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TECHNICAL REPORT
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EVALUATION AND SELECTION OF REFRIGERATION SYSTEMS FOR LUNAR SURFACE AND SPACECRAFT APPLICATIONS

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To

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FOREWORD

This refrigeration system selection report was prepared in preliminary form at the conclusion of an eight month systems study which extended from December 1969 through July 1970. This final version of the report has been restructured, but is essentially the same in content as the preliminary form.

In August 1970 NASA-MSC directed that VMSC work under this contract, which was scheduled to proceed into preliminary design of the recommended system, be put in a hold period. This action was taken to permit development of overall NASA planning for missions/vehicles for which a preliminary design could be prepared. Studies of large 60 man earth orbit space bases prominent at the start of this work were subsequently scaled down to separate space stations with diameters of 33 ft, 22 ft, and eventually 15 ft modular components carried to orbit in the shuttle cargo bay. Modular space stations, experiment modules, and the shuttle orbiter which all possess limited radiator area now appear as the most likely vehicles for which a preliminary design of a refrigeration system could be made. The mechanical vapor compression system recommended herein could be particularly beneficial for use on the emerging space shuttle orbiter during ferry flights, pre-launch operations, orbital mission phases, flyback, and post landing operation.

The primary influencing factor on application of the vapor compression system is the electrical power penalty allocated to the system as is discussed, and analyzed parametrically herein. In retrospect, the lowest power penalty applied herein of 300 lb/kw_e may be somewhat high for the 7 day shuttle mission where reserve capacity of the fuel cell power plants is used intermittently. New guidelines are presently being formulated as well on the fail operational, fail operational, fail safe requirements of the power system which can affect power penalties. In addition, the original cost assumptions made for the overall effectiveness trade studies of five competing refrigeration systems appear to have been significantly low, but it is believed that the overall conclusions and recommendations drawn 14 months ago remain valid.

1.0 SUMMARY

The purpose of this study was to evaluate the various refrigeration machines which could be used to provide heat rejection in Environmental Control Systems (ECS) for lunar surface and spacecraft applications, and to select: (1) the best refrigeration machine for satisfying each individual application, and (2) the best refrigeration machine for satisfying all of the applications. Conventional single phase pumped fluid radiators were considered in the evaluation as a baseline only: the purpose of the study was to select the best refrigeration system and not to choose between conventional radiators and refrigerated heat rejection systems for the specific applications.

The refrigeration machines considered in the study included:

(1) Vapor compression cycle (work-driven)
(2) Vapor adsorption cycle (heat-driven)
(3) Vapor absorption cycle (heat-driven)
(4) Thermoelectric (electrically-driven)
(5) Gas cycle (both reversed Brayton and reversed Stirling cycles) (work driven)
and (6) Steam-Jet (heat-driven)

Various working fluids were considered for each type of refrigeration machine, and a selection of working fluids was made for each machine. A preliminary screening of the types of refrigeration machines was also made, resulting in the following specific refrigeration machines and working fluids being considered in the trade study:

(1) Vapor compression using Refrigerant 12 or 22, depending on system size
(2) Vapor absorption with Refrigerant 22 (R22) and Dimethyl Ether of Tetraethylene Glycol (E-181) as working fluids
(3) Vapor absorption with Lithium Bromide (LiBr) and water as working fluids
(4) Vapor absorption with LiBr/H_2O working fluids and a turbine/compressor to recover work from the absorbent flow stream
(5) Vapor adsorption using water as the refrigerant and type 13X zeolite as the solid adsorbent
(6) Conventional radiator with R21 coolant (included for comparative purposes only)

A computer routine was written which calculates performance of the candidate refrigeration machines under various operating conditions. The optimum weight system for each of the candidate machines in each application can be found with this computer routine. The computer routine determines a specific weight for each machine which includes power penalty, required radiator area penalty, and thermal energy source penalty.

The significant operating parameters are the effective environment heat sink temperature and the required evaporator temperature.

An Effectiveness Function was used in the refrigeration machine selection. The Effectiveness Function considers not only the optimum refrigeration system weight, but also the system volume penalty, maintenance requirements, redundancy requirements, technical risk, and development and fabrication costs. The Effectiveness Function relates these items through trade factors which are dependent on the mission (e.g., launch cost in dollars per pound, crewman time cost in dollars per hour, etc.), and accountable factors which are dependent on the particular refrigeration system (e.g., system weight in pounds, required crewman maintenance in hours, etc.). Both of these factors were estimated for the next generation of spacecraft, and for the specific refrigeration systems.

The selected refrigeration system for each mission considered is given below:

Mission	Refrigeration System
Earth Orbit	Vapor Compression[1]*
Lunar Orbit	Vapor Compression[1]
Lunar Surface Base	Vapor Adsorption Vapor Absorption, $LiBr/H_2O$ with a turbine/compressor
Lunar Surface EVA	Vapor Compression
Transmartian	Vapor Adsorption[2] Vapor Absorption, $LiBr/H_2O$ with a turbine/compressor
Space Shuttle[3]	Vapor Compression[4]

*Notes

(1) The conventional radiator system is superior to vapor compression unless the sink temperature is high or there is a severe shortage of available radiator area.

(2) Vapor compression is a strong third.

(3) Only the orbital portion of the shuttle mission was considered, although the refrigeration system may well be more competitive when considered for all mission phases.

(4) For the assumed area limited situation, the vapor compression system was superior to a conventional integral radiator (even on a specific weight basis), but was inferior to a deployed conventional radiator system.

The recommended order for development of the various types of refrigeration machines is:

(1) Vapor compression because it is the superior system in near-term applications (space shuttle and space station), it is applicable to all missions, and it is the superior system in the greater number of applications.

(2) Water adsorption in zeolite, because it provides a lightweight system in hot environments, it is insensitive to radiator (condenser) coatings degradation (because of the ~200°F operating temperature), it eliminates technical problems in zero-gravity refrigerant/absorbent separation common to most heat-driven refrigeration machines, and it provides a completely independent approach should any problems develop in application of the vapor compression refrigeration machine.

(3) Freon absorption using R-22 and E-181 as working fluids. This system has the advantage that it utilizes low-grade waste heat. This refrigeration machine has the disadvantage that it requires zero-gravity liquid/gas separation, and it has limited applicability.

(4) Water absorption using $LiBr/H_2O$ working fluids with a turbine/compressor to recover energy from the absorbent flow. This is the lightest weight system for a hot environment, however, it involves great development costs and high technical risk.

2.0 INTRODUCTION

Refrigeration systems for manned and unmanned spacecraft have not been used to date for rejecting vehicle ECS loads because of the convenient availability of a low temperature sink for heat rejection by radiation from a conventional pumped fluid radiator system. The low effective sink temperature is achieved by the use of radiator coatings with controlled spectral properties. However, with increasing time in the space environment, the properties of the coatings tend to degrade, thus increasing the effective sink temperature and reducing the heat rejection capacity of the radiator system. In currently operational manned spacecraft, coating degradation has not been a significant problem because of the relatively short mission durations and the availability of relatively large areas on the vehicle exterior which could be used for radiator panels.

The designers of the next generation of spacecraft will, in most instances, have difficulty in obtaining sufficient integral radiator area on the vehicle exterior because the heat rejection requirement of a manned spacecraft tends to grow linearly with the vehicle volume, which grows faster than the vehicle exterior, and because more of the available exterior area will be required for experiments, access doors for maintenance and repair, viewing ports, and docking facilities for other spacecraft. Furthermore, the presence of supporting equipment (such as solar cells, solar absorbers, experiments) and docked vehicles tends to significantly increase the effective sink temperature and thus to reduce the heat rejection per unit of radiator surface area. Thus, designers of future spacecraft will probably be faced with the choice of using deployed radiators (to increase area and to reduce the equivalent heat sink temperature), or of using a refrigeration system to increase the heat rejection temperature of the radiator.

The purpose of this study is to select the best refrigeration system for use in advanced spacecraft from the wide variety of mechanically, thermally, and electrically driven systems which are available. The selection must be made in the context of the space application, which places a premium on weight, volume, power requirements, reliability, and, more recently, on recurring cost. For long-duration spacecraft, maintainability and spare parts requirements are also significant and must be considered. Lastly, the development cost and technical risk of the system must be considered.

Seven unique future missions are identified in the study, and assumptions about the state of space technology in the future time frame were made. It was assumed that large space stations were operational in the time frame of reference, along with reuseable shuttle systems from the earth's surface to earth orbit, and from earth orbit to lunar orbit (See Figure 2-1, and References[1] and [2]).

Potential refrigeration machines were identified, and after a preliminary analysis, were screened to five candidate machines. A computer routine was prepared to optimize each of the candidate refrigeration machines for the selected missions. An effectiveness function which considers all significant parameters was then used to determine the most effacacious refrigeration machine for each mission. A recommendation of the best refrigeration machine for development was made based on the results of the selection for the various missions.

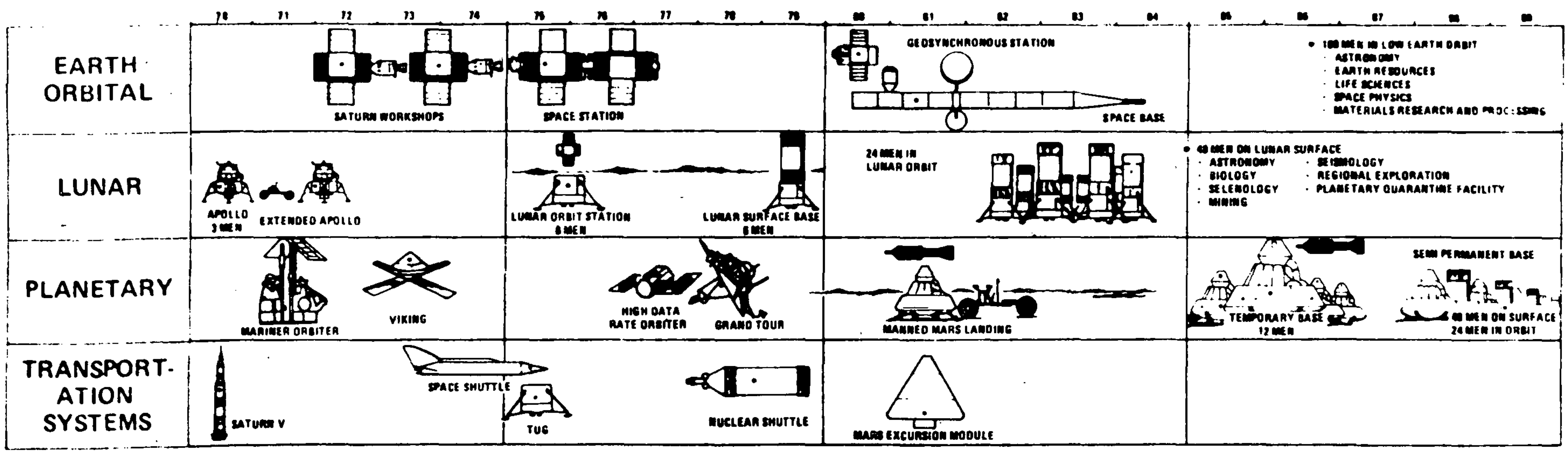


FIGURE 2-1 INTEGRATED SPACE PROGRAM 1970-1990

3.0 REFRIGERATION SYSTEM SELECTION CRITERIA

This section discusses the missions on which a spacecraft refrigeration system might be required, design requirements, system integration factors which affect selection, and selection criteria.

3.1 Missions

There are some potential missions on which refrigeration systems could be beneficial. In general, these include:

(1) Orbital missions involving large vehicles operating in close proximity to each other, so that the "effective" radiation sink temperature is increased.

(2) Missions involving vehicles with large internal heat loads relative to the surface area available for use as radiators.

(3) Missions involving long-term, or repetitive use of a vehicle such that a long-term degradation of radiator surface optical properties results in a high "effective" radiative sink temperature.

(4) Lunar surface missions of more than one month duration, so that the hot lunar "noon" must be endured.

(5) Planetary missions which involve flight trajectories which involve solar distances of less than one A.U. (the "opposition" Mars mission is an example of this).

The use of refrigeration systems on the above missions is obviously not the only solution which would be available, but it is one alternative.

The primary purpose of this study is to identify the type of refrigeration system which is the leading candidate for use in an advanced spacecraft. Since there are no firm committments for future spacecraft development at this time, several possible missions were considered in this study.

(1) Earth Orbit

(2) Lunar Orbit

(3) Lunar Surface Base

(4) Lunar Surface EVA

(5) Mars Exploration

3.2 Design Requirements

In order to establish the range of operating conditions, and other significant characteristics of future manned spacecraft, a literature review was conducted. Tables 3-1 and 3-2 present the results of this review (based on data from References [1] - [7] and the contract statement of work [SOW]). Some of the parameters in Table 3-1 have wide ranges of values by virtue of definition in the SOW. The evaporator temperature range is more likely to be 35 to 45°F than the specified 0 to 65°F. The effective environmental sink temperature lower limit of -78°F is the maximum value which will occur in the most favorable radiator orientation (i.e., always facing away from the sun) in a 270 n.m. orbit (Reference [8]). The upper limit of +100°F is based on an upward facing radiator in the bottom of a shallow crater on the lunar surface at lunar noon; the radiator optical properties are $\epsilon = 0.9$ and $\alpha = 0.3$. In the evaluation of the candidate refrigeration systems, the specific environments for each of the baseline missions are considered. The radiator area penalty is varied parametrically from 0.1 lb/ft^2 to 2.0 lb/ft^2 for each mission. However, in the selection of refrigeration systems radiator weights which are reasonable for the particular application are accorded the most significance. The range of values is considered to demonstrate the impact of radiator weight on refrigeration system selection. The range of power penalties of 300 lb/kw$_e$ to 700 lb/kw$_e$ was specified in the SOW, and this represents a reasonable range of values to be expected in the immediate future. These values reflect a fail-operational, fail-operational, fail-safe reliability criteria. The SOW specified that it was to be assumed that there is no waste heat penalty. The upper limit of 100 lb/kw$_t$ was based on an estimate of the maximum weight of interface equipment necessary to deliver the waste heat to the refrigeration system. Vehicle heat load varies from 3500 BTU/hr for a lunar surface EVA to 150,000 BTU/hr for a large space base. Heat load ranges were selected for each particular mission to demonstrate the effect of heat load on system selection.

Table 3-2 lists the baseline missions considered in the study, and the specific parameters used in refrigeration system sizing for each mission. The basis for the parameters such as heat load, environment sink temperature, power penalty, etc. is given in Appendix B.

3.3 Vehicle Integration

Integration of the heat rejection system into the vehicle can have a significant impact on refrigeration system selection. The ground rules for safety, reliability, maintenance, etc. can also have a strong influence on system selection.

For example, systems currently under development, such as the space shuttle, have a fail-operational, fail-operational, fail-safe reliability criteria for the power system, while the heat rejection system has a fail-operational, fail-safe reliability criteria. Thus a refrigeration system which consumed large amounts of power would be more favorable if it used a dedicated power system rather than the primary system power supply because of the difference in reliability criteria. Decisions on use of a dedicated power supply would have to be made by the program management on each specific vehicle as it is developed.

