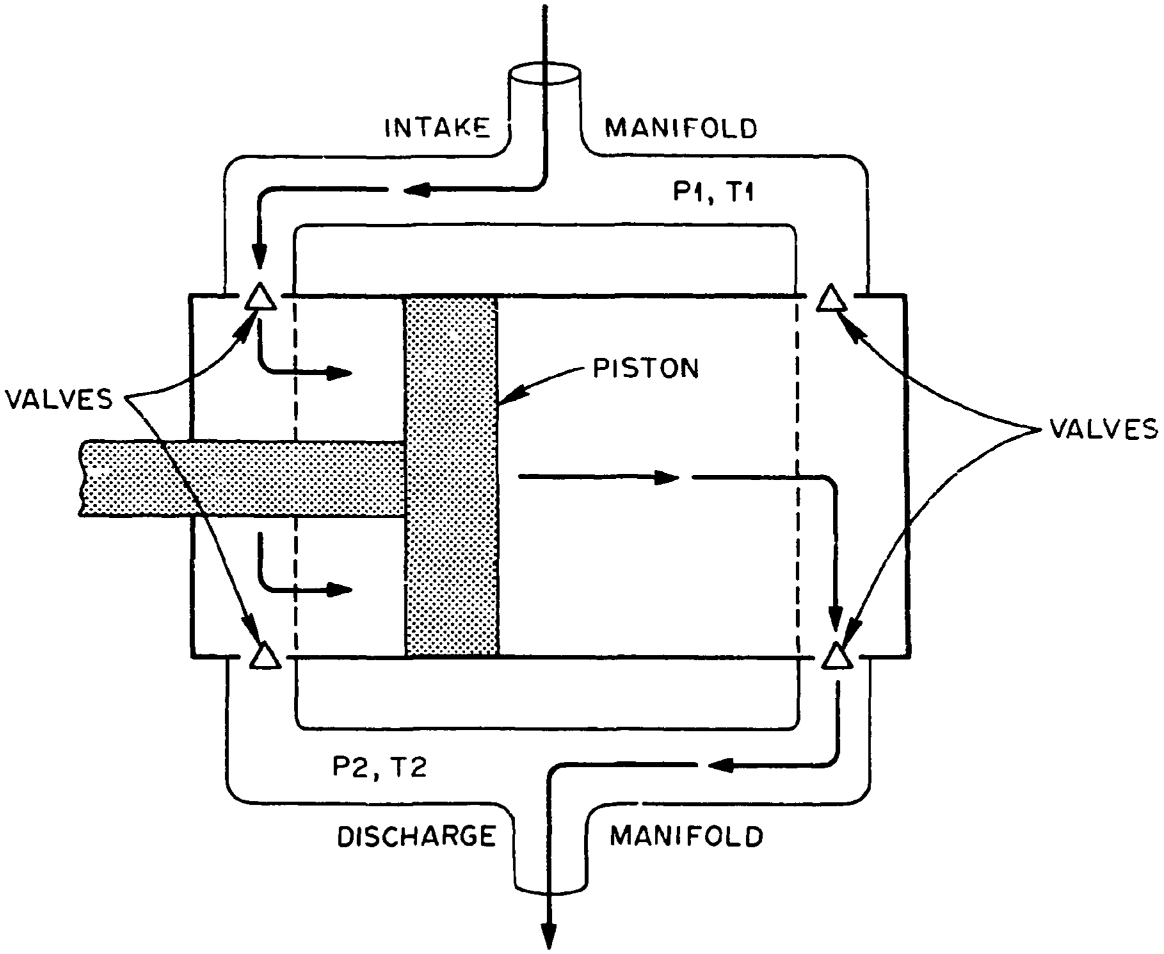

Fig. 28 Diagram of one double-acting compression stage of the Cooper-Penjax compressor.

DP - Diameter of piston
DR - Diameter of rod
XS - Length of stroke
T1 - Intake gas temperature
T2 - Discharge gas temperature
CP - Specific heat at constant pressure
CV - Specific heat at constant volume
P1 - Intake gas pressure
P2 - Exhaust gas pressure
RHO - Gas density
MGI - Mass of gas in cylinder before compression stroke starts
MGF - Mass of gas remaining in cylinder at end of compression stroke
DMG - Mass of gas discharged in one stroke
FS - SF_6 flow in lb/min
RPM - Compressor speed (strokes/min)

Considering both sides of the piston we have,

$$PD = \pi \; (1/2 \; DP^2 + 1/4 \; DR^2) \cdot XS \cdot RPM$$

Now consider one compression stroke. Initially, the volume to the right of the piston contains a mass of gas,

$$MGI = RHO(P1,T1) \cdot (VSTR + VMIN).$$

After compression the cylinder contains a mass of gas,

$$MGF = RHO \; (P2,T2) \cdot VMIN.$$

Let RHO1 = RHO (P1,T1) and RHO2 = RHO (P2,T2).

The amount of gas discharged is given by DMG = RHO1 (VSTR + VMIN) - RHO2 · VMIN.

$$DMG = VSTR \cdot RHO1 \; [1.0 - \frac{RHO2 - RHO1}{RHO1} \cdot \frac{VMIN}{VSTR}]$$

If: VEF = [ ] = Volumetric efficiency

then,

$$DMG = VSTR \cdot RHO1 \cdot VEF$$

This definition of VEF assumes perfect valve operation. It seems that Cooper-Penjax has used a valve efficiency factor of 0.93, which results in VEF → 0.93 as P1 → P2.

We have therefore redefined VEF as follows,

$$VEF = 0.93 - \frac{VMIN}{VSTR} \cdot \frac{(RHO2 - RHO1)}{RHO1} .$$

We are now able to calculate the SF_6 flow rate through a single stage of the compressor in terms of the compressor geometry, piston displacement, the temperature and pressure of the intake and exhaust pressure.

$$FS\ (lb/min) = RHO1\ (T1,P1) \cdot VEF\ (T1,P1,T2,P2) \cdot PD$$

In practice we have three stages in series and are given only the temperature and pressure at the inlet to Stage 1 and the pressure at the outlet of Stage 3. Of course, we have the additional restriction that the flow through all stages must be the same. We use the following iterative procedure to arrive at compression ratios for each stage that will produce equal flow in each stage.

(1) Set the inlet temperature and pressure for Stage 1 and the exhaust pressure for Stage 3 to the given (fixed) value.

(2) Choose (guess) an inlet and exhaust pressure for Stage 2. This automatically sets the exhaust pressure for Stage 1 and the inlet pressure for Stage 3.

(3) Calculate the flow, exhaust temperature (after passing through heat exchangers) for each stage in succession.

(4) Adjust the inlet and exhaust pressure for Stage 2 so as to minimize the difference in flow rates between Stages 1, 2 and 3. Go back to step (3).

This procedure would be quite slow except that in a normal transfer calculation the inlet and exhaust conditions [see step (1)] change slowly from one integration step to the next. Therefore, the initial guess [step (2)] can be quite good.

7.1 Reproduction of the Compressor-Performance Calculations of Cooper-Penjax

To assure ourselves that we understand how the compressor works and how to calculate its performance, we have calculated compression ratios, flow rates, volumetric efficiencies, etc., under the same conditions as used by Cooper-Penjax. To achieve satisfactory agreement, the following assumptions were required: (1) SF_6 is an ideal gas (i.e., P1 • V1/T1 = P2 • V2/P2); and (2) Compression is isentropic, with T2/T1 = (P2/P1) EXP (1-1/γ), where γ = Cp/Cv = 1.1.

We compared our calculations to those of Cooper-Penjax under the following set of conditions.

(1) The inlet temperature of Stages 1, 2 and 3 was fixed at 85, 90 and 90°F, respectively.

(2) Inlet pressure equaled 14.7 psia and the discharge pressure equaled 74.7, 84.7, 114.7, 214.7, 314.7, 414.7, 514.7 and 614.7 psia.

(3) Inlet pressure equaled 54.7 psia and the discharge pressure equaled 264.7, 274.7, 314.7, 414.7 and 614.7 psia.

Agreement was generally good, as shown in the following table:

Quantity	Agreement
Compression ratio	0 - 2%*
Temperature rise due to compression	within 1-2°F in 50-90°F
Volumetric efficiency	±1%
Flow rate	±1%**

*Cooper-Penjax included a slight pressure drop (1 - 3 psi) across the heat exchangers. Since we did not include this effect, our compression ratios are slightly smaller.

**Cooper-Penjax did not achieve flow balance through subsequent stages to better than about 1%.

7.2 Computation of Temperature Increase Due to Compression

Problem:

Compress a quantity of gas from an initial pressure P1 and temperature T1 to a new pressure P2. Calculate the new temperature T2.

Assumption:

The compression is assumed to be isentropic, i.e., no heat is transferred to or from the surrounding medium. This assumption is not perfect, but it should be very nearly so. Note that the compressor is expected to operate with a static water jacket or at most a very small flow of cooling water.

Method:

Calculate (by two-dimensional table interpolation) the entropy, S(T1,P1), before compression. Calculate a one-dimensional entropy table, at the new pressure P2, S(T,P2). Search and interpolate this table to find T2 = T(S1,P2).

8.0 CALCULATION OF HEAT-TRANSFER COEFFICIENTS

The overall heat-transfer coefficient for a heat exchanger of the type with which we are concerned is usually calculated by adding together the heat-flow resistances which arise from the following four sources:

1) R1 - due to the "fluid film" associated with the gas side,
2) R2 - due to the "fluid film" associated with the coolant (water) side,
3) R3 - 2 "fouling resistance" associated with the gas side,
4) R4 - a "fouling resistance" associated with the coolant side, that is,

$$\frac{1}{U} = R1 + R2 + R3 + R4 = \frac{1}{U1} + \frac{1}{U2} + \frac{1}{U3} + \frac{1}{U4} .$$

In our application, it turns out that R1 is not only the dominant resistance but is also the most variable and difficult to calculate with certainty. Typically,

$$U2 \simeq U3 \simeq U4 \simeq 1000 \text{ Btu/(hr} \cdot \text{ft}^2 \cdot {}^\circ\text{F)}$$

whereas

$$U1 \sim 100\text{-}200.$$

In the case of HEX3 (the third stage exchanger/condenser), we have three regions to consider in the most general case. In the first region the SF_6 gas is cooled to the dew point, in the second region partial or complete condensation takes place, and in the third region SF_6 liquid is further cooled. Therefore, we will have three transfer coefficients associated with HEX3.

The following definitions are associated with HEX3:

UGP = transfer coefficient in gas region due to SF_6 gas film only

UG = total transfer coefficient in gas region UG = 1/(1/UGP+R2+R3+R4)

UCP = transfer coefficient in condensing region due to SF_6 gas-liquid film only

UC = total transfer coefficient in condensing region UC = 1/(1/UCP+R2+43+R4)

ULP = transfer coefficient in liquid region due to SF_6 liquid film only
UL = total transfer coefficient in liquid region UL = 1/(1/ULP+R2+R3+R4).

8.1 Film Coefficients (Water Side)

For water flow normal to a bank of staggered tubes, McAdams[5] gives

$$U2 = 222(1 + 0.0067 \cdot Tf)\ VSM/DO^{0.4}$$

where Tf is the average of the bulk and wall temperature (in °F), VSM is the maximum velocity (through the minimum cross section in ft/sec), and DO is the o.d. of one tube in inches.

For HEX3 we have 65 tubes which make four passes through a 16-in diameter shell. The tube-to-tube clearance is 5/32 in., and the baffel spacing is 6.0 in. This leads to a minimum open area of approximately 15 sq in. If Tf is taken to be 75°F and CWF3 = 105 GPM, we find that U2 ≈ 1000 Btu/h · ft^2 · °F).

8.2 Film Coefficients (SF_6 Side - No Phase Change)

For turbulent flow inside clean tubes McAdams[5] gives

$$\begin{aligned} U &= 0.023\ Cp \cdot G(Cp \cdot \mu/k)^{-2/3} \cdot (D \cdot G/\mu)^{-0.2} \\ &= 0.023\ Cp^{1/3} \cdot k^{2/3} \cdot \mu^{-0.466} \cdot G^{0.8}/D^{0.2}, \end{aligned} \qquad (8.1)$$

where

Cp = fluid specific heat at constant pressure (Btu/(lb. °F),
k = fluid thermal conductivity (Btu/h · ft^2 · °F),
μ = fluid viscosity (lb/h · ft),
D = inside diameter of tube in feet = 0.04125 ft,
G = mass flow rate per unit area of tube (lb/h · ft^2).

Equation (8.1) applies to either a liquid or gas if the appropriate values of Cp, k and μ are used.

Since the film coefficient does not depend strongly on Cp ($U \propto Cp^{1/3}$), and Cp is fairly constant over the temperature and pressure range of interest, we have used

$$Cp = 0.18 \text{ Btu/(lb} \cdot {}^\circ\text{F)}.$$

Values of the viscosity for both liquid and gaseous SF_6 are given in Section 9, Table III, for temperatures in the range 0-200°F. Values of the thermal conductivity for liquid SF_6 are also given in this table. Recall that these quantities are essentially independent of pressure (see Ref. 7) and that for an ideal gas the thermal conductivity is related to the viscosity by $k \text{ (gas)} \approx Cv \cdot \mu \text{ (gas)}$.

Since UGP and ULP are proportional to the ratio $k^{0.666}/\mu^{0.466}$, and this ratio varies only modestly for both liquid and gaseous SF_6, we find that

$$UGP \approx CGAS \cdot G^{0.8}$$

and

$$ULP \approx CLIQ \cdot G^{0.8},$$

where CGAS and CLIQ are constants associated with the "gas" and "liquid" cases, respectively.

In Fig. 29 we display UG and UL as a function of SMF (note $G \propto SMF$). Recall that UG and UL are the total coefficients (i.e., "water film" and fouling resistances are included). The "width" of the UG curve indicates how much this coefficient increases as the SF_6 temperature increases from 50 to 200°F. We have fitted the lower UG curve and the UL curve to the form

$$U = A + B \cdot SMF + C \cdot SMF^{0.8}$$

and obtained an excellent fit in both cases with

$$UG = -2.745 - 0.003628 \cdot SMF + 0.0516 \cdot SMF^{0.8} \quad (8.2)$$

$$UL = -1.426 - 0.002135 \cdot SMF + 0.0341 \cdot SMF^{0.8} \quad (8.3)$$

We have used the coefficients given by equation (8.2) for HEX1, HEX2 and HEX3 (in the gas region). Observe that at SMF = 50,000 lb/h, UG is equal to 118, which is in close agreement with the value of 113, 115, 117 Btu/h $\cdot$ ft^2 $\cdot$ °F) given by Bos-Hatten for HEX1, HEX2 and HEX3, respectively.