TABLE 3-1
EXPECTED RANGE OF OPERATING CONDITIONS
FOR REFRIGERATION SYSTEMS

Parameter	Expected Range
Evaporator Temperature	$0^{o}F$ to $65^{o}F$
Effective Environmental Sink Temperature	$-78^{o}F$ to $+ 100^{o}F$
Radiator Area Penalty	0.1 to 2.0 LB/FT^2
Electrical Power Penalty	300 to 700 LB/KW_e
Waste Heat Penalty	0 LB/KW_t
Available Waste Heat Temperature	300 to $400^{o}F$
Vehicle Heat Load	3,500 to 150,000 BTU/HR

TABLE 3-2

BASELINE MISSION DEFINITION

BASELINE MISSIONS	LAUNCH COSTS WEIGHT $/LB	LAUNCH COSTS VOLUME $/CU.FT.	VEHICLE HEAT LOAD BTU/HR	VEHICLE HEAT LOAD KW_t	DESIGN ENVIRONMENT SINK TEMP. °F	RADIATOR LIMITATION (AREA OR ORIENTATION)	POWER PENALTY LB/KW_e	POWER SYSTEM COST $\$/W_e$	CREW SIZE	NO. OF VEHICLES BUILT
Earth Orbit										
Space Shuttle Orbiter (Based on 100 launches)	5,000	25,000	35,000	10.2	20 (Integral Area) -78 (Deployed Radiator)	900 Sq. Ft. Inside Cargo Bay Doors	300	50	4	10
Space Station (Solar Cell Power) With High Cost Shuttle Launch Vehicle	200	472	120,000	35	20 (Integral Area) -78 (Deployed Radiator)	2500 Sq. Ft. Integral area	500	50-150	12	10
Space Base or Station With Saturn V Launch	1,000	7,500	260,000 ± 120,000	76 ± 41	0	None	300 500 700	50	60(Base)	10
Low Cost Shuttle Launch	50	236								
High Cost Shuttle Launch	200	472								
Lunar Orbit Station (Medium Cost Shuttle to Earth Orbit and Nuclear Tug to Lunar Orbit)	500	2,500	120,000	35	100	None	700	50	12	5
Lunar Base (Carried to Lunar Surface by Reusable Vehicle from Above Lunar Orbit)	1,000	5,000	50,000	14.6	100	None	700	50	6	5
Lunar Surface EVA (Originating from Lunar Base Described Above)	1,000	20,000 (Based on PLSS* Package Density of 20 lb/cu. ft.)	3,500	1	100	None	700	50	1	10
Mars Excursion (Based on Shuttle and Tug on Earth Orbit, Orbital Assembly and Launch)	8,000	40,000	120,000	35	100	None	700	50	12	3

*Apollo Portable Life Support System

In earth orbit the power penalty for sun-side operation is much lower for solar cell power systems than for full orbit operation. This is because batteries are used to store energy produced on the sun-side of the orbit for use on the dark-side. The battery storage represents a significant portion of the power penalty for solar power systems. It would be possible to design a refrigeration system to cool down the structure of the vehicle on the sun-side and to absorb heat on the dark-side of the orbit. Flight experience with Apollo suggests that is a plausible approach. Fusible heat sink devices could be applied to this same concept. A similar concept would be to use a refrigeration system (with its large power demand) on the sun-side of the orbit (where power is available at a low cost), and to use a radiator system (with its low power requirement) on the dark side of the orbit (where the effective sink temperature is low and power cost is high). The power penalty for complete orbit operation for solar cells is in the range of 300-350 lb/Kw_e at a minimum, while the power penalty for sun-side only operation is in the range of 125-150 lb/Kw_e.

All of the above concepts for reducing power penalty are vehicle or perhaps even mission oriented, and are difficult to consider in a general evaluation of refrigeration systems for use on future spacecraft.

System integration factors may also have an impact on the radiator area penalty. If the spacecraft has a meteoroid shield, the radiator can be integrated with the meteoroid shield at the cost of only the tubes which are welded to the shield plus the associated headers, return lines, valves, etc. For a conventional single-phase pumped fluid system using a halocarbon (e.g. Refrigerant 21) as the coolant fluid, the required tube diameters are very small (about 1/8 inch in diameter) so that the radiator area weight penalty may be as low as 0.1 lb/ft^2. In another case where there is limited external area available for use as radiator surface, then deployed radiator panels would be required. The space shuttle orbiter is a candidate for this type of system. The radiator weight penalty may be as much as 2 lb/ft^2 in such a case.

The availability of vehicle external area for use as radiator surface is obviously very highly dependent upon the individual spacecraft and on the vehicle heat rejection requirements. Refrigeration systems offer another dimension in heat rejection system selection since the required radiator area can be reduced at the expense of power penalty.

The use of waste heat to drive an absorption refrigeration machine offers some interesting system integration possibilities. Waste heat used to drive the absorption refrigeration machine reduces the amount of heat which must be rejected through the power system radiators. However, for this to result in a savings in power system radiator size and weight, the refrigeration system would have to accept a fixed amount of waste heat from the power system at all times: This is probably not a particularly stringent demand for an absorption refrigeration machine, however, this might necessitate a change in the reliability criteria for the refrigeration system heat rejection equipment from the life support system level (fail-operational, fail-safe) to the power system level (fail-operational, fail-operational, fail safe).

Any discrepancy between the temperature level required to drive the refrigeration system and the optimum heat rejection level of the power system would result in a penalty to the power system. The optimum heat rejection temperature for a Brayton Cycle Nuclear Power System would be about 275°F as compared to the availability range of 300-400°F given in the SOW. The optimum temperature required for driving the refrigeration system would be in the range of 350°F-600°F.

Integration of a refrigeration system into a spacecraft has an interesting effect on the required radiator area. For a work-driven machine, the coefficient of performance (COP) is

$$COP = \text{heat load } (q_L)/\text{work input}(w)$$

The required heat rejection is then

$$q_r = q_L + w$$

and, substituting COP for w from the above equation for COP,

$$q_r = q_L + \frac{q_L}{COP} = q_L \left(1 + \frac{1}{COP}\right)$$

For a mechanical vapor compression machine, the COP may be much larger than unity; in most instances for spacecraft it will be 3 or 4 as discussed in the Appendices. Thus, the amount of heat to be rejected is 25%-30% greater for the refrigeration system than for a single-phase fluid radiator. However, the heat rejection temperature is much higher. In a typical case a refrigeration system with a COP of 3 and a heat rejection temperature of 110°F operating in a 0°F heat sink environment would yield a heat rejection per unit radiator area of

$$q_r/\eta A^* = \sigma \epsilon \left[T_r^4 - T_s^4\right] = (0.173 \times 10^{-8})\ (0.9)\ \left[(570)^4 - (460)^4\right]$$

$$q_r/\eta A = 95 \text{ BTU/hr ft}^2$$

$$q_r = q_L \left(1 + \frac{1}{COP}\right) = q_L \left(1 + \frac{1}{3}\right) = 1.33\ q_L$$

$$\therefore q_L/\eta A = \frac{95 \text{ BTU/hr ft}^2}{1.33} = 71 \text{ BTU/hr ft}^2$$

For the pumped fluid system with an average radiating temperature of 65°F, which would correspond to a radiator inlet temperature of around 90°F and a radiator outlet temperature of 40°F, the heat rejection per unit area would be

$$\frac{q_r}{\eta A} = \frac{q_L}{\eta A} = \sigma \epsilon \left(T_r^4 - T_s^4\right)$$

$$\frac{q_L}{\eta A} = (0.173 \times 10^{-8})\ (0.9)\ \left[(525)^4 - (460)^4\right]$$

$$\frac{q_L}{\eta A} = 48 \text{ BTU/hr ft}^2$$

*ηA is radiator fin effectiveness times radiator area

Thus the pumped fluid radiator system would require about 50% more radiator area than would the mechanical vapor compression refrigeration system.

For a heat-driven refrigeration machine the situation is somewhat different. The system COP is

$$\text{COP} = \frac{\text{heat load } (q_L)}{\text{heat supplied } (q_H)}$$

The required heat rejection is

$$qr = q_L + q_H$$

substituting from the equation for COP for q_H

$$qr = q_L + \frac{q_L}{\text{COP}} = q_L \left(1 + \frac{1}{\text{COP}}\right)$$

For absorption refrigeration machines, the COP is normally around 1/2 (though it can be higher; the best commercial units have a COP of about 0.7). Thus the heat rejection for an absorption unit is:

$$qr = q_L \left(1 + \frac{1}{1/2}\right) = 3q_L$$

This indicates that an absorption unit operating with a heat rejection temperature of 110°F would need to reject about 2.25 times as much heat as the vapor compression system, as indicated below

$$\frac{(qr)_{abs}}{(qr)_{V.C.}} = \frac{3q_L}{1.33\ q_L} = 2.25$$

Thus the absorption system requires 2.25 times as much radiator area as the vapor compression system. In order to reduce the radiator area requirement to the same level as the mechanical vapor compression system, the heat rejection per unit area would have to be

$$\left[\frac{qr}{\eta A}\right]_{abs} = 2.25 \left[\frac{qr}{\eta A}\right]_{V.C.} = 2.25\ (95) = 215 \text{ BTU/hr ft}^2$$

The required radiating temperature would be approximately 205°F. Of course, increasing the condenser temperature to this level would have the effect of reducing the COP, so an even higher temperature would be required.

The above gives an example of the types of system integration considerations which are difficult to evaluate in a general study, but which could have a profound influence on a specific system selection study.

3.4 Selection Criteria

The purpose of this study is to identify the order of priority which should be attached to development of candidate refrigeration systems for potential application in unspecified future spacecraft. The approach taken is to rank the systems according to desirability for various types of systems for potential future missions which can be identified (as given in Table 3-2). These rankings can then be evaluated to identify the single system which is most suitable for early development. This selection is based on applicability to the various missions, flexibility, and cost. Applicability to the early vehicles, and the probability that the vehicle will actually be developed are significant parameters in the overall evaluation.

Table 3-3 lists the parameters which are considered in the refrigeration system ranking. An attempt was made to quantify these factors to arrive at a realistic ranking of the candidate systems (which are described in Section 4.0). Appendix A presents the effectiveness function which was used to rank the competing systems. Appendix B presents the application of the effectiveness function to the refrigeration system comparison, and also presents the supporting data used in the quantifying of the trade parameters. In Appendix B, an attempt is made to convert all trade parameters to a common basis: in this case 1970 dollars were the common base. This allows competing systems to be compared on a single basis. (The basis is dollars above a common base, not absolute dollars. This is explained in Appendix B.)

Aside from the inherent difficulties in quantifying some of the parameters, there is also the problem that dollars in one year are not really interchangeable with dollars in another year. This is because Federal Agencies work from budgets that are established on a year-to-year basis by an external force and the allocated funding level cannot be exceeded in a given fiscal year, and must be spent within that year (however, there is some slack in this part of the system). The result is that large peaks are not tolerated (except in very unusual circumstances) and large valleys are not allowed to occur. For example, this policy hinders the spending of large amounts of development money in one time period to vastly reduce recurring costs in the future. Thus, dollars in one year are not directly tradeable with dollars in a future year. (It should be realized that this problem impacts the trade factors in the effectiveness function even if some basis other than dollars were used. This is because the problem is inherent in trading development costs and recurring costs, and is not related to the way in which the trade is made).

After the ranking of systems for each mission is complete, then the ranking for immediate development priority is made. This ranking takes into account subjective factors such as flexibility, applicability to near-term programs, and applicability to all missions.

TABLE 3-3

SELECTION CRITERIA

Factor	Degree of Difficulty in Quantitizing	Comments
Weight	Straight Forward	Traditionally used in industry, sometimes as the only selection criteria (in well developed technology, weight usually reflects other parameters such as cost and volume). In aerospace, weight becomes less important as launch system capability improves.
Volume	Straight Forward	Importance depends on the launch system; for systems with characteristically high packaging, density and volume can be considered to be reflected by weight. However, when widely different systems are competing, volume must be considered.
Development Cost	Difficult	Usually becomes more important as a particular technology matures. Capital availability is a significant factor in establishing significance on a particular program.
Recurring Cost	Difficult	Usually becomes more important as number of units to be produced increases; significance is related to current and future capital availability.
Maintainability and Repairability	Very Difficult	Becomes more important as life of system increases.
Reliability	Very Difficult	Becomes less significant as technology matures. Systems become more redundant and repairable, so failures can be tolerated.
Redundancy and Spares	Very Difficult	Redundancy is required with long life systems because cost factors tend to result in lower component reliability. Spares requirement can have strong impact on system volume and weight.
Technical Risk	Very Difficult	High technical risk becomes less acceptable as technology matures because dramatic breakthroughs are not required to achieve mission objectives, and development costs become more significant (since available capital can be used profitably to exploit current programs.)
System Integration	Very Difficult	As technology matures the trend is toward modular components which can be easily integrated into any system. System integration is important (and expensive) when vehicle capabilities are limited, and missions are of the "one-shot" variety. Subsystems developed for general future use should not be dependent on use of another particular type of sub-system (e.g. a specific type of power system).
Flexibility	Subjective	Very important when system is not developed for a specific application. Can be a significant factor in reducing costs as technology matures (since it can reduce both development and recurring costs).
Availability	Subjective	Can significantly reduce development and recurring costs of a vehicle if subsystems which are available can be used. Reduces technical risk in subsystem development if similar components exist.

4.0 CANDIDATE SYSTEMS

This section presents the various types of refrigeration systems which could be used in spacecraft, the preliminary screening of the systems, and a description of the systems selected for detailed evaluation.

4.1 Types of Refrigeration Systems

Table 4-1 presents the various refrigeration processes which were considered in this study. Mechanical vapor compression is the most widely used of all refrigeration processes, and is now encountered almost daily in the life of most Americans in the form of air conditioning or food preservation. Vapor absorption processes comprise the bulk of the remaining refrigeration equipment now in service. Other processes are either commercially non-competitive, or have special applications such as the use of the reversed Brayton Cycle for aircraft cooling (mechanical vapor compression is also used in this service), or the Stirling Cycle to provide cryogenic cooling. Thermoelectric systems are used primarily in military electronics cooling. These refrigeration processes are discussed and analyzed in some detail in preliminary reports submitted in this work (References [9] - [11]). Table 4-1 presents some of the conclusions drawn in those reports.

4.2 Candidate Refrigeration Systems

The refrigeration systems selected for detailed consideration are (from Table 4-1):

(1) Mechanical vapor compression using a halocarbon as the refrigerant.

(2) Vapor absorption using R22 as the refrigerant and E181 as the absorbent.

(3) Vapor absorption using water as the refrigerant and lithium bromide (LiBr) as the absorbent.

(4) Vapor absorption using water/LiBr, and augmented by a turbine/compressor (this system is described in Section 4.3.2).

(5) Vapor adsorption using water as the refrigerant and synthetic zeolite as the adsorbent.

Reasons for the selection of these systems (drawn from the advantages and disadvantages indicated on Table 4-1) are as follows:

(1) Mechanical vapor compression was chosen because it is the most widely used system in terrestrial applications, it has a characteristically high COP, it is compact, and it is relatively simple to control. The large power requirement is the only severe drawback.