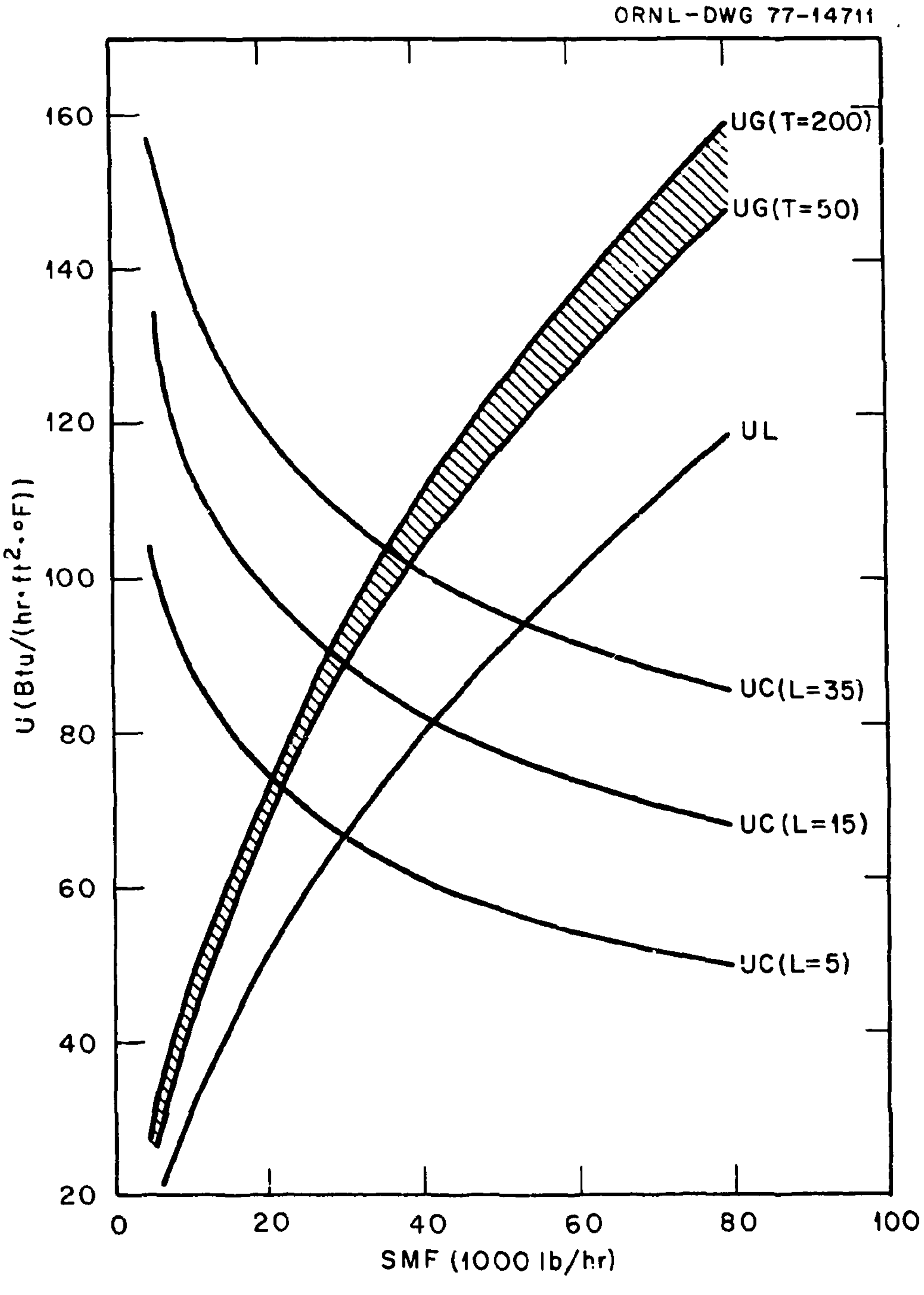

Fig. 29 Heat-transfer coefficients for SF_6 in the gaseous, condensing, and liquid phases as a function of the SF_6 mass flow rate, SMF (1000 lb/h). The curves labeled (L = 5), (L = 15), and (L = 35) were calculated assuming "condensation lengths" of 5, 15, and 35 ft, respectively.

8.3 SF_6 Film Coefficient (Condensation Region)

In the condensation region McAdams gives

$$UCP = 0.76\ (k/D)\ (D^3 \cdot \rho^2 \cdot g/\mu\Gamma)^{1/3}$$

$$= 0.76 \cdot k \cdot \rho^{2/3}\ g^{1/3}/(\mu^{1/3}\ \Gamma^{1/3}),$$

where

$\Gamma = w/(2L) = SMF/(2L \cdot Nt)$,

k = liquid thermal conductivity (Btu/(h • ft^2 • °F),

ρ = liquid density (lb/ft^3),

g = acceleration due to gravity = 4.18×10^8 ft/h^2,

μ = liquid viscosity (lb/h • ft),

w = mass flow per tube (lb/h • tube),

Nt = number of tubes,

SMF = SF_6 mass flow for all tubes (lb/h),

L = condensation length (feet).

All of these quantities are self-explanatory and calculable except for L, the tube length over which condensation takes place. The "average" film thickness is related to 1/L. Here we have a self-consistency problem, since L depends on U, which depends on L (fortunately $U \propto L^{1/3}$ only).

We have calculated the coefficient UC in the condensation region for the values of the condensation length L = 5-35 feet (the total path length in HEX3 is 44 feet). The results for L = 5, 15 and 35 feet are shown in Fig. 29 as a function of SMF. Figure 30 displays UC versus L for various values of SMF. In Fig. 31 we present the curve L versus SMF, which corresponds to UC = 90. Also shown in this figure are L versus SMF curves which actually occurred in transfer computations carried out with UC = 90. We see that the computed L-values are generally larger than the values implied by the assumption UC = 90 (i.e., our assumption was conservative). Since a factor of 2 increase in L produces only a 25% increase in UC, we have chosen to assume that

$$UC = 90\ \text{Btu}/(\text{h} \cdot \text{ft}^2 \cdot {}^\circ\text{F}).$$

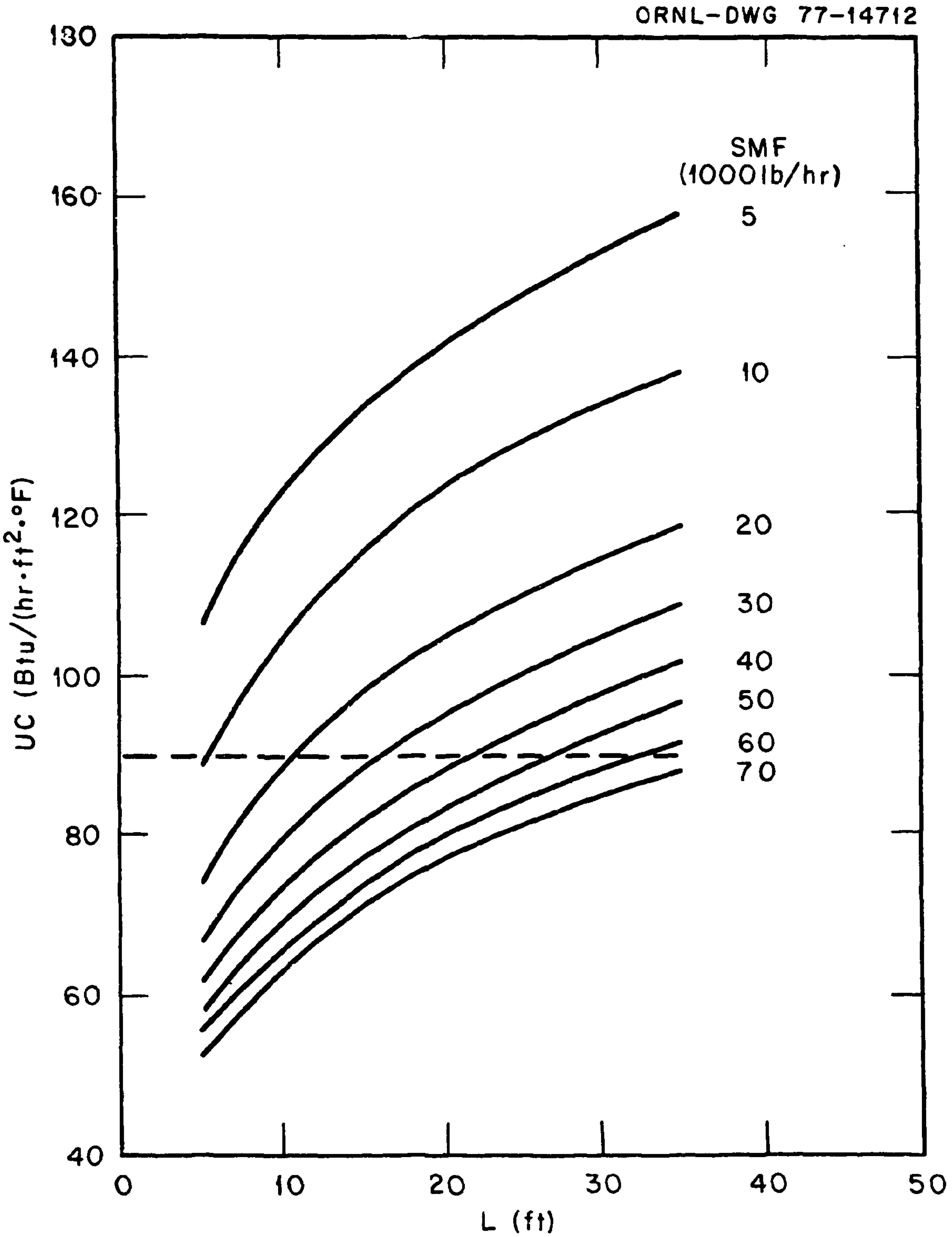

Fig. 30 Heat-transfer coefficient, UC, for SF_6 in the condensing phase as a function of the SF_6 mass flow rate, SMF, and the condensation length, L.

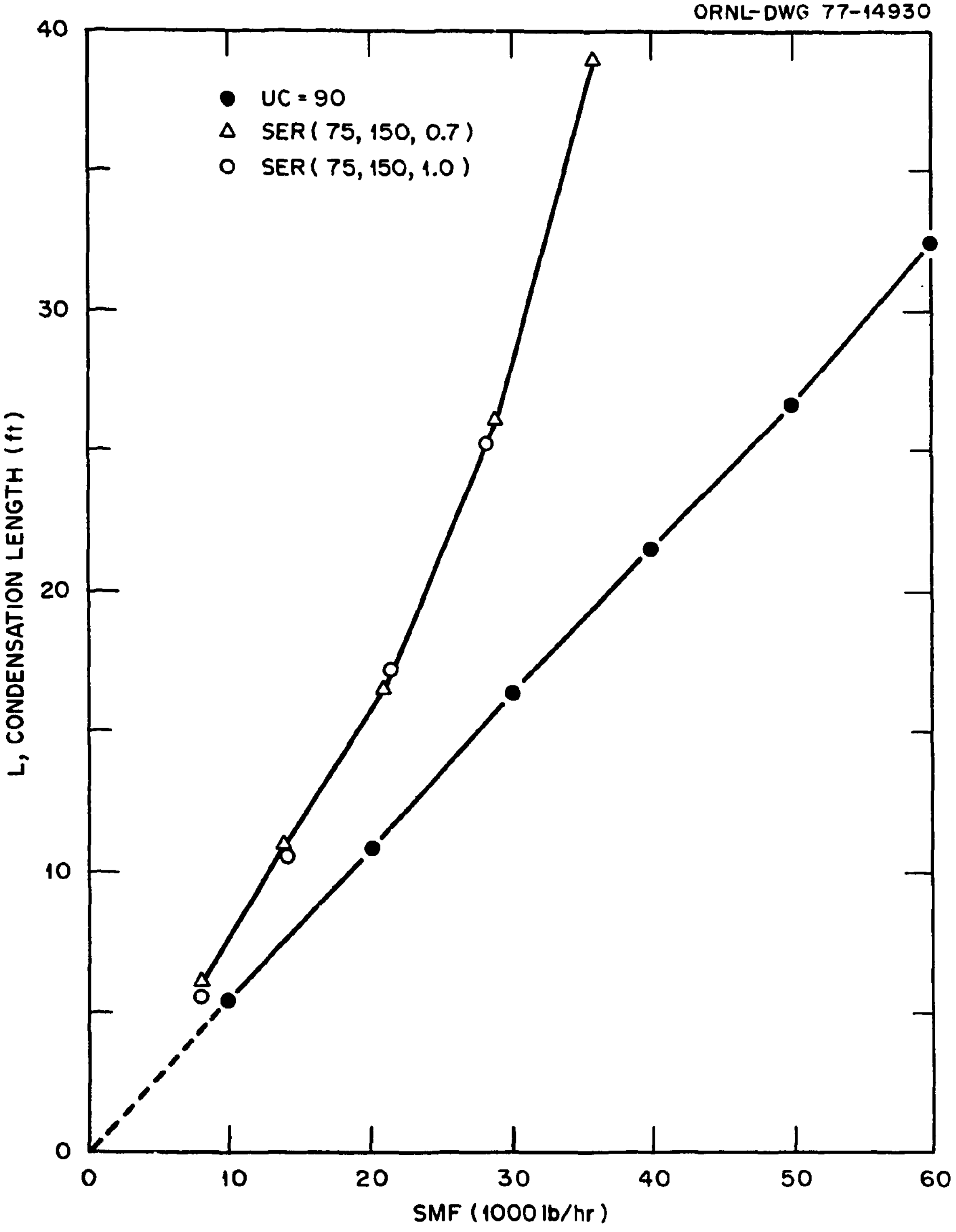

Fig. 31 Calculated condensation lengths (open circles and pluses) are compared to the condensation lengths assumed (solid dots) in carrying out the calculations. (See text.)

We are not attempting to calculate the "true value" of UC, or any other coefficient for that matter, but rather to arrive at a reasonably conservative value. In addition, we have used values of UNORM = 0.6 to 1.0 to investigate the effect produced by a degradation of all heat-transfer coefficients.

9.0 THERMODYNAMIC DATA, FORMULAS AND RELATED QUANTITIES

This section presents a summary of the thermodynamic data and many of the formulas used in this work. Almost all of this material has been taken directly from or derived from information given in Refs. 1 and 2. This summary is given because Refs. 1 and 2 may not be readily available to some readers. Although, the computer calculations usually made use of more extensive tables of enthalpy, entropy, specific heat, etc. than are given here, the tables presented should be sufficient for general reference.