(2) Vapor absorption systems offer the advantage that they require very little shaft work, and are primarily driven by low grade energy in

TABLE 4-1

CANDIDATE REFRIGERATION SYSTEMS

Technique	Common Working Fluids	How Driven	General Comments	Candidates	Advantages	Disadvantages
Mechanical Vapor Compression	Halocarbons:	Mechanical Work	The most widely used refrigeration technique capacities go from fractional ton units (e.g. home refrigerators) to thousands of tons (commercial building refrigeration). Applications include automobiles, commercial aircraft, homes, buildings, ships, industrial processes, etc.	Selected as a candidate (both reciprocating and centrifugal compressors)	1. Compact Equipment 2. High COP (relatively low heat rejection required) 3. Flexible 4. Well-Developed Technology	1. Large Power Requirement 2. Rotating Machinery
	R12 R22	Primarily used with reciprocating compressor				
	R11	Centrifugal compressor				
Vapor Absorption	NH_3/Water Water/LiBr R22/E181	Heat	The second most widely used refrigeration system. Primarily water/LiBr in units with capacity over 100 tons (about 1/3 of market). NH_3/water systems have a small fraction of the small tonnage (3-5) market. Frequently used where a large heat source is available (as in a steam generating plant).	Water/LiBr and R22/E181	1. Heat Driven 2. Moderately Well-Developed Technology	1. Low COP (relatively large heat rejection requirement 2. Zero-g Separator is imperative 3. NH_3 is good thermodynamically, but is toxic, flammable, and corrosive
Vapor Adsorption	Sulfur Dioxide/ Silica Gel Water/Zeolite	Heat	SO_2/silica gel systems were used on refrigerated railway cars in the 1920's and 1930's. Water/zeolite has never been used, but is attractive for use in a vacuum environment. No equipment is presently commercially available.	Water Zeolite Selected	1. Heat Driven 2. No Zero-g Separation 3. No Rotating Machinery	1. Low COP 2. Technology not Developed
Steam-Jet Vapor Compression	Water (other fluids could be used)	Heat	Used primarily in the 1920's and 1930's in steam generating plants.	Not Selected	1. Heat Driven 2. No Zero-g Separation 3. No Rotating Machinery	1. Very Low COP 2. Essentially cannot operate with condenser temperature above 120°F
Gas Cycle	Air (Reversed Brayton)	Centrifugal Compressor	Used in military and commercial aircraft. Operates in an open cycle manner using gas turbine compressor bleed air.	Not Selected	1. Well-Developed Technology 2. No Zero-g Separation	1. Very Low COP 2. Very High Power Requirement 3. Rotating Machinery (normally used in open-cycles in aircraft where power penalty is low and no heat rejection equipment is needed)
	Helium (Stirling)	Reciprocating compressor	Widely used in military applications to provide cryogenic temperatures to infrared detectors. Some commercial use in refrigerated trucks.	Not Selected	1. Reasonable COP 2. Compact	1. Technology not Well Developed 2. Rotating Machinery
Vortex Tube (Also called Hilsh-Ranque Tube)	Air	Compressed Air Supply	Used in portable cooling for men in protective clothing in industrial applications. Also used to cool electronics in small spacecraft on the Launch Pad.	Not Selected	1. Compact 2. Lightweight 3. No Moving Parts Where Cooling Effect is Produced	1. Very Low COP 2. Not Effective in Closed Cycle Operations
Thermoelectric (Peltier Effect)	None: Uses semiconductor materials	Direct conversion of electricity	Mostly used to provide spot cooling at very low loads in military electronic systems. Some experimental use in cooling of large military equipment vans.	Not Selected	1. Direct Conversion of Electricity 2. Compact 3. No Moving Parts	1. Extremely Low COP 2. Large Power Requirement

the form of heat. This feature is potentially attractive for integration with nuclear power sources through utilization of waste heat. There were three sets of fluids chosen for evaluation: R22/E181 offers good performance with fluids which are suitable for use in regions of the spacecraft which do not always have close thermal control; LiBr/water offers excellent performance, however, the water may have to be protected from freezing, and there is a possibility of solid salt forming in return and delivery lines if temperature limits are not maintained; LiBr/water with a turbine/compressor offers higher condenser temperatures coupled with a good COP. The main disadvantages of these systems are the need for a zero-gravity separator (for separation of the refrigerant and the absorbent), a relatively large heat rejection requirement, and possibly some control difficulties.

(3) The vapor adsorption system was selected because it is heat-driven, and it does not require zero-gravity separation of fluids. The use of water as the refrigerant permits relatively efficient operation with very high condenser temperatures. Disadvantages of this approach are a large fixed weight for the adsorbent bed, and a COP of less than 1/2 which results in a large heat rejection requirement (this is off-set to a large degree as far as radiator area is concerned because very high condenser temperatures can be used). The sulfur dioxide/silica gel system which was once used commercially was not considered because sulfur dioxide was considered to be too toxic for use in a manned spacecraft.

(4) Steam-jet vapor compressor has some attractive features from the standpoint of use of waste heat for refrigeration, particularly if the spacecraft used a steam power plant. This system also has the advantage of requiring no rotating machinery, and no zero-g separation of fluids. One of the primary disadvantages of this system in terrestrial applications is the high vacuum required in the evaporator (about 0.12 psia); atmospheric leakage into the evaporator dramatically reduces performance. However, in the hard vacuum of space, leakage into the evaporator would not be a factor. The inherently low COP of the steam jet refrigeration system results in a very high heat rejection requirement, which is a severe liability. As the condenser temperature is raised, the amount of steam which must be employed increases dramatically, reducing the COP, increasing the heat rejection requirement, and increasing the size of the return pump, piping, etc. These factors combine to make the system non-competitive for condenser temperatures above 120°F (with a source steam temperature of 350°F). This system was not chosen for further consideration because it has no clear-cut advantage over the adsorption system, and it has a much more limited operating range.

(5) Gas cycle refrigeration has found wide application in the air-conditioning of high performance commercial and military aircraft, where the reversed Brayton Cycle with air as the working fluid is used. The reversed Brayton Cycle is uniquely suited to air-conditioning of aircraft, since the use of air as the working fluid eliminates the need for heat exchanger equipment between the working fluid and the air (as would be required with a mechanical vapor compression machine) and permits open-cycle operation. The inherent low COP of the reversed Brayton Cycle is off-set because there is no large heat rejection required (due to open-cycle operation) and the power penalty associated with the use of jet engine compressor bleed air is small. The result of these

factors is that the Brayton Cycle air-conditioning unit for use with jet aircraft is very compact and lightweight. Except for the use of air as the working fluid, these advantages are not applicable to spacecraft, where the very low COP imposes a large power penalty, and the low COP coupled with the impracticability of open-cycle operation result in a large heat rejection requirement. The only significant advantage of the Brayton Cycle for spacecraft refrigeration is the lack of a requirement for "zero-g" fluid separation. This advantage is more than off-set by the disadvantages.

(6) Gas cycle refrigeration employing the Stirling Cycle, which is comprised of two constant volume and two isothermal processes, is usually not competitive with vapor compression machines in terrestrial applications. The Stirling Cycle is able to achieve cryogenic temperatures in a single stage, which gives it an advantage over compound machines in producing cryogenic temperatures. This has led to widespread use of the Stirling Cycle for provision of cooling at cryogenic temperatures to infrared detectors in military applications. For provision of spacecraft ECS heat rejection, the Stirling Cycle would be similar to vapor compression units except that COP would be somewhat lower, and the inability to use a direct condensing radiator, which the vapor cycle may be able to do, offers a lower potential than the vapor cycle. For this reason the Stirling Cycle was not selected for further application.

(7) Vortex tubes are primarily used where a high pressure supply of air is readily available, and the unit is required to be compact, lightweight, and inexpensive. They are used in terrestrial applications such as cooling men wearing protective clothing in industrial facilities and in ground cooling of small, unmanned spacecraft prior to launch. The extremely low COP of this refrigeration technique makes it unsuitable for use in manned spacecraft where closed cycle operation is mandatory.

(8) Thermoelectric devices have inherently low COP's when operating between the temperature levels normally encountered in a spacecraft ECS, i.e., from a low temperature side of around 40°F, to a high temperature side of 90-110°F. The advantages of direct conversion of electricity to cooling effect in a compact unit with no moving parts is more than off-set by the high penalties for power and for heat rejection equipment associated with spacecraft.

4.3 Description of Refrigeration Systems Selected for Detailed Evaluation

This section describes the candidate refrigeration systems in more detail. It also describes a typical spacecraft radiator system which has been included in the detailed evaluation as a baseline. (In some environments the radiators are not effective because of the radiation environment; the lunar surface at noon is an example of this. For example, a radiator in a crater with a 10 to 1 diameter to depth ratio, with optical properties of $\alpha = 0.3$ and $\epsilon = 0.9$, would have a radiation equilibrium temperature of 100°F at lunar noon).

4.3.1 Mechanical Vapor Compression

The vapor compression refrigeration system has been extensively used in commercial applications, however, it has not been used to date for manned spacecraft applications. The system operates with a high COP even in severe thermal environments and, therefore, minimizes radiator area requirements. However, since shaft work is mandatory in its operation and this energy source is expensive in space, the system equivalent weight is highly sensitive to the power penalty factor.

Ideal Cycle

The ideal vapor compression cycle (Figure 4-1) has a COP near that of the Reversed Carnot Cycle. In the vapor compression cycle, heat is transferred during constant pressure vaporization and constant pressure condensation of a refrigerant fluid. With working fluids which have relatively high latent heats of vaporization the system flowrate is much lower than in the conventional radiator, thus effecting some weight savings due to the smaller heat exchanger surface required, etc. In the ideal cycle, vapor from the evaporator is compressed in an isentropic process and then, in the condenser, heat is rejected at constant pressure from the fluid to the environment, as the vapor is condensed. The refrigerant leaving the condenser is then in a saturated liquid state. To bring the pressure back down to that of the evaporator, the fluid passes through an expansion valve in an irreversible process. This process flashes a portion of the fluid and reduces the temperature of the two-phase mixture to the evaporator temperature. Finally, this mixture of liquid and vapor refrigerant is returned to the evaporator where the liquid is vaporized at constant pressure by heat removed from the refrigerated space.

Actual System

An actual cycle will differ from the ideal cycle in several ways. Losses due to fluid flow friction and heat transfer to or from the surroundings occur. Irreversibilities exist in the compressor as well as in the necessary heat transfer processes. Additionally, compressors operate most effectively with fluid in a vapor phase. Introduction of a mixture of liquid and vapor may cause mechanical problems and decrease the efficiency of the compressor. To insure that the pipe friction losses do not cause the condensation of any vapor between the evaporator and compressor, the saturated vapor is usually superheated a few degrees in the evaporator. A more economical system (in terms of system weight) incorporates an intercooler between the fluid leaving the evaporator and that exiting the condenser.

The intercooler is essentially a heat exchanger that causes heat transfer from the high temperature, high pressure saturated liquid to the lower temperature saturated vapor. The result is that superheated vapor enters the compressor as a single phase fluid and subcooled liquid enters the expansion valve. One disadvantage of superheating the compressor suction vapor too much is that the increased specific volume of the vapor increases the work of compression required. But, subcooling the liquid allows an increase in the amount of heat that can be added to the fluid through

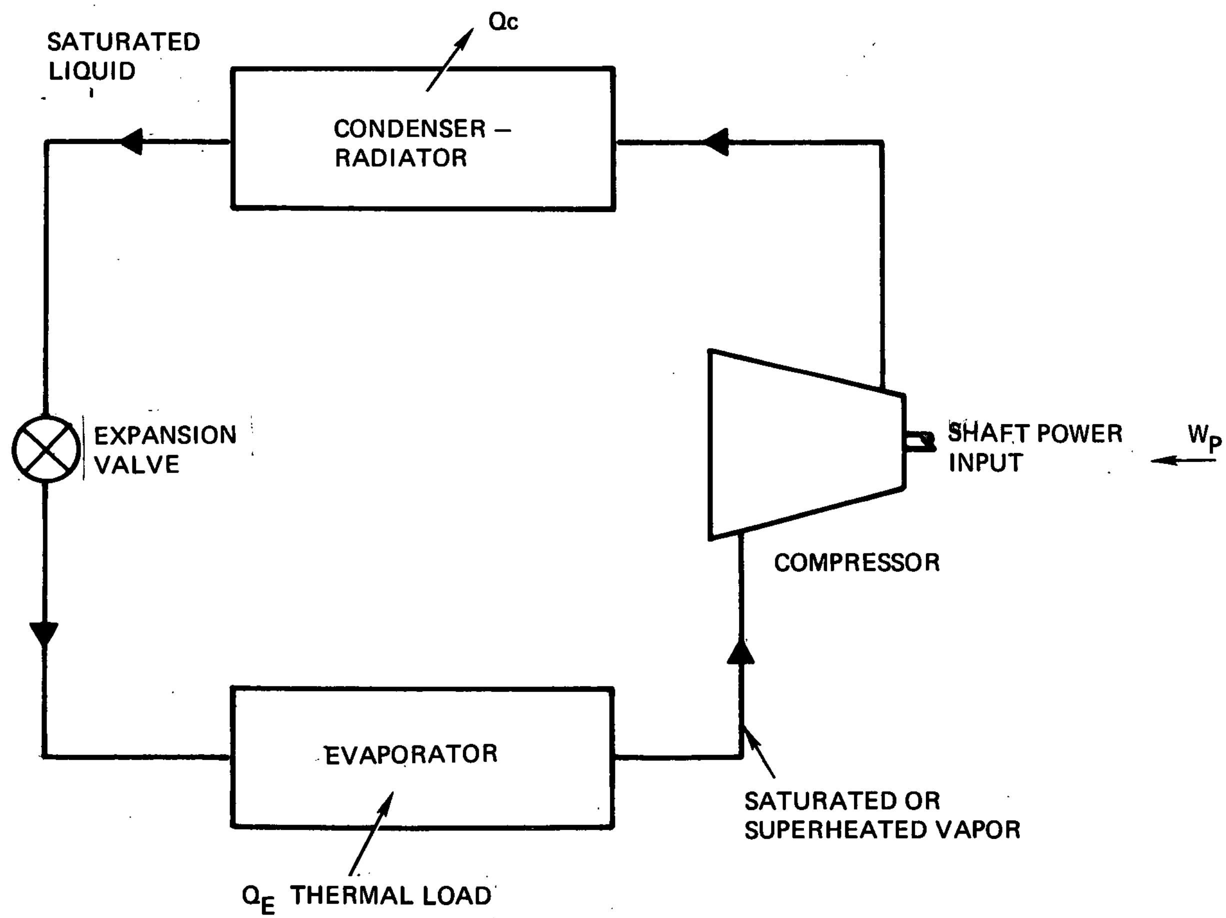


FIGURE 4-1 SIMPLE VAPOR COMPRESSION SYSTEM

the evaporator. The net effect of intercooling in this manner is that single-phase vapor flow to the compressor is insured without superheating in the evaporator, with no change in the total refrigeration effect (with R12 there is a slight improvement according to Carrier in Reference [12]).

System Operating Limits

The evaporation and condensation pressures (and thus temperatures) used in the vapor compression cycle have a strong influence on the magnitude of the COP. Decreases in the evaporator pressure or increases in the condenser pressure result in a reduction of the COP. This is because the ideal cycle COP is reduced and the work required to increase the pressure of the working fluid in the compressor increases, and the net amount of heat that can be absorbed per unit of refrigerant flow in the evaporator is decreased. Another consideration is that low evaporator pressures mean larger refrigerant specific volumes, and consequently larger equipment. Increased pressures reduce the specific volume and the size of the equipment, but require that the system components be designed to withstand the high internal pressure (thus resulting in increased weight).

The permissible operating limits of the system involve the physical properties of the refrigerant used. Since a change of phase is involved, the temperature levels of the evaporator and the condenser must be below the critical value. For best results, it has been found that pressures and temperatures near the critical should be avoided. A secondary limit may be imposed by the freezing temperature of the working fluid, when it is possible for the system to encounter that temperature even when the system is non-operational.