The information given through Table VII is relevant to the general properties of SF_6 and is not dependent on my specific system.

Tables VIII through X give the storage-tank pressure, liquid mass fraction, and liquid volume fraction as a function of storage inventory and temperature.

PHYSICAL PROPERTIES, FORMULAS AND CONVERSION FACTORS

SPECIAL NOTATION USED IN THIS SECTION --------------------------

DEGF - DEGREES FARENHEIT
DEGR - DEGREES RANKIN
DEGC - DEGREES CELSIEUS
DEGK - DEGREES KELVIN
TR - TEMPERATURE IN DEGR
TF - TEMPERATURE IN DEGF
TC - TEMPERATURE IN DEGC
TK - TEMPERATURE IN DEGK
MM - MILIIMETER
M - METER
SQCM - SQUARE CENTIMETER (AREA)
SQFT - SQUARE FOOT (AREA)
SQM - SQUARE METER (AREA)
CUFT - CUBIC FEET (VOLUME)
LBF - POUNDS FORCE

DENLIQ - DENSITY OF LIQUID SF6 IN LB/CUFT
DENVAP - DENSITY OF SATURATED VAPOR IN LB/CUFT
HLIQ - ENTHALPY OF LIQUID SF6 IN BTU/LB
HVAP - ENTHALPY OF SATURATED VAPOR IN BTU/LB
SLIQ - ENTROPY OF LIQUID SF6 IN BTU/(LB*DEGF)
SVAP - ENTROPY OF SATURATED VAPOR IN BTU/(LB*DEGF)

ALSO

A*B MEANS A MULTIPLIED BY B
A**B MEANS A RAISED TO THE POWER, B
EXP(A) MEANS THE EXPONENTIAL FUNCTION OF A
LOG(A) MEANS THE NATURAL LOG OF A

PHYSICAL PROPERTIES ---

MOLECULAR WEIGHT	146.05
SUBLIMATION TEMP (AT 1 ATM)	-83 DEGF
MELTING POINT (AT 32.5 PSIA)	-59.4 DEGF
CRITICAL TEMP	114.17 DEGF
CRITICAL PRESSURE	537.24 PSIA
CRITICAL VOLUME	0.02184 CUFT/LB

```
TEMPERATURE ---------------------------------------------------

TK = TC + 273.15

TR = TF + 459.67

TR = 1.8*TK

PRESSURE ------------------------------------------------------

1 ATM = 14.6960 PSIA
1 ATM = 760.0   MM HG
1 ATM = 101,325  NEWTON/(SQM)  OR PASCAL
1 ATM = 1,013,250 DYNE/(SQCM)

SPECIFIC HEAT OR ENTROPY -------------------------------------

1 BTU/(LB*DEGR) = 1 CAL/(GRAM*DEGK)
1 BTU/(LB*DEGR) = 4.184 JOULE/(GRAM*DEGK)

ENTHALPY -----------------------------------------------------

1 BTU/LB = 0.55556 CAL/GRAM
1 BTU/LB = 2.3244 JOULE/GRAM

THERMAL CONDUCTIVITY -----------------------------------------

1 BTU/(FT*HR*DEGR) = 0.0041338 CAL/(CM*SEC*DEGK)
1 BTU/(FT*HR*DEGR) = 0.017296  JOULE/(CM*SEC*DEGK)

COEFFICIENT OF HEAT TRANSFER ---------------------------------

1 BTU/(SQFT*HR*DEGR) = 1.3562E-4 CAL/(SQCM*SEC*DEGK)
1 BTU/(SQFT*HR*DEGR) = 5.6783E-4 WATT/(SQCM*DEGK)

VISCOSITY ----------------------------------------------------

1 POISE = 1 DYNE/(SQCM)
1 POISE = 0.1 NEWTON*SEC/SQM
1 POISE = 0.01 CENTIPOISE
1 POISE = 5.8014E-7 LBF*HR/SQFT
1 POISE = 0.1 KG/(M*SEC)
```

```
LIQUID DENSITY EQUATION (FROM REF 1) ---------------------------

D = DC + A*X + B*X**2 + C*X**3 + E*X**4

WHERE

X = (1 - T/TC)**0.333333

AND

TC = 573.84
DC = 45.2603
A  = 91.9015
B  = -17.6597
C  = 81.3180
E  =-39.2356

D IS IN CUFT/LB AND T IS IN DEGR


VAPOR PRESSURE EQUATION (FROM REF 1) ---------------------------

P = EXP(A + B/T + C*T + D*T*T + E*(F-T)*LOG(F-T)/T)

WHERE P IS IN PSIA AND T IS IN DEGR

AND

A =  4.4989695
B = -2.5362734E-3
C =  8.8359605E-3
D =  7.4029008E-6
E =  9.4261788E-1
F =  5.8762867E+2


IDEAL HEAT CAPACITY EQUATION (FROM REF 1) ----------------------

CP = A + B*T + C*T**2 + D*T**3 + E/T**2

WHERE

A = -3.7614482
B =  7.6399860E-2
C = -5.5217754E-5
D =  1.4492276E-8
E = -1.0852493E+5

CP (SPECIFIC HEAT AT CONSTANT PRESSURE) IS IN BTU/(LB*MOLE)
T IS IN DEGR

(APPLIES AT TEMPERATURES BETWEEN 360 AND 1300 DEGR)
```

```
EQUATION OF STATE (FROM REF 1) ---------------------------------

P = R*T/(V-BV)  + SUM(TERM(I),I=1,4)

WHERE

TERM(I)  =  (A(I)  +  B(I)*T  +  C(I)*EXP(-K*T/TC))/(V-BV)**(I+1)

R   =  7.3476040E-2
BV  =  5.2439219E-3
K   =  6.8830220
TC  =  573.84

A(1)  = -1.8572400
A(2)  =  2.4588159E-2
A(3)  = -1.5402326E-4
A(4)  = -7.4948301E-9

B(1)  =  1.1341384E-3
B(2)  = -1.1061881E-5
B(3)  =  0.0
B(4)  =  9.2983884E-10

C(1)  = -8.8420987E+1
C(2)  =  1.6808557
C(3)  =  0.0
C(4)  =  -4.7154273E-5

P IS IN PSIA, T IS IN DEGR AND V IS IN CUFT/LB.
```

TABLE I

LATENT HEAT OF VAPORIZATION OF SF6

TF	BTU/LB
20.0	37.8
40.0	34.6
60.0	30.7
80.0	25.8
100.0	18.8
110.0	12.4
113.0	8.6
114.0	5.9
114.15	3.9

TABLE II

SPECIFIC HEAT (AT CONSTANT PRESSURE) (BTU/(LB*DEGF)) OF SUPERHEATED SF6
VAPOR VS TEMPERATURE (DEGF) AND PRESSURE (PSIA)

TF =	40	60	80	100	120	140	160	180	200
PSIA									
0	.151	.155	.160	.164	.168	.171	.175	.178	.182
50	.159	.162	.165	.168	.171	.174	.178	.181	.184
100	.169	.170	.171	.173	.175	.178	.180	.183	.186
150	.184	.180	.179	.179	.180	.182	.184	.186	.188
200	.208	.196	.190	.187	.186	.186	.187	.189	.190
250	0	.220	.205	.179	.194	.192	.192	.192	.193
300	0	0	.229	.211	.203	.199	.197	.196	.197
350	0	0	.280	.233	.216	.207	.203	.201	.200
400	0	0	0	.274	.234	.218	.211	.206	.204
450	0	0	0	.404	.265	.234	.220	.213	.209
500	0	0	0	0	.328	.256	.232	.221	.215
550	0	0	0	0	.558	.292	.248	.230	.222
600	0	0	0	0	0	.357	.270	.242	.229