The particular mission requirements under consideration involve operation between limits of 30-50°F up to 140-160°F. These limits are well within the operating limits of several commonly used refrigerants. The primary limiting factor in fluid selection is the power penalty associated with production of the necessary shaft work to drive the compressor.

Control of the system to minimize power consumption will be of prime importance in the spacecraft application; because of the inherently large power penalty. A simple on/off control system could be used; however, this might increase the size of the spacecraft power system. An integrated control system such as is used in commercial buildings to reduce the power demand, could be used to advantage. Figure 4-2 shows a mechanical vapor compression system with a complete control system to reduce power consumption during part-load operation. The system shown has a centrifugal compressor with variable guide vanes and a variable speed drive to achieve efficient operation at reduced load; the expansion valve is controlled to allow operation with varying condenser pressure, and there is a central controller to interpret signals received from sensors and select the optimum operating conditions.

In order to optimize the performance of a mechanical vapor compression system, a direct radiator/condenser should be used, rather than using a separate single-phase cooling system which receives the heat load from the refrigerant

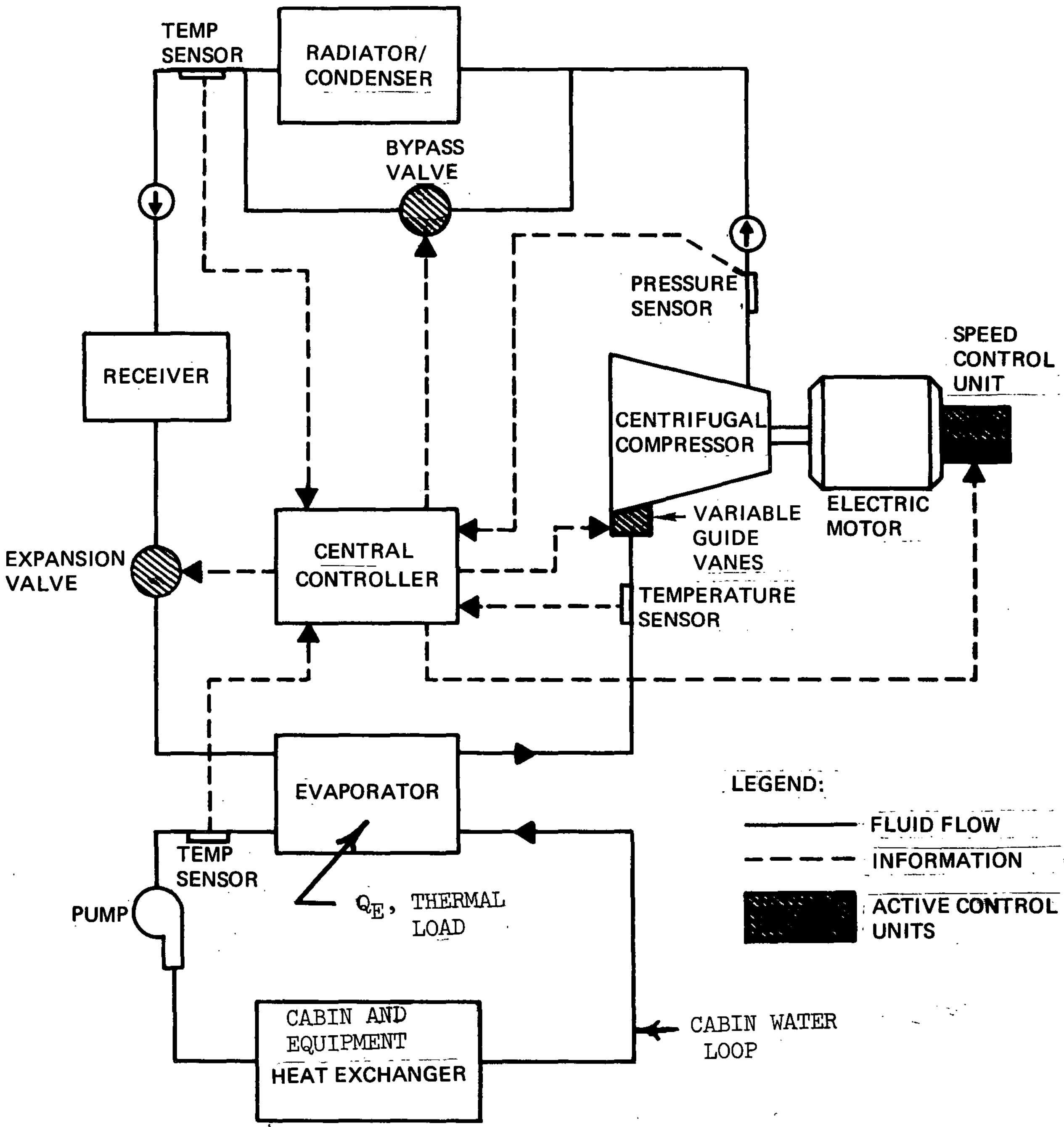


FIGURE 4-2 VAPOR COMPRESSION SYSTEM CONTROL

in the condenser, and which transfers the heat load to space through a conventional pumped fluid radiator (termed herein a secondary radiator). Not only does the direct condenser/radiator reduce weight because of the obvious savings from elimination of the pump, fluid, HX, etc. for the secondary radiator, but also the direct condenser/radiator requires less radiating area than does the secondary radiator. This is because the heat transfer irreversibilities in the intercooler HX between the refrigeration loop and the secondary radiator loop result in a lower heat rejection temperature from the single-phase radiator than the condenser/radiator would have. In addition, the uniform tube temperature of the condenser/radiator results in a higher radiating fin effectiveness than can be attained with the secondary radiator.

Potential Problem Areas

The high state of development of mechanical vapor compression refrigeration systems in commercial operations provides a low technical risk associated with development of this system for spacecraft application.

The primary technical problem areas are related to operation of the system in a zero-gravity environment (which is required for some potential missions). Development of a compressor which can operate in zero-g is a necessity; basically this is a lubrication problem. It is also vitally important to perfect the condenser/radiator because of the gains in thermodynamic cycle efficiency and the reduction in system hardware associated with use of this concept. Finally, a control system which insures optimum operating efficiency for all heat load and thermal environment variations is needed.

4.3.2 Vapor Absorption Systems

The basic elements of a vapor absorption system are a condenser, evaporator, absorber, pump, regenerator and generator, as shown in Figure 4-3. The evaporator and condenser function in the same manner as in the vapor compression cycle; however, the compressor is replaced by the absorber, liquid pump and generator. Since the vapor pressure of the refrigerant (solute) is reduced in the presence of the solvent, the refrigerant is condensed at low pressure and intermediate temperature in the absorber forming a strong (in solute) liquid solution. This strong solution is supplied to a pump to increase its pressure and then passes through a regenerative heat exchanger to the generator. In the generator, the solution temperature is increased by energy supplied by an external heat source thus vaporizing part of the solute refrigerant and producing a weak liquid solution. Since a zero or low gravity field is assumed, a two-phase flow of refrigerant and weak liquid solution will leave the generator and the two phases must be separated in a zero "g" separator. (In terrestrial applications, the phase separation is accomplished by density gradient, which is a gravity dependent process.) The weak solution is throttled down to the evaporator pressure and is returned to the absorber through the regenerative heat exchanger. The refrigerant vapor is condensed, throttled, and then evaporated by the low temperature heat load as in the vapor compression system. The vapor absorption system has a theoretical COP, which is given by the following equation:

$$(COP)_{Theoretical} \leq \left[\frac{T_{Evaporator}}{(T_{Absorber} - T_{Evaporator})}\right] \times \left[\frac{(T_{Generator} - T_{Condenser})}{T_{Generator}}\right]$$

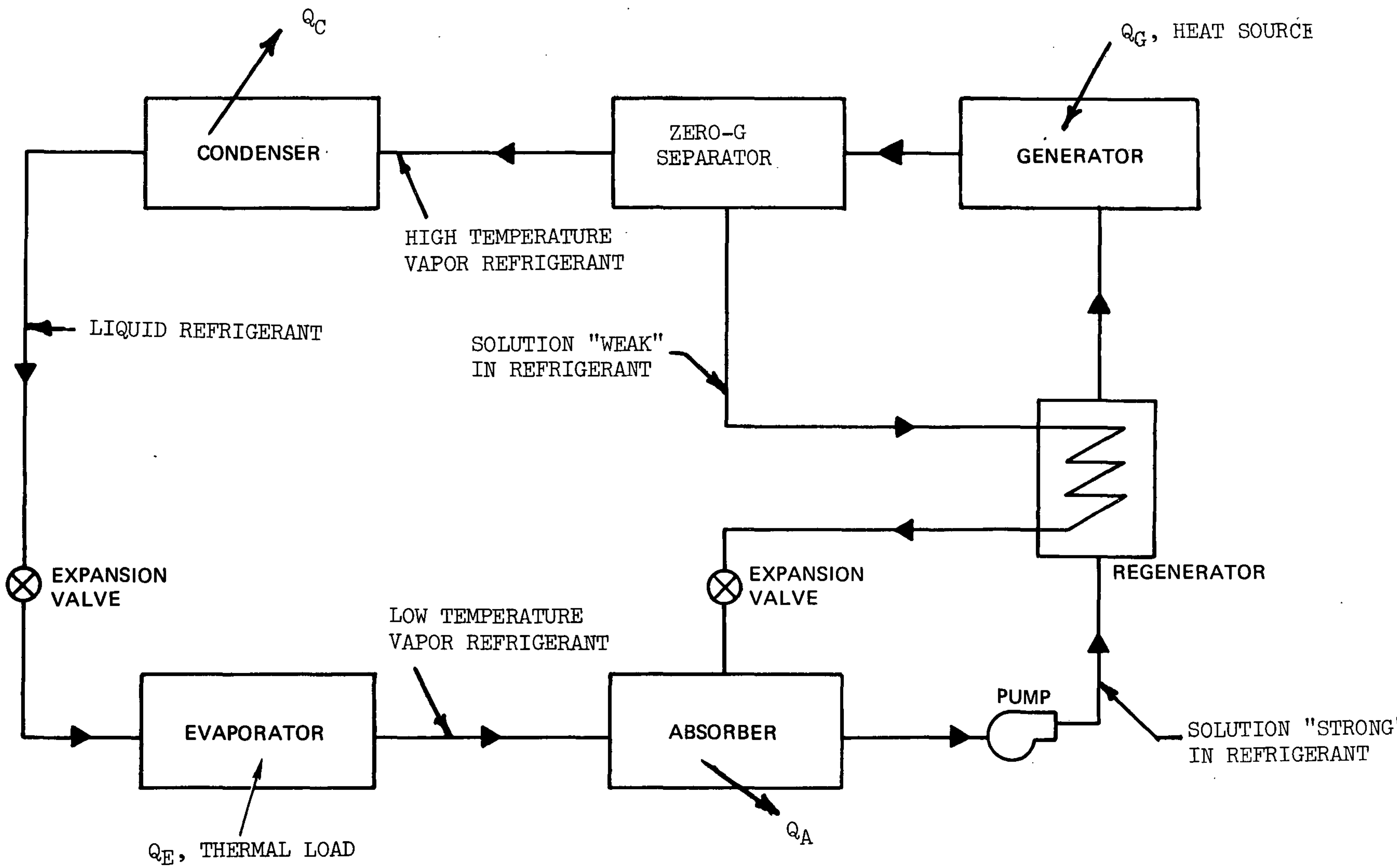


FIGURE 4-3 **SIMPLE ABSORPTION REFRIGERATION SYSTEM**

The first term on the right of the equation represents the ideal or Carnot COP of the refrigeration portion of the cycle, and the second term represents the efficiency of the power portion of the cycle. It is obvious that the efficiency of the power portion of the cycle will be enhanced as the generator temperature is increased. The generator temperature is limited by the thermodynamic properties of combinations of refrigerants and absorbers, and also possibly in this case, by the available waste heat temperature. By the above equation, the theoretical COP for an absorption system operating with an evaporator temperature of 40°F, a generator temperature of 350°F, and an ambient temperature of 100°F would be:

$$(COP)_{theoretical} = \left[\frac{500}{560-500}\right]\left[\frac{810-560}{810}\right] = 2.57$$

In an actual case, however, the working fluid properties coupled with the heat transfer irreversabilities in the system reduce the Carnot COP and power portions of the cycle by about 50% each. Thus the actual COP will be about 1/4 of the theoretical, or about 0.64 in this case. The COP of the best commercial absorption refrigeration systems is about 0.7.

The choice of working fluids for the vapor absorption cycle is governed primarily by three properties of the refrigerant and the absorbent: (1) the critical temperature of the refrigerant (solute), which should be much larger than the condenser temperature (which is dictated by the external environment); (2) the vapor pressure of the absorbent, which should be small; and (3) the quantity of the refrigerant easily absorbed in the absorbent, which should be large, implying a large heat release accompanying the formation of the solution of refrigerant and absorbent (i.e., a large deviation from Raoult's Law). In addition to these primary considerations, it is also desirable in the spacecraft application for the refrigerant to have a large heat of vaporization, a low freezing point, and good chemical stability at high temperatures. The absorption systems selected for detailed evaluation are discussed below:

R22/E181 Absorption System

A vapor absorption system with working fluids of R22 and dimethyl ether of tetraethylene glycol (E-181) is shown schematically in Figure 4-4. Included in the system are a "zero-g" separator and a "zero-g" rectifier, which are needed in weightless environment applications. Because of the large weight penalty incurred with secondary radiator loops, the absorber, intercooler and condenser are considered to be integral parts of direct radiators. The system shown in Figure 4-4 uses a hydraulic motor rather than an expansion valve such as is used in more conventional systems. The hydraulic motor is used to remove work from the high-pressure weak solution as the weak solution is expanded to the absorber pressure. The work obtained from the hydraulic motor is used to drive the system pump, thus reducing the electric power needed to drive the pump. This is significant for applications involving a large electrical power weight penalty, such as is usually the case in spacecraft.