TABLE III

VISCOSITY AND THERMAL CONDUCTIVITY OF SF6

AS A FUNCTION OF TEMPERATURE

TF	MULIQ	MUGAS	KLIQ	KGAS *
0.0	0.395	0.0140	0.0413	0.0061
20.0	0.357	0.0145	0.0393	0.0063
40.0	0.323	0.0148	0.0375	0.0064
60.0	0.297	0.0152	0.0358	0.0066
80.0	(0.277)	0.0157	(0.0341)	0.0068
86.0		0.0160		0.0070
86.0				0.0081**
100.0	(0.260)	0.0162	(0.0327)	0.0071
120.0		0.0167		0.0073
140.0		0.0172		0.0075
160.0		0.0177		0.0077
180.0		0.0182		0.0079
200.0		0.0186		0.0081

WHERE

MULIQ DENOTES VISCOSITY OF LIQUID IN CENTIPOISE
MUGAS DENOTES VISCOSITY OF GAS IN CENTIPOISE
KLIQ DENOTES THERMAL CONDUCTIVITY OF LIQUID IN UNITS OF BTU/(HR*FT*DEGF)
KGAS DENOTES THERMAL CONDUCTIVITY OF GAS IN UNITS OF BTU/(HR*FT*DEGF)

MULIQ, MUGAS AND KLIQ ARE FROM REFERENCE 1

VALUES IN PARENTHESES ARE DERIVED VIA EXTRAPOLATION

* VALUES OF KGAS ARE ESTIMATED FROM THE ASSUMPTION THAT

KGAS = 0.18*MUGAS

** SINGLE VALUE FROM REFERENCE 1

NOTE-VISCOSITY(LB/(HR*FT))=2.42*VISCOSITY(CENTIPOISE)

TABLE IV

THERMODYNAMIC PROPERTIES OF SATURATED LIQUID AND VAPOR

TF	PSIA	DENLIQ	DENVAP	HLIQ	HVAP	SLIQ	SVAP
30.00	176.46	97.708	6.377	14.141	50.461	.0305	.1047
31.00	179.21	97.461	6.483	14.385	50.543	.0310	.1047
32.00	181.99	97.212	6.591	14.629	50.625	.0315	.1047
33.00	184.80	96.961	6.701	14.875	50.707	.0320	.1047
34.00	187.65	96.709	6.812	15.122	50.787	.0325	.1047
35.00	190.52	96.455	6.925	15.370	50.867	.0330	.1047
36.00	193.43	96.199	7.041	15.619	50.946	.0334	.1047
37.00	196.37	95.941	7.158	15.870	51.025	.0339	.1047
38.00	199.35	95.682	7.276	16.122	51.103	.0344	.1047
39.00	202.35	95.421	7.397	16.375	51.180	.0349	.1047
40.00	205.39	95.158	7.519	16.629	51.257	.0354	.1047
41.00	208.47	94.893	7.644	16.884	51.333	.0359	.1047
42.00	211.57	94.627	7.770	17.140	51.409	.0364	.1047
43.00	214.72	94.358	7.898	17.397	51.485	.0369	.1047
44.00	217.89	94.087	8.028	17.656	51.559	.0374	.1047
45.00	221.10	93.814	8.161	17.915	51.633	.0379	.1047
46.00	224.34	93.539	8.296	18.175	51.706	.0384	.1047
47.00	227.62	93.262	8.433	18.437	51.778	.0389	.1047
48.00	230.94	92.983	8.571	18.699	51.849	.0394	.1047
49.00	234.28	92.701	8.713	18.963	51.919	.0399	.1047
50.00	237.67	92.417	8.857	19.228	51.989	.0405	.1047
51.00	241.09	92.131	9.003	19.494	52.058	.0410	.1047
52.00	244.54	91.842	9.152	19.762	52.126	.0415	.1047
53.00	248.03	91.550	9.303	20.032	52.193	.0420	.1047
54.00	251.56	91.256	9.457	20.302	52.259	.0425	.1047
55.00	255.13	90.960	9.614	20.573	52.324	.0430	.1047
56.00	258.73	90.660	9.774	20.845	52.388	.0435	.1047
57.00	262.37	90.358	9.936	21.119	52.451	.0440	.1047
58.00	266.04	90.053	10.100	21.393	52.513	.0446	.1047
59.00	269.76	89.745	10.268	21.669	52.574	.0451	.1047
60.00	273.51	89.434	10.439	21.946	52.634	.0456	.1046
61.00	277.30	89.120	10.614	22.224	52.693	.0461	.1046
62.00	281.13	88.802	10.791	22.503	52.750	.0466	.1046
63.00	284.99	88.482	10.970	22.783	52.807	.0471	.1046
64.00	288.90	88.157	11.154	23.064	52.862	.0477	.1046
65.00	292.84	87.830	11.342	23.347	52.915	.0482	.1045
66.00	296.83	87.498	11.533	23.631	52.967	.0487	.1045
67.00	300.85	87.163	11.729	23.917	53.018	.0492	.1045
68.00	304.92	86.823	11.926	24.203	53.068	.0498	.1045
69.00	309.03	86.480	12.129	24.491	53.116	.0503	.1044
70.00	313.17	86.133	12.336	24.781	53.162	.0508	.1044
71.00	317.36	85.781	12.548	25.072	53.206	.0514	.1044
72.00	321.59	85.424	12.764	25.364	53.249	.0519	.1043
73.00	325.86	85.063	12.981	25.658	53.291	.0524	.1043
74.00	330.18	84.697	13.207	25.953	53.329	.0530	.1043
75.00	334.54	84.326	13.437	26.250	53.366	.0535	.1042
76.00	338.94	83.949	13.673	26.549	53.401	.0540	.1042
77.00	343.38	83.567	13.914	26.849	53.433	.0546	.1041
78.00	347.87	83.179	14.157	27.150	53.465	.0551	.1041
79.00	352.41	82.785	14.410	27.454	53.493	.0557	.1040

TABLE IV (CONTINUED)

THERMODYNAMIC PROPERTIES OF SATURATED LIQUID AND VAPOR

TF	PSIA	DENLIQ	DENVAP	HLIQ	HVAP	SLIQ	SVAP
80.00	356.99	82.385	14.669	27.759	53.518	.0562	.1040
81.00	361.61	81.978	14.934	28.066	53.541	.0568	.1039
82.00	366.28	81.563	15.206	28.375	53.561	.0573	.1038
83.00	371.00	81.142	15.478	28.684	53.581	.0579	.1037
84.00	375.77	80.713	15.764	28.997	53.595	.0584	.1037
85.00	380.58	80.276	16.059	29.313	53.606	.0590	.1036
86.00	385.44	79.830	16.362	29.631	53.613	.0596	.1035
87.00	390.35	79.374	16.673	29.951	53.617	.0601	.1034
88.00	395.31	78.910	16.984	30.271	53.621	.0607	.1033
89.00	400.32	78.434	17.315	30.597	53.617	.0613	.1032
90.00	405.39	77.948	17.657	30.927	53.608	.0618	.1031
91.00	410.50	77.450	18.010	31.260	53.594	.0624	.1030
92.00	415.67	76.939	18.374	31.596	53.576	.0630	.1029
93.00	420.89	76.415	18.734	31.931	53.560	.0636	.1027
94.00	426.17	75.876	19.126	32.275	53.530	.0642	.1026
95.00	431.50	75.321	19.535	32.625	53.494	.0648	.1024
96.00	436.89	74.749	19.960	32.980	53.451	.0654	.1023
97.00	442.34	74.158	20.401	33.340	53.401	.0660	.1021
98.00	447.85	73.546	20.827	33.698	53.351	.0667	.1019
99.00	453.42	72.911	21.312	34.070	53.285	.0673	.1017
100.00	459.05	72.250	21.823	34.450	53.209	.0679	.1015
101.00	464.74	71.561	22.362	34.837	53.124	.0686	.1012
102.00	470.50	70.840	22.928	35.233	53.027	.0693	.1010
103.00	476.33	70.081	23.433	35.642	52.916	.0700	.1007
104.00	482.23	69.280	24.081	36.062	52.790	.0707	.1004
105.00	488.19	68.430	24.786	36.500	52.645	.0715	.1001
106.00	494.23	67.519	25.505	36.955	52.480	.0722	.0997
107.00	500.35	66.537	26.285	37.434	52.290	.0730	.0993
108.00	506.55	65.465	27.138	37.940	52.071	.0739	.0988
109.00	512.82	64.278	28.085	38.484	51.814	.0748	.0983
110.00	519.18	62.934	29.147	39.077	51.511	.0758	.0977
111.00	525.63	61.365	30.368	39.740	51.146	.0770	.0970
112.00	532.18	59.434	31.814	40.516	50.694	.0783	.0961
113.00	538.81	56.785	33.603	41.510	50.110	.0800	.0950
114.00	545.56	51.332	35.970	43.379	49.314	.0832	.0936

TABLE V

ENTHALPY (BTU/LB) OF SUPERHEATED SF6 VS TEMPERATURE AND PRESSURE

TF =	40	60	80	100	120	140	160	180	200
PSIA									
0	57.622	60.681	63.828	67.060	70.373	73.762	77.226	80.759	84.359
25	57.013	60.143	63.348	66.626	69.978	73.400	76.890	80.447	84.067
50	56.375	59.583	62.850	66.179	69.572	73.029	76.548	80.129	83.771
75	55.702	58.997	62.333	65.718	69.155	72.649	76.199	79.806	83.470
100	54.988	58.382	61.795	65.240	68.726	72.260	75.843	79.477	83.164
125	54.225	57.734	61.233	64.745	68.284	71.860	75.478	79.141	82.853
150	53.401	57.046	60.644	64.230	67.827	71.449	75.104	78.799	82.536
175	52.499	56.310	60.023	63.693	67.354	71.026	74.722	78.450	82.214
200	51.493	55.516	59.365	63.131	66.864	70.591	74.330	78.093	81.886
225	0	54.646	58.663	62.542	66.354	70.141	73.928	77.728	81.553
250	0	53.677	57.908	61.918	65.822	69.676	73.514	77.355	81.213
275	0	0	57.084	61.257	65.266	69.195	73.088	76.973	80.866
300	0	0	56.172	60.548	64.681	68.695	72.650	76.582	80.512
325	0	0	55.114	59.796	64.068	68.176	72.198	76.182	80.151
350	0	0	53.912	58.945	63.409	67.631	71.730	75.770	79.783
375	0	0	0	58.062	62.717	67.064	71.248	75.348	79.407
400	0	0	0	56.944	61.955	66.463	70.745	74.913	79.022
425	0	0	0	55.984	61.151	65.834	70.224	74.466	78.629
450	0	0	0	54.007	60.223	65.159	69.680	74.005	78.226
475	0	0	0	0	59.266	64.448	69.114	73.530	77.814
500	0	0	0	0	57.999	63.671	68.519	73.038	77.392
525	0	0	0	0	57.175	62.845	67.898	72.531	76.960

TABLE VI

ENTROPY (BTU/(LB*DEGF)) OF SUPERHEATED SF6 VS TEMPERATURE AND PRESSURE

TF =	40	60	80	100	120	140	160	180	200
PSIA									
0	.196	.202	.208	.214	.220	.225	.231	.237	.242
25	.142	.148	.154	.160	.166	.172	.178	.183	.189
50	.132	.138	.144	.150	.156	.162	.168	.174	.179
75	.125	.132	.138	.144	.150	.156	.162	.168	.173
100	.120	.127	.133	.140	.146	.152	.158	.163	.169
125	.116	.123	.130	.136	.142	.148	.154	.160	.166
150	.112	.120	.126	.133	.139	.145	.151	.157	.163
175	.109	.116	.123	.130	.136	.143	.149	.155	.160
200	.105	.113	.121	.127	.134	.140	.146	.152	.158
225	0	.110	.118	.125	.132	.138	.144	.150	.156
250	0	.108	.116	.123	.130	.136	.142	.149	.155
275	0	0	.113	.121	.128	.134	.141	.147	.153
300	0	0	.110	.118	.126	.132	.139	.145	.151
325	0	0	.108	.116	.124	.131	.137	.144	.150
350	0	0	.105	.114	.122	.129	.136	.142	.148
375	0	0	0	.112	.120	.127	.134	.141	.147
400	0	0	0	.109	.118	.126	.133	.139	.146
425	0	0	0	.107	.116	.124	.131	.138	.144
450	0	0	0	.103	.114	.122	.130	.137	.143
475	0	0	0	0	.112	.121	.128	.135	.142
500	0	0	0	0	109	.119	.127	.134	.141
525	0	0	0	0	.107	.117	.125	.133	.140

TABLE VII

DENSITY (LB/CUFT) OF SUPERHEATED SF6 VS TEMPERATURE AND PRESSURE

TF = PSIA	40	60	80	100	120	140	160	180	200
0	.000	.000	.000	.000	.000	-0.000	.000	.000	.000
25	.698	.669	.643	.618	.596	.575	.556	.538	.521
50	1.435	1.370	1.311	1.258	1.210	1.166	1.125	1.087	1.052
75	2.217	2.106	2.009	1.923	1.844	1.773	1.709	1.649	1.593
100	3.051	2.884	2.740	2.613	2.501	2.399	2.307	2.223	2.146
125	3.951	3.710	3.508	3.334	3.181	3.045	2.923	2.812	2.710
150	4.929	4.593	4.318	4.087	3.888	3.712	3.556	3.415	3.287
175	6.011	5.544	5.179	4.878	4.623	4.402	4.208	4.034	3.877
200	7.231	6.582	6.097	5.711	5.391	5.118	4.879	4.669	4.480
225	0	7.730	7.086	6.593	6.195	5.861	5.573	5.321	5.097
250	0	9.024	8.161	7.533	7.039	6.634	6.290	5.992	5.730
275	0	0	9.347	8.540	7.931	7.441	7.033	6.683	6.378
300	0	0	10.678	9.630	8.875	8.286	7.804	7.396	7.043
325	0	0	12.250	10.795	9.873	9.168	8.602	8.130	7.726
350	0	0	14.064	12.142	10.960	10.105	9.438	8.890	8.428
375	0	0	0	13.540	12.111	11.087	10.305	9.675	9.148
400	0	0	0	15.382	13.402	12.142	11.218	10.491	9.891
425	0	0	0	17.192	14.774	13.254	12.171	11.334	10.655
450	0	0	0	20.398	16.403	14.469	13.180	12.213	11.444
475	0	0	0	0	18.074	15.759	14.237	13.126	12.256
500	0	0	0	0	20.432	17.204	15.367	14.082	13.097
525	0	0	0	0	21.122	18.751	16.557	15.076	13.965