Certain physical properties of the working fluids limit the range of temperature over which the system can be applied. Refrigerant 22 is not

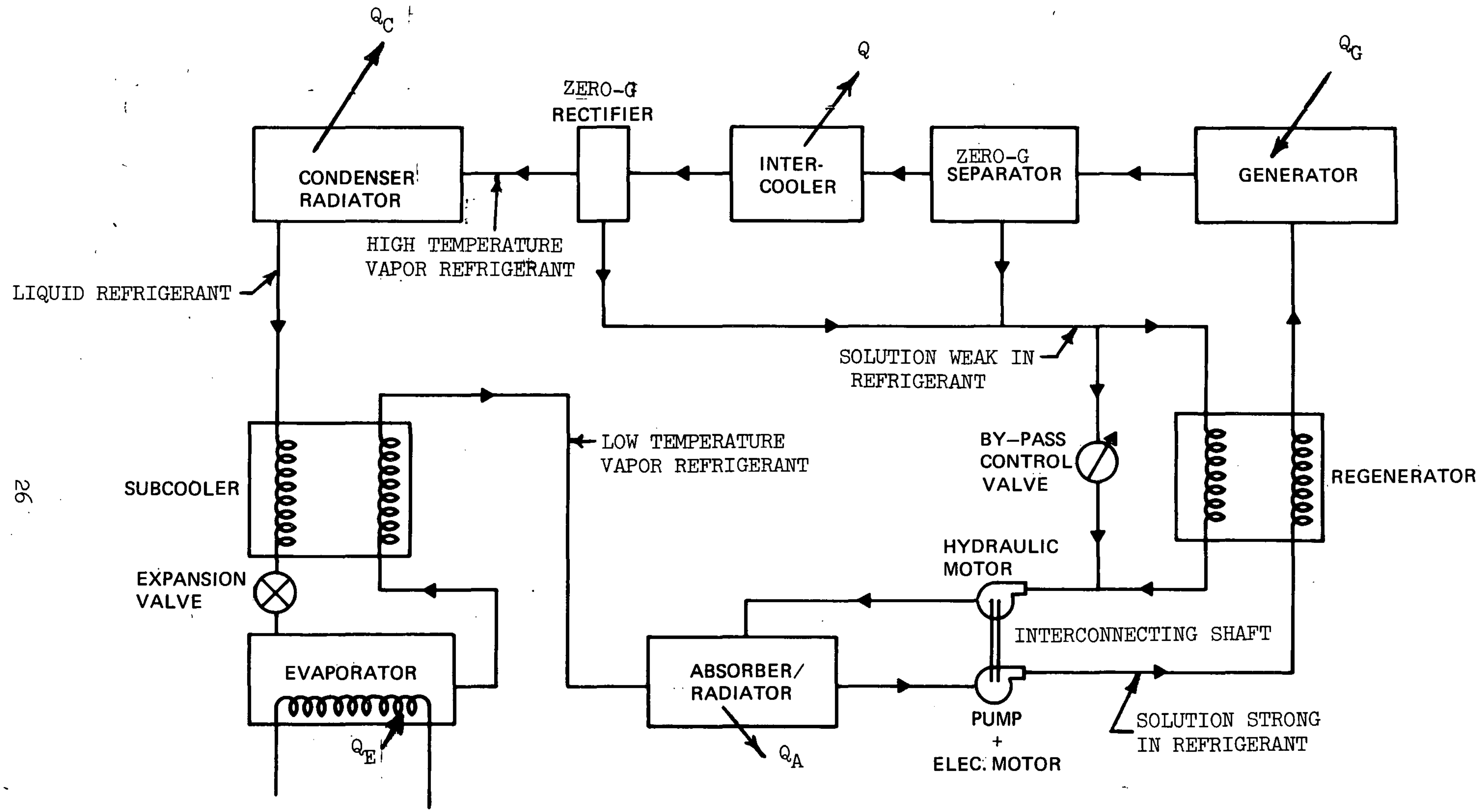


FIGURE 4-4 R22/E181 ABSORPTION REFRIGERATION SYSTEM

recommended for service involving continuous exposure to temperatures in excess of 350°F in the presence of aluminum and oil. Thus, the temperature in the generator and the COP are limited by the materials which are available for use. The system was optimized as a function of a refrigerant concentration, generator temperature, and condenser and absorber temperatures for operating conditions representative of earth orbit and lunar surface applications. The following system parameters were used in the optimization process:

Absorber Concentration:	61% (Mole fraction of R-22)
Generator Concentration:	35.4% (mole fraction of R22)
Generator Temperature:	350°F

The most significant technical problems with the R22/E181 absorption system involve the development of an efficient zero-g separator, for separation of the refrigerant vapor and the liquid absorbent, and the development of "direct" condenser and absorber radiators which will operate with a wide range of heat loads and external environments. (The use of a secondary radiator system greatly increases system weight because of the additional hardware required, and because of the attendant loss of thermodynamic efficiency). The required control system for this system will be much more complicated than that shown in Figure 4-4, and development of this required control system will also be a significant technical problem area.

Lithium Bromide - Water Absorption Cycles

In commercial applications, the simple lithium bromide-water absorption cycle, which is shown in Figure 4-5 is employed. This system, with the inclusion of a zero "g" separator, may be used without serious modification in earth orbital applications. However, for more severe thermal environments such as on the lunar surface, the physical properties (in particular, the precipitation of the solid salt LiBr) limit the useful range of the cycle. The basic cycle was modified, as shown in Figure 4-6, to incorporate a compressor to raise the pressure of the refrigerant from the low values required with a 40° to 55°F evaporator to a level compatible with a high temperature absorber. A steam driven turbine was added to power the compressor, and a high temperature mixer/absorber provides condensation of the turbine steam, as well as dilution of the return lithium bromide, to prevent precipitation of the LiBr in the regenerators. Thus, the system is actually an absorption refrigerator with a self-contained power plant and vapor compression refrigerator. As a consequence, the entire system operates at high temperatures except in the evaporator, and the mixer/absorber for the turbine/compressor cycle functions at a temperature which is about equal to that of the generator in the conventional cycle.

Both the simple and the turbine/compressor LiBr/H_2O systems employ secondary loops to transfer the heat to the radiator(s). While the radiating temperature is decreased thus increasing the required radiator area, secondary loops are considered mandatory to prevent precipitation of solid LiBr and the possibility of freezing of the refrigerant (water). The radiator is greatly simplified as a result (the heat load maximum-minimum ratio is only 2:1; except when the system is inoperative) but the system is complicated by the

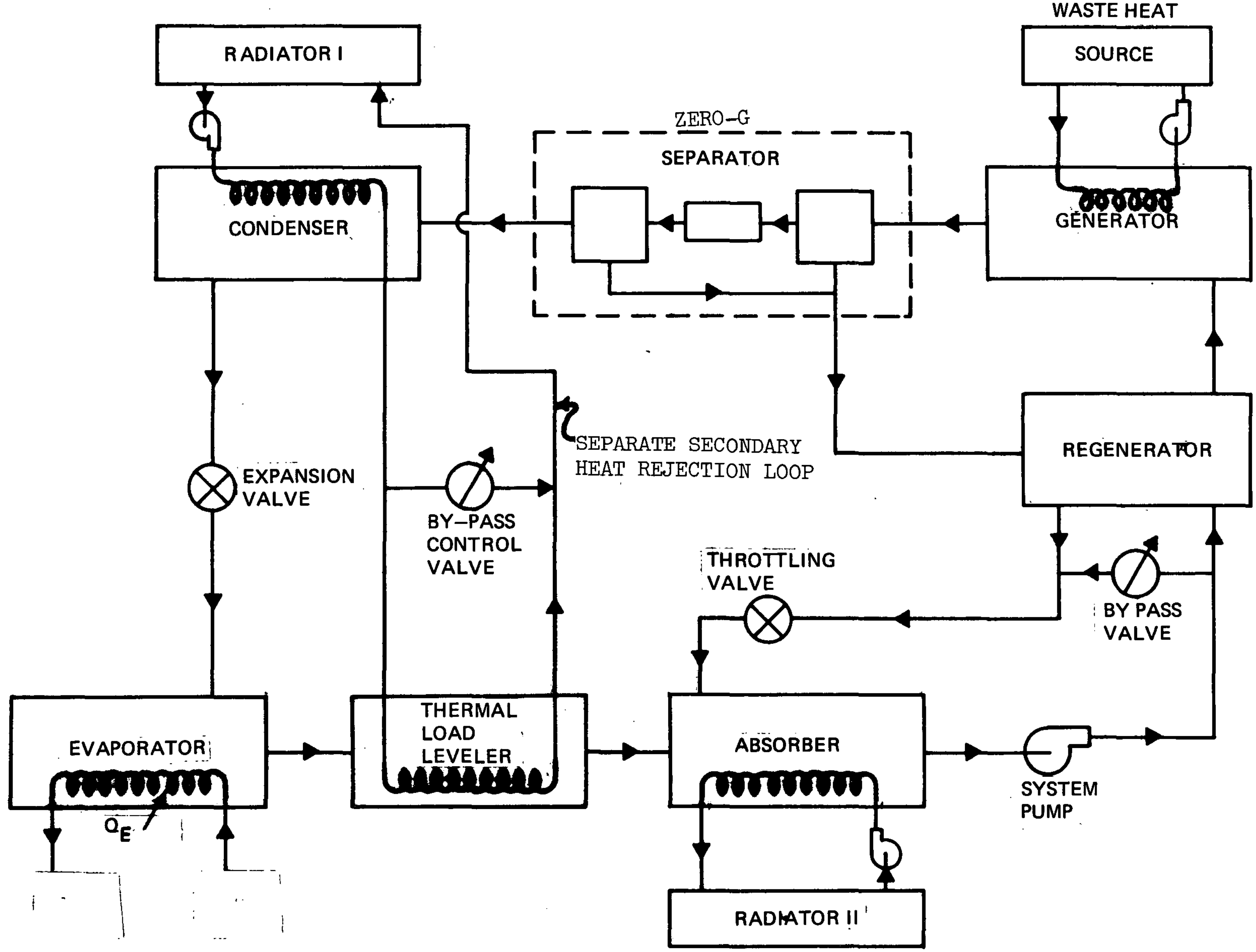


FIGURE 4-5 LIBR/H_2O SIMPLE ABSORPTION SYSTEM

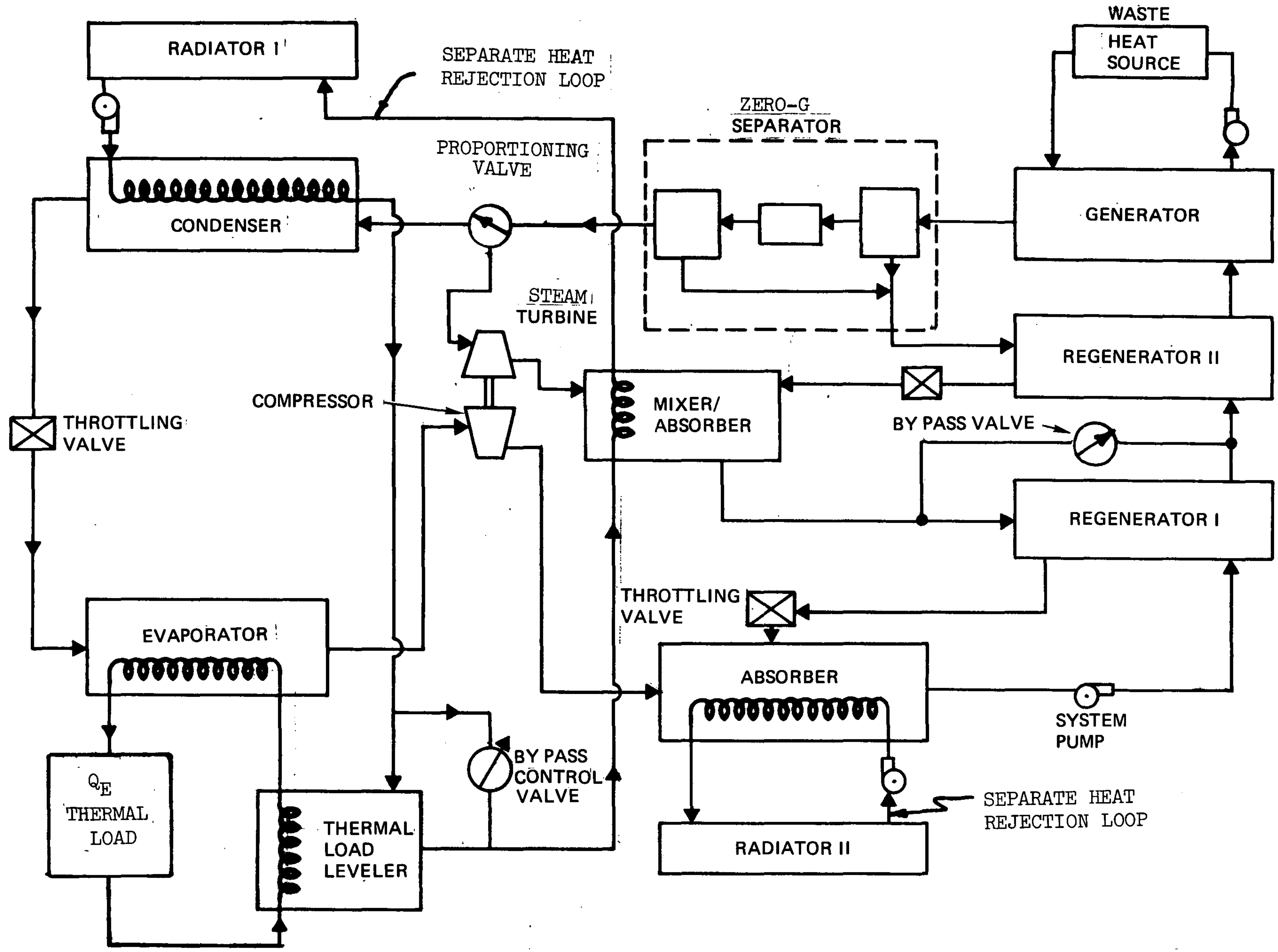


FIGURE 4-6. **LIBR/H_2O ABSORPTION SYSTEM WITH TURBINE/COMPRESSOR**

necessity of extra components, controls, and a condenser and an absorber which are integrated into the secondary loop heat exchanger.

The low load control problem of both LiBr/H_2O absorption systems is alleviated by use of the secondary loop to vaporize the liquid refrigerant which passes through the evaporator. This procedure can be accomplished by a heat exchanger thermal load leveler on the refrigerant line as shown in Figure 4-5, or by a heat exchanger on the thermal load side to control the inlet temperature to the evaporator as shown in Figure 4-6. This type of control is highly desirable for spacecraft operation since heat load varies with vehicle environment, internal activity, power profiles, etc. while this refrigeration system operates at a single thermal load at all times. However, the system can use additional waste heat to make up the decrement in system heat load during low load conditions.

The use of secondary radiator fluid loops for heat load control greatly simplifies the control of the heat rejection system; however, the refrigerator thermodynamic efficiency is greatly reduced. Indeed, for hot environments the simple LiBr system is so limited in range ($T_{[Condenser]} - T_{[Evaporator]}$) that the system is excessively large. The turbine/compressor system alleviates this problem and significantly reduces the size of the radiator. Due to the high radiator temperature the system is not very sensitive to coating degradation and the low power requirement makes it insensitive to the power penalty factor. The large number of components in the system, however, results in a large system weight, and in a low system reliability, which, in turn, requires a large amount of spares, redundancy, and repair time.

4.3.3 Vapor Adsorption Refrigeration

Vapor adsorption is among the oldest methods of refrigeration, having originated with Faraday in the Nineteenth Century. Faraday used ammonia as the refrigerant and silver chloride as the adsorbent; however, this process was apparently never used commercially, and remained a laboratory curiosity. The sulfur dioxide/silica gel system discussed previously was used commercially; however, it is no longer in use. The water/molecular sieve system proposed in this work has not been used commercially; one disadvantage is the low evaporator pressure of 0.12 psia, which makes atmospheric leakage into the system intolerable.

A schematic of the molecular sieve adsorption refrigeration machine proposed for lunar surface application is shown in Figure 4-7. The machine shown has four molecular sieve beds which are used for adsorbing refrigerant. The four beds are alternated in being cooled by radiator fluid flow while adsorbing refrigerant, and in being heated by fluid flow from a heat supply while desorbing refrigerant. Adsorption takes place at the condenser outlet temperature, and at the pressure corresponding to the vapor pressure of the refrigerant at the desired evaporator temperature. Desorption is accomplished at as high a temperature as is consistent with materials limitations and the available heat supply, and at a pressure corresponding to the vapor pressure of the refrigerant at the condenser temperature. Other variations are possible with regard to sequencing of radiator fluid flow through the condenser and the adsorbent bed. The sequencing shown is not necessarily the optimum.