TABLE VIII

PRESSURE (PSIA) OF SF6 IN STORAGE VS TEMPERATURE AND MASS

	TP =	60	70	80	90	100	110	112	114
PMX	LBS-SF6								
5.	13500.	79.7	81.6	83.4	85.2	87.0	88.8	89.1	89.5
10.	27000.	147.5	151.5	155.4	159.3	163.2	167.1	167.8	168.6
15.	40500.	203.8	210.3	216.7	223.0	229.3	235.5	236.7	237.9
20.	54000.	249.5	258.8	268.0	276.9	285.8	294.6	296.3	298.0
25.	67500.	273.5	297.6	309.8	321.7	333.4	345.0	347.3	349.6
30.	81000.	273.5	313.2	342.9	358.0	372.8	387.4	390.3	393.2
35.	94500.	273.5	313.2	357.0	386.6	404.7	422.6	426.1	429.6
40.	108000.	273.5	313.2	357.0	405.4	429.8	451.1	455.3	459.5
45.	121500.	273.5	313.2	357.0	405.4	448.9	473.7	478.6	483.5
50.	135000.	273.5	313.2	357.0	405.4	459.0	491.1	496.8	502.4
55.	148500.	273.5	313.2	357.0	405.4	459.0	504.1	510.5	516.8
60.	162000.	273.5	313.2	357.0	405.4	459.0	513.2	520.4	527.5
65.	175500.	273.5	313.2	357.0	405.4	459.0	519.2	527.2	535.2
70.	189000.	273.5	313.2	357.0	405.4	459.0	519.2	531.6	540.3
75.	202500.	273.5	313.2	357.0	405.4	459.0	519.2	532.2	543.6
80.	216000.	273.5	313.2	357.0	405.4	459.0	519.2	532.2	545.6
85.	229500.	273.5	313.2	357.0	405.4	459.0	519.2	532.2	545.6
90.	243000.	273.5	313.2	357.0	405.4	459.0	519.2	532.2	545.6
95.	256500.	273.5	313.2	357.0	405.4	459.0	519.2	532.2	545.6
100.	270000.	273.5	313.2	357.0	405.4	459.0	519.2	532.2	545.6

STORAGE VOLUME = 6000 CUBIC FEET

TABLE IX

MASS FRACTION OF LIQUID SF6 IN STORAGE VS TEMPERATURE AND MASS

	TP =	60	70	80	90	100	110	112	114
PMX	LBS-SF6								
5.	13500.	0	0	0	0	0	0	0	0
10.	27000.	0	0	0	0	0	0	0	0
15.	40500.	0	0	0	0	0	0	0	0
20.	54000.	0	0	0	0	0	0	0	0
25.	67500.	8.2	0	0	0	0	0	0	0
30.	81000.	25.7	10.1	0	0	0	0	0	0
35.	94500.	38.2	25.3	8.4	0	0	0	0	0
40.	108000.	47.6	36.7	22.5	2.5	0	0	0	0
45.	121500.	54.9	45.6	33.5	16.6	0	0	0	0
50.	135000.	60.7	52.7	42.3	27.8	4.3	0	0	0
55.	148500.	65.5	58.5	49.6	37.1	16.9	0	0	0
60.	162000.	69.4	63.4	55.6	44.7	27.5	0	0	0
65.	175500.	72.8	67.5	60.6	51.2	36.4	.7	0	0
70.	189000.	75.7	71.0	65.0	56.8	44.0	13.9	0	0
75.	202500.	78.2	74.1	68.8	61.6	50.6	25.4	12.3	0
80.	216000.	80.4	76.7	72.1	65.9	56.4	35.5	25.0	.3
85.	229500.	82.3	79.1	75.0	69.6	61.5	44.3	36.2	19.9
90.	243000.	84.0	81.2	77.6	72.9	66.1	52.2	46.2	37.4
95.	256500.	85.6	83.0	79.9	75.9	70.1	59.3	55.0	53.0
100.	270000.	87.0	84.7	82.0	78.6	73.8	65.6	63.1	67.1

STORAGE VOLUME = 6000 CUBIC FEET

TABLE X

VOLUME FRACTION OF LIQUID SF6 IN STORAGE VS TEMPERATURE AND MASS

	TP =	60	70	80	90	100	110	112	114
PMX	LBS-SF6								
5.	13500.	0	0	0	0	0	0	0	0
10.	27000.	0	0	0	0	0	0	0	0
15.	40500.	0	0	0	0	0	0	0	0
20.	54000.	0	0	0	0	0	0	0	0
25.	67500.	1.0	0	0	0	0	0	0	0
30.	81000.	3.9	1.6	0	0	0	0	0	0
35.	94500.	6.7	4.6	1.6	0	0	0	0	0
40.	108000.	9.6	7.7	4.9	.6	0	0	0	0
45.	121500.	12.4	10.7	8.2	4.3	0	0	0	0
50.	135000.	15.3	13.8	11.6	8.0	1.3	0	0	0
55.	148500.	18.1	16.8	14.9	11.8	5.8	0	0	0
60.	162000.	21.0	19.9	18.2	15.5	10.3	0	0	0
65.	175500.	23.8	22.9	21.5	19.2	14.7	.3	0	0
70.	189000.	26.7	26.0	24.9	23.0	19.2	7.0	0	0
75.	202500.	29.5	29.0	28.2	26.7	23.7	13.6	7.0	0
80.	216000.	32.4	32.1	31.5	30.4	28.1	20.3	15.2	.2
85.	229500.	35.2	35.1	34.8	34.2	32.6	26.9	23.3	14.8
90.	243000.	38.1	38.2	38.1	37.9	37.0	33.6	31.4	29.5
95.	256500.	40.9	41.2	41.5	41.6	41.5	40.3	39.6	44.1
100.	270000.	43.8	44.3	44.8	45.4	46.0	46.9	47.7	58.8

STORAGE VOLUME = 6000 CUBIC FEET

10.0 CONCLUSIONS AND RECOMMENDATIONS

(1) Performance of the system with a CWT of 85°F will be marginal for both SERIES and PARALLEL configurations.

(2) Operation of the system with water cooler than 40 and 60°F is not recommended for the SERIES and PARALLEL configurations, respectively.

(3) With CWT $\leq$ 75°F a degradation of up to 40% in U3 can be tolerated. That is, the system should work if U3 $\geq$ 70 Btu/(h ft.^{2} °F).

(4) We recommend that manually operated valves be provided that facilitate switching from the PARALLEL to the SERIES configuration.

(5) HEX3 should be operated with the maximum CWF consistent with the design, etc., until experience indicates that less cooling can be tolerated.

(6) Records should be kept that will provide an empirical correlation of PSTOR with CWT.

(7) I recommend that the system initially be operated with the following set of parameters:

Configuration = SERIES
CWT = 50-70°F
CWF1 = CWF2 = 40 GPM
CWF3 $\geq$ 150 GPM

(8) Watch everything carefully!

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