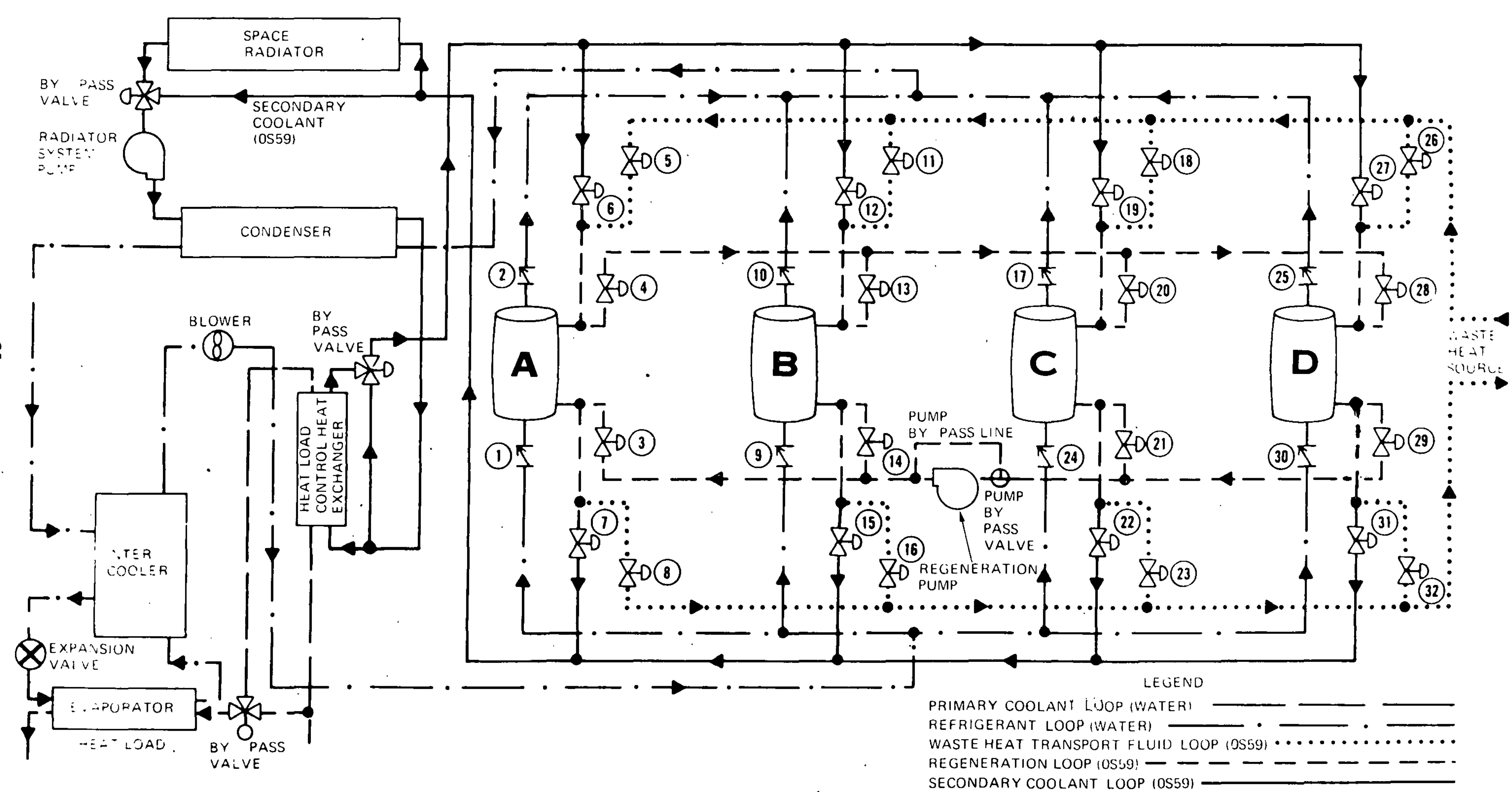


FIGURE 4-7

FOUR–BED ADSORPTION REFRIGERATION SYSTEM FLOW SCHEMATIC

Analysis has shown synthetic zeolite type 13x to be the most effective adsorbent for use with water in an adsorption refrigeration system. Since water is the working fluid, secondary radiator loops were considered mandatory. Due to the high heat rejection temperature of the adsorption system, the radiator is not very sensitive to coating degradation or to change in the environment. The fixed weight for the system is high; but since most of that weight is in high reliability adsorption beds, the major factor affecting the reliability is the large number of frequently cycled valves. The low-load problem is alleviated by by-passing part of the secondary loop flow to increase the evaporator load to maximum capacity at all times. The heat rejection is reduced by this method and the maximum/minimum ratio is a very modest 1.5/1 resulting in a very simple radiator system design.

4.3.4 Conventional Pumped Fluid Radiators (for Comparison Only)

The pumped fluid radiator heat rejection system is a highly developed and space qualified system (currently operational on the Apollo Mission). A typical advanced mission system shown in Figure 4-8 can be employed whenever the effective radiation sink temperature is lower than the temperature at which the thermal load is required. The working fluid, which could be ethylene glycol-water or one of the halocarbon family, picks up the heat load in the cold-side of a heat exchanger and then flows through an external radiator which consists of series and/or parallel flow tubes. Heat is rejected to the environment (sink) by radiation heat transfer from the radiator surface, thus cooling the working fluid which then flows through a circulation pump back to the cold side of the heat exchanger.

Since the radiator must be designed to reject the maximum heat load in hottest expected environments, low internal heat loads in cold environments can cause the radiator fluid to freeze in the radiator making the heat rejection system inoperative. Techniques such as selective stagnation and two-dimensional radiators (Reference [13]) have been developed by VMSC to solve this problem and no new developments, other than detail panel design and system integration for the specific application are required.

The spectral nature of the radiation environment in space provides an effective means of attaining low sink temperatures by judicious selection of coatings. However, due to the nuclear particle and ultraviolet radiation bombardment of all spacecraft surfaces, the radiator coatings degrade in long-term use resulting in an increase in the effective sink temperature. Other factors such as reflection and radiant interchange between other spacecraft and/or deployed equipment also reduce the heat rejection per unit area and increase the radiator area requirement. The only significant penalty for a large radiator system is for the radiator surface area, thus the penalty factor assigned for the radiator area is of prime importance. The use of integral external spacecraft structure which must be present in any event results in very low penalties for the radiator area. However, for large spacecraft; the internal heat generation per unit of external area increases rapidly as do the requirements for deployed equipment, viewports, hatches, docking ports, etc. The available external integral area may be insufficient so that deployed radiators may be required, thus significantly increasing the weight and complexity of the system. For some environments, such as in

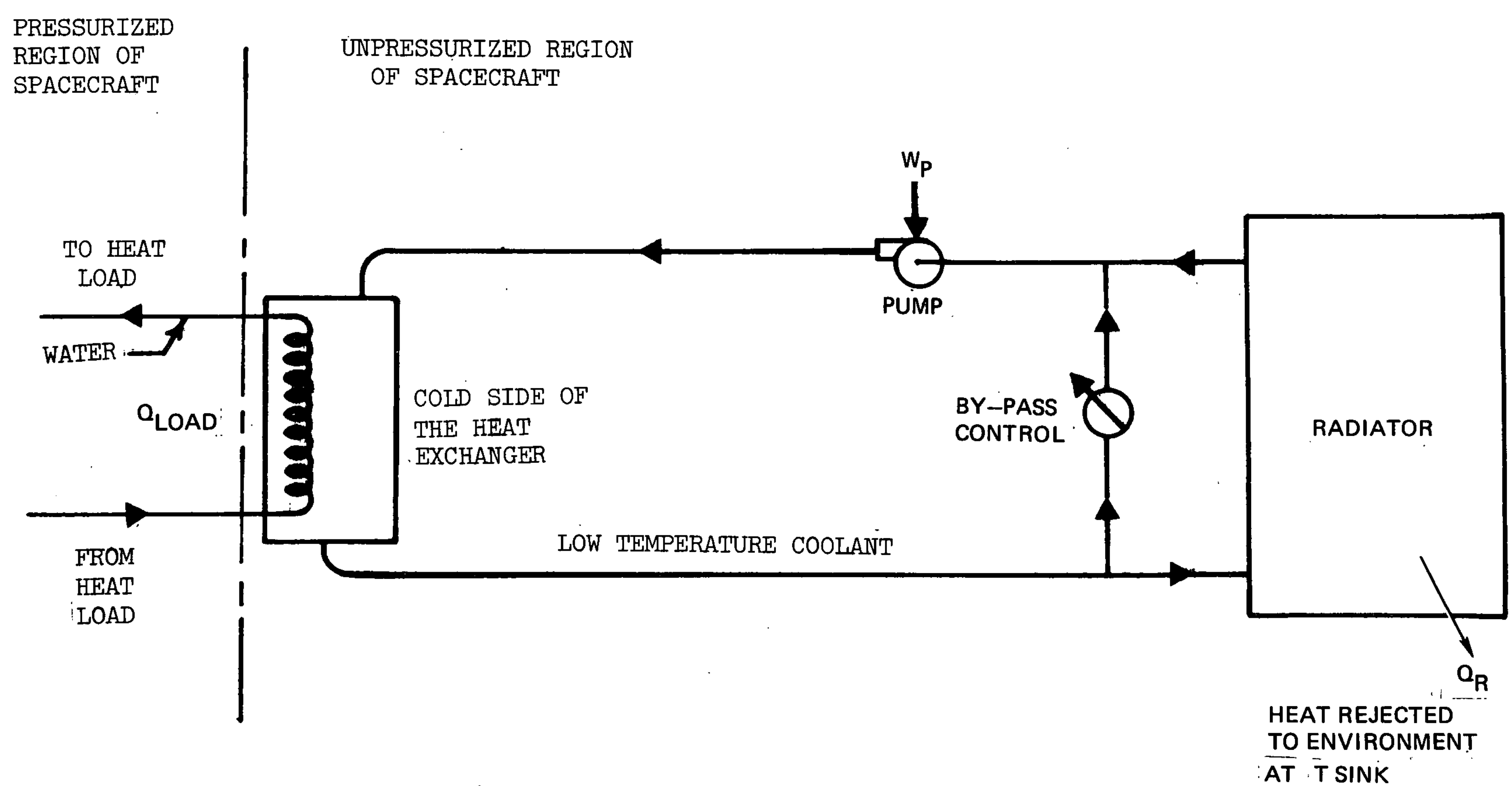


FIGURE 4-8 CONVENTIONAL PUMPED FLUID RADIATOR

a 10:1 crater at noon on the lunar surface, simple radiators cannot be employed because of a very high sink temperature so that no heat rejection would be possible without some thermal radiation shielding. Shielded radiators tend to be heavier and bulkier than conventional radiators, and they impose orientation constraints. Shielded radiators also present a complicated design problem for each specific application, and were not considered for comparative purposes in this study.

5.0 SYSTEM COMPARISON

This section presents the comparisons of candidate systems which were made in this work. These include a comparison of the total specific system weights, a comparison on the basis of the Effectiveness Function, and a comparison on the basis of subjective factors.

5.1 Candidate System Specific Weight Comparison

The specific weight for the refrigeration systems was defined as the installed or "fixed" weight which includes all hardware and lines plus the power penalty in terms of weight plus the radiator area penalty in terms of weight. (Waste heat penalty could also be included, but this was usually considered to be zero). The power and radiator weights were considered in terms of weight penalties because these two parameters are very strongly influenced by specific vehicle design. Details of individual vehicle designs are, of course, much beyond the scope of this work. Because of the wide variation possible in these parameters, several penalties were considered in the weight comparisons. Table 5-1 lists the specific values of evaporator temperature (TEVAP), effective radiator sink temperature (TSINK), electrical power penalty (FEPP), and radiator area penalty (FRAP) which were evaluated in the study. System weight comparisons were made for each possible combination of the parameters, for a total of 120 separate comparisons. The comparisons are made in terms of total system specific weight (in lb/KW_t) as a function of system radiator (i.e., condenser) operating temperature. The curves presented do not include any weight for system redundancy, back-up systems or spares. The weight of a redundant system would not be twice that shown, since power penalty includes power system redundancy, and a second radiator would require only additional tubes and fluid, and would not require additional fin material and structure.

The computer routine used for determining the weight for each system as a function of radiator temperature and for plotting and results is discussed in Appendix C. The specific runs made in this work, and all of the specific weight vs. radiator temperature curves are given in Appendix D. Figure 5-1 presents the specific weight versus radiator temperature for all five candidate systems for a set of conditions which is similar to a space shuttle orbiter application; that is, for an evaporator temperature of 35°F, an environmental sink temperature of 20°F, a power penalty of 300 LB/KW_e, and a radiator weight of 1 LB/ft^2. The results show that the vapor compression system is the lightest out to a radiator temperature of about 120°F. Beyond that, the complex lithium bromide/water absorption system is the lightest. Of course, this heat-driven system is not really a candidate for the shuttle due to lack of a suitable heat source. Figure 5-2 presents results for a set of conditions which is similar to a lunar orbiting station. The results are very similar to those in Figure 5-1; the complex lithium bromide/water absorption refrigeration system is the lightest, followed closely by the vapor compression system. Consideration of redundancy requirements would alter these results, since a backup vapor compression system would require only a small amount of additional fixed weight while the complex lithium bromide absorption system would require a great deal (primarily because this system essentially includes a power generation system.) Figure 5-3 gives the results for a Lunar Surface Base with a high sink temperature of 100°F. In this case, the minimum weight systems are the H_2O/zeolite adsorption and the complex $LiBr/H_2O$ systems, with the minimum weight occurring at radiator temperatures

TABLE 5-1 OPERATING CONDITIONS AND PENALTY FACTORS CONSIDERED IN TRADE ANALYSES*

Evaporator Temperature (TEVAP)	35°F** and 45°F
Effective Sink Temperature (TSINK)	-78°F, 0°, 20° and 100°F
Electrical Power Penalty (FEPP)	300, 500, and 700 LB/KW_e
Radiator Area Penalty (FRAP)	0.1, 0.6, 1.0, 1.5 and 2.0 LB/FT^2
Thermal Energy Penalty	0 LB/KW_t

*All 120 possible combinations were evaluated by the Refrigeration System Comparison routine for five active refrigerators and the resulting plots are presented in Appendix D.

**The evaporator temperature of 35°F corresponds to a cabin coolant return temperature of 40°F; the 5 °F difference represents the temperature difference across the heat exchanger equipment.

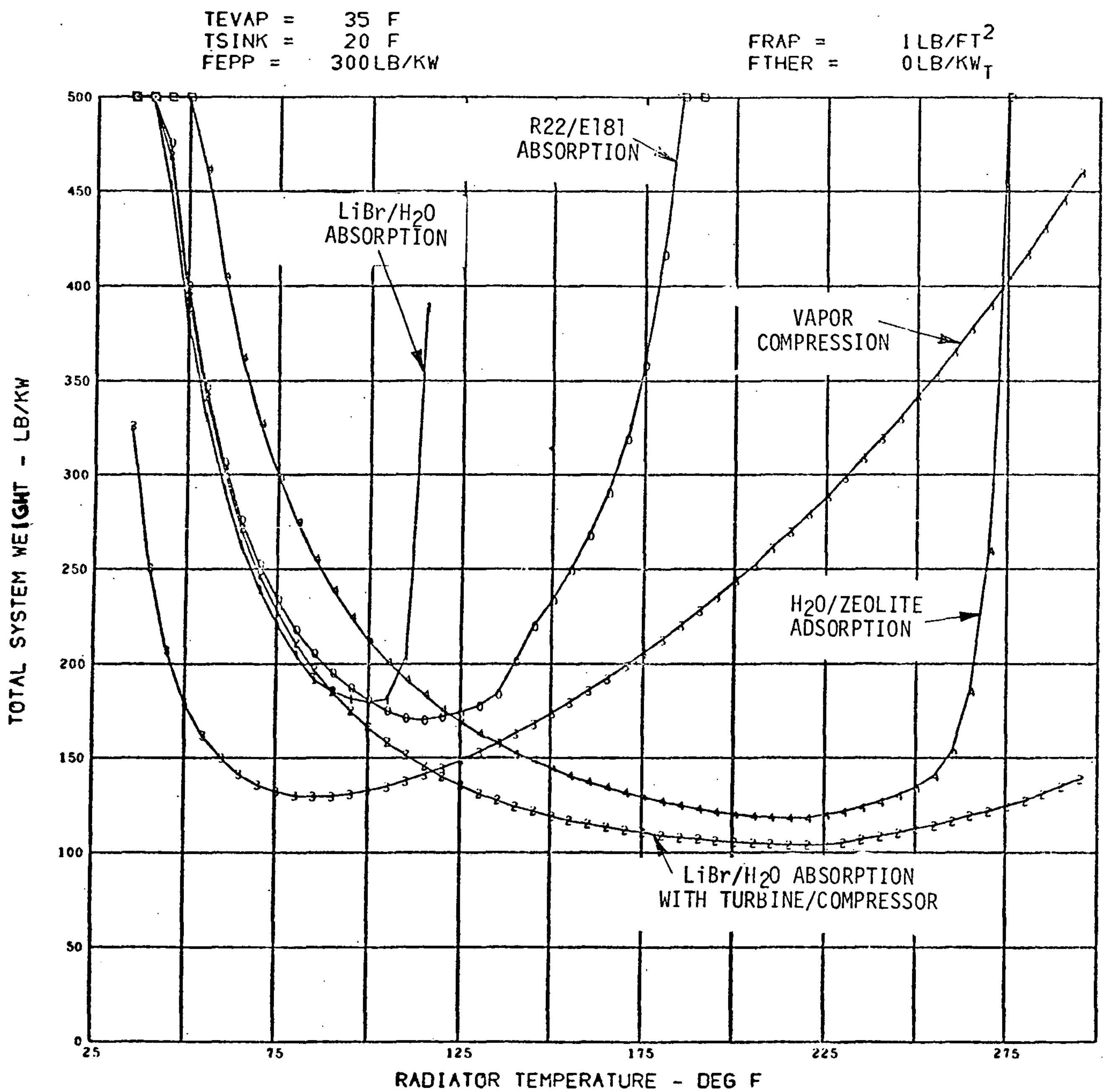


FIGURE 5-1 REFRIGERATION SYSTEM WEIGHT FOR SPACE SHUTTLE ORBITER APPLICATION

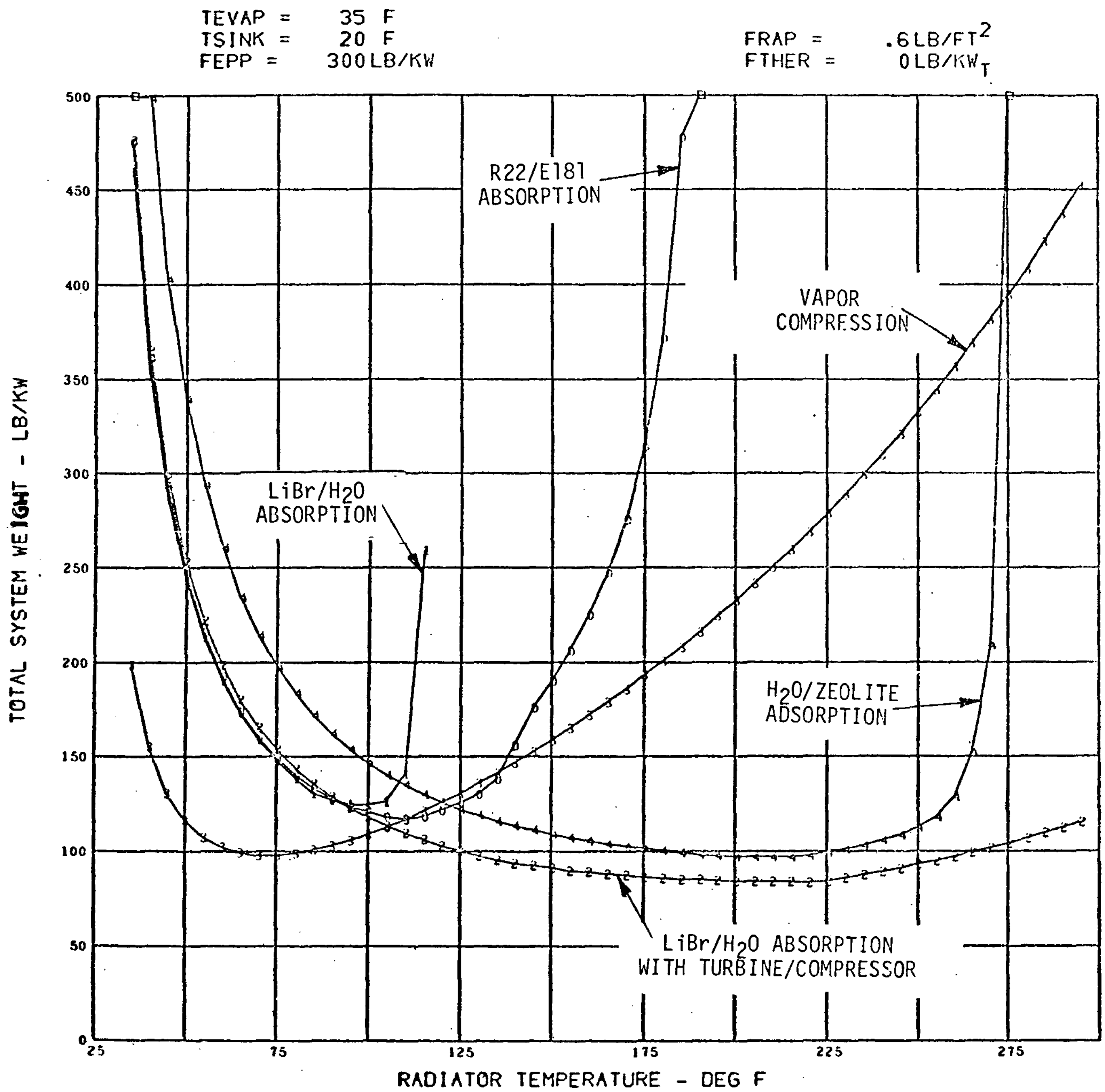


FIGURE 5-2 REFRIGERATION SYSTEM WEIGHT
FOR LUNAR ORBIT APPLICATIONS

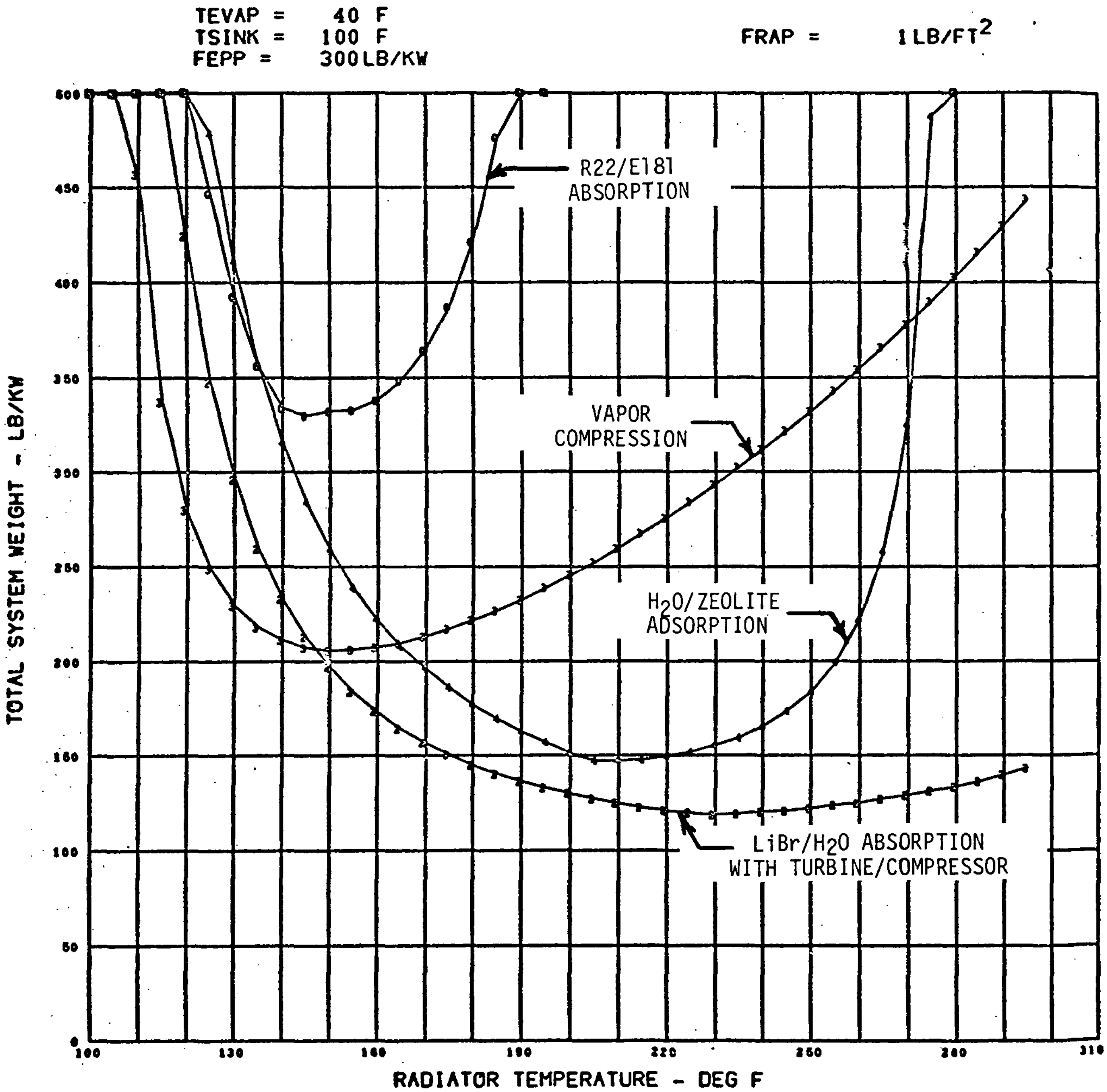


FIGURE 5-3

REFRIGERATION SYSTEM FOR LUNAR SURFACE BASE APPLICATION in the 200-230°F range. For radiator temperatures below 150°F the vapor compression system is the lightest. It should be pointed out that these results are for the hottest period of the lunar day, and that an ordinary radiator could be used throughout the lunar night, and for much of the lunar day. Also, the power penalty of 300 lb/KW_e may be high because a solar cell power system without batteries could be dedicated to driving the refrigeration system, since it would only be required in the daylight period. This would enhance the mechanical vapor compression system. Figure 5-4 presents results for a Lunar Surface EVA Application. The results are much the same as for the Lunar Surface Base. It should be pointed out, however, that these results are for a continuously operating system. If the EVA duration were limited to 8 hours, then battery power could be used with the result that the power penalty would be about 40 lb/KW_e (for a Li-CuF_2 secondary storage battery with an energy density of 200 watt-hr/lb and a 60% depth of discharge). The power penalty would thus be shifted to the base, but even there it would be lower since the battery could be recharged at a lower rate and/or when power demands are low. This would make the mechanical vapor compression system the lightest with a specific weight of 120 lb/KW_t with a radiator temperature of 140°F. Figure 5-5 presents results which are representative of a transmartian vehicle application. The H_2O/zeolite adsorption and the complex LiBr/H_2O absorption are the lightest weight systems. In this application the use of a dedicated solar cell system without batteries is possible, and this would enhance the mechanical vapor compression system.

The effect of radiator area penalty on system specific weight is shown by Figure 5-6. The system specific weight has been optimized for radiator temperature at each radiator area penalty. This figure shows that, for the conditions of 40°F evaporator, 0°F sink, and 700 lb/KW_e power penalty, the vapor compression system is the lightest for area penalties up to 0.72 lb/ft^2 (along with the conventional pumped fluid radiator which was included for comparative purposes only). Above this radiator area weight penalty out to 2 lb/ft^2 (the maximum value considered), the complex LiBr/H_2O absorption system is the lightest, followed closely by the H_2O/zeolite adsorption system. The reason for the advantage of these two systems at large radiator area penalties is that they operate relatively efficiently with radiator temperatures in the 200-250°F range.

The effect of radiator area penalty on required radiator area is shown in Figure 5-7. This curve shows that the complex LiBr/H_2O absorption and H_2O/zeolite adsorption systems require the least radiator area for any radiator area penalty. This is because the use of water as the refrigerant in these systems makes them efficient at relatively high operating temperatures. More significantly, Figure 5-7 shows that the mechanical vapor compression system shows an area advantage over the conventional pumped fluid radiator for radiator area penalties above 0.6 lb/ft^2. Of course, a reduction in power penalty from 700 lb/KW_e would shift the deviation point to lower values of radiator area penalty.

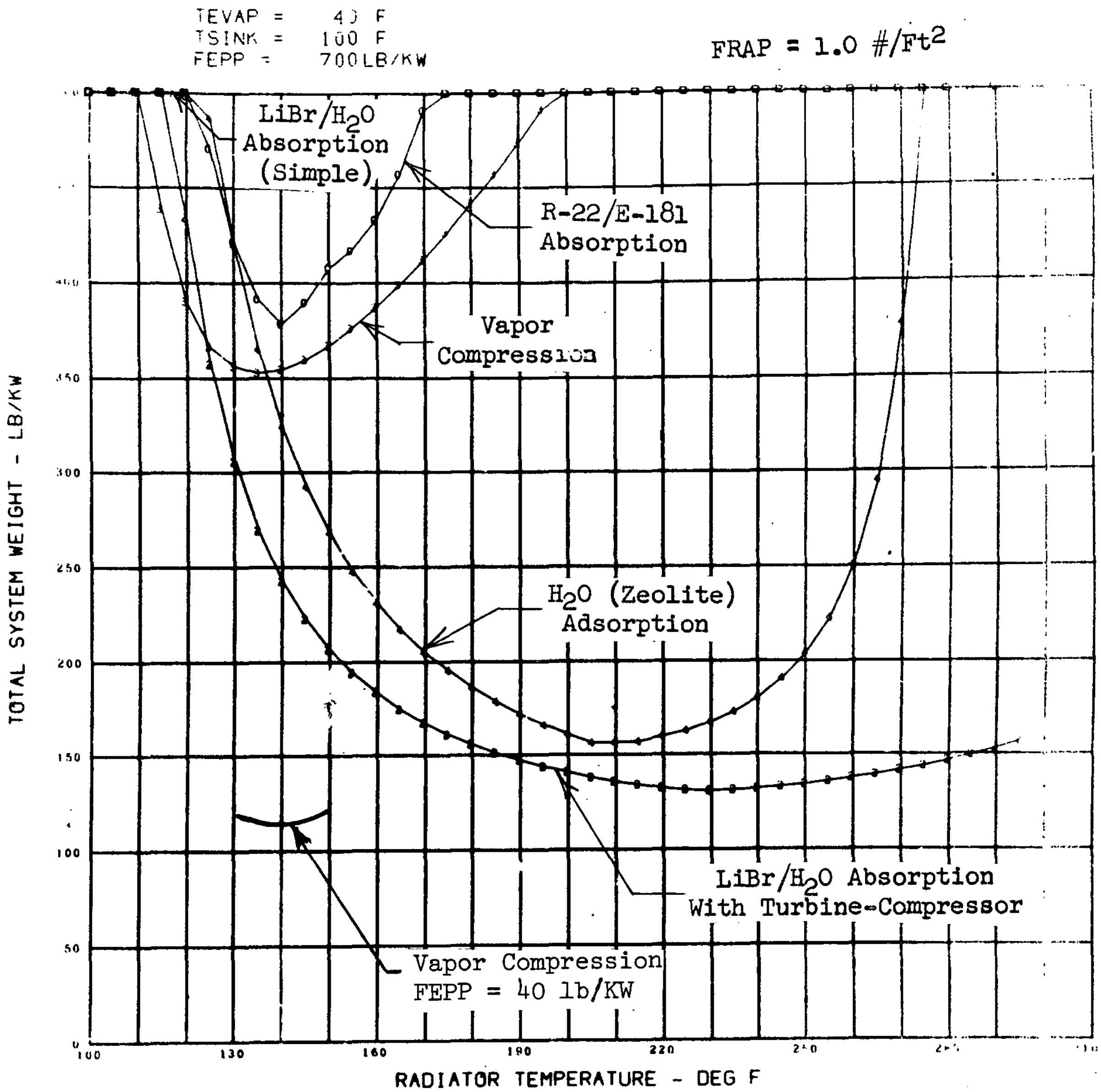


FIGURE 5-4

Refrigeration System Weight
For Lunar Surface EVA Application

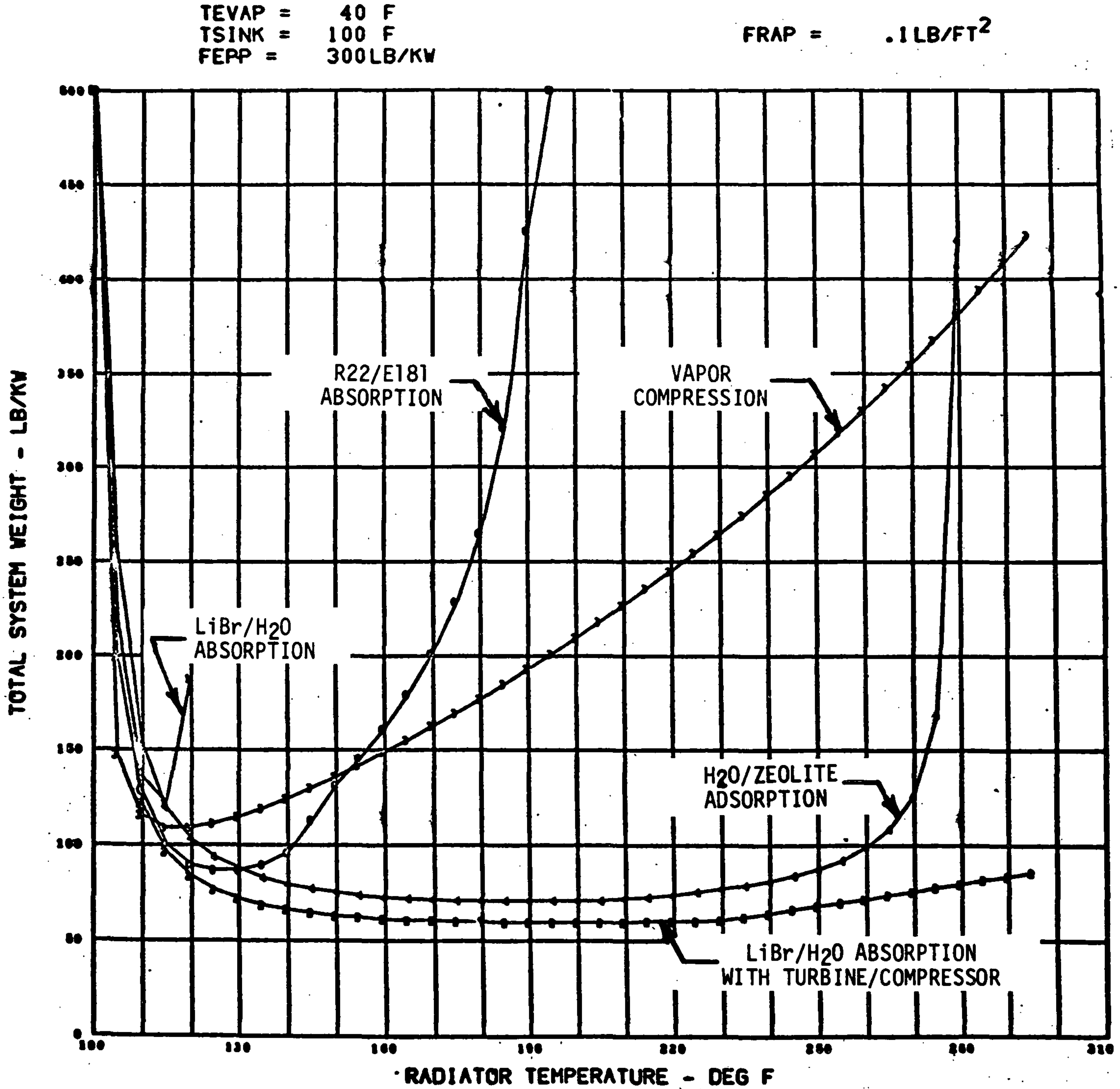


FIGURE 5-5

REFRIGERATION SYSTEM WEIGHT
FOR TRANSMARTIAN APPLICATION

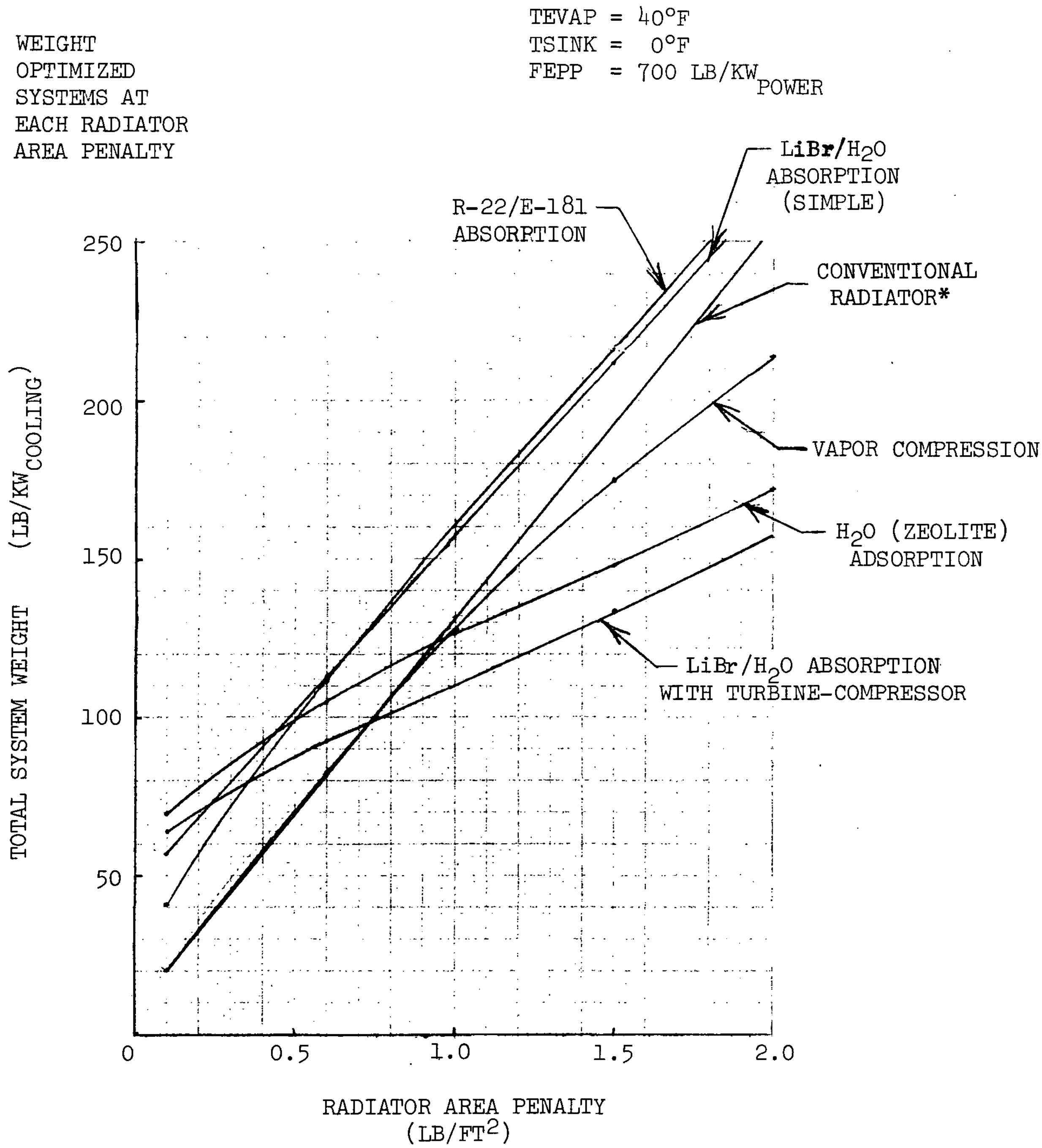


FIGURE 5-6

EFFECT OF RADIATOR AREA PENALTY UPON
HEAT REJECTION SYSTEM WEIGHT
IN EARTH ORBIT

*AVERAGE RADIATING TEMPERATURE IS 45°F

WEIGHT
OPTIMIZED
SYSTEMS AT
EACH RADIATOR
AREA PENALTY

TEVAP = 40°F
TSINK = 0°F
FEPP = 700 LB/KW_{POWER}

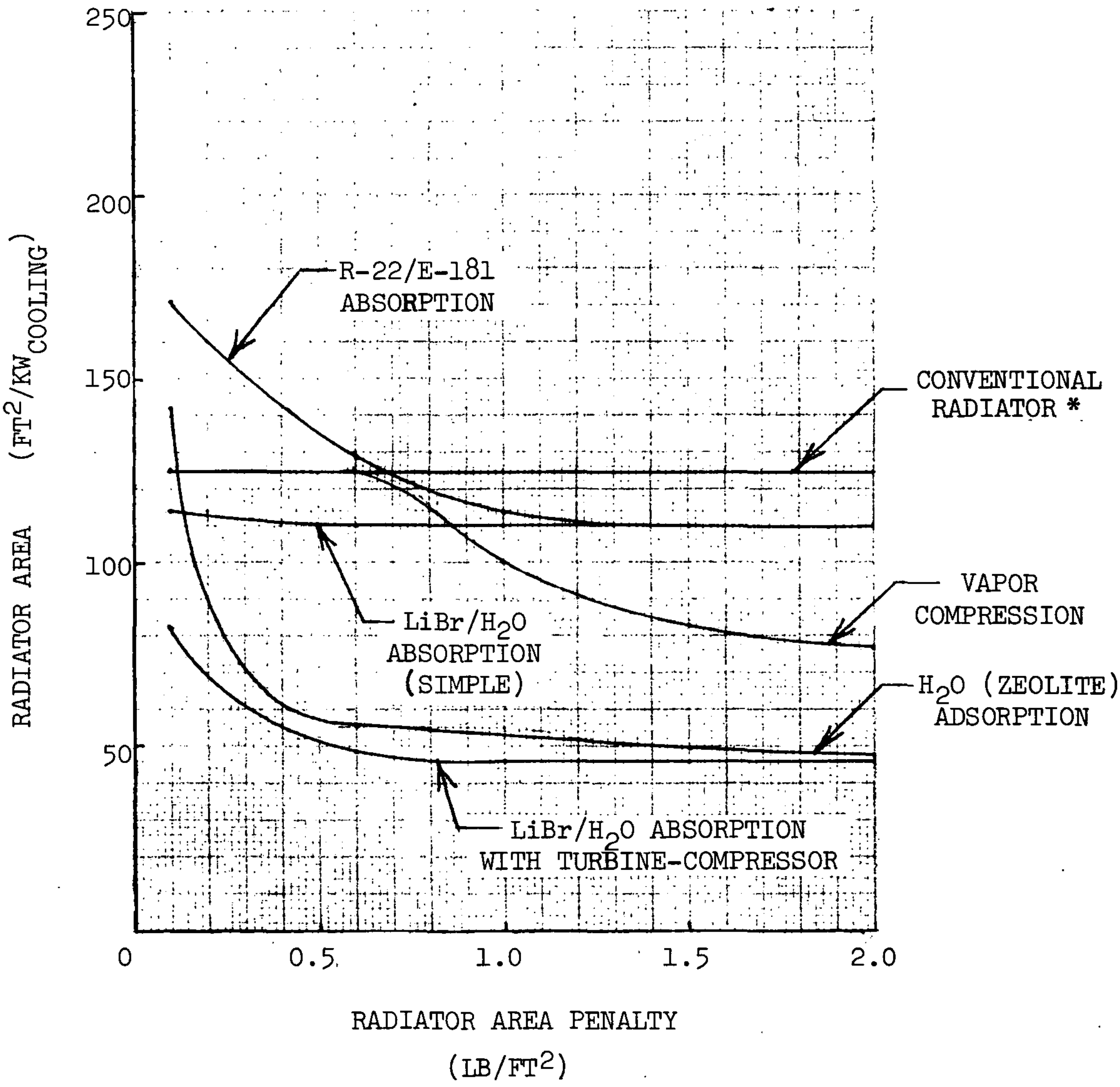


FIGURE 5-7

COMPARISON OF RADIATOR AREA REQUIREMENTS IN EARTH ORBIT

*AVERAGE RADIATING TEMPERATURE IS 45°F

5.2 System Effectiveness Comparison

As discussed in Section 3.4, the five candidate refrigeration systems (plus integral and deployed conventional radiator systems for comparative purposes) were rated with an effectiveness function for the missions shown in Table 3-2. The theory of the effectiveness function is presented in Appendix A, and the application of it to refrigeration system comparison is given in Appendix B, including sample calculations.

The specific factors considered in the effectiveness function were:

(1) Development cost

(2) Fabrication cost

(3) Launch costs

(4) Equivalent volume penalty costs

(5) Maintenance costs

(6) Redundancy and spares costs

(7) Technical risk costs

(8) Systems integration and interface costs

The effectiveness function was evaluated in terms of costs; however, another base, such as weight, could have been used. Thus, theoretically, the difference in the effectiveness function for two systems represents the actual difference in cost. The effectiveness function does not represent the actual cost of the system; however, because many factors which are judged to be approximately the same for all systems, such as administrative costs, are not included.

The comparisons of the effectiveness function for the candidate systems for the various applications are given in Figures 5-8 through 5-16.

Figure 5-8 shows a comparison of cost (above a common base) in millions of dollars for seven possible heat rejection systems (the 5 basic refrigeration systems considered plus conventional and deployed radiators for comparison), for a space base mission with a $1000/lb launch cost. The bars show the cost for systems with heat rejection capacities of $76KW_t \pm 41KW_t$. The vapor compression refrigeration system has a considerably lower cost than any of the other refrigeration systems. It is interesting that in this case, the vapor compression refrigeration system is only a narrow loser to the conventional radiator systems.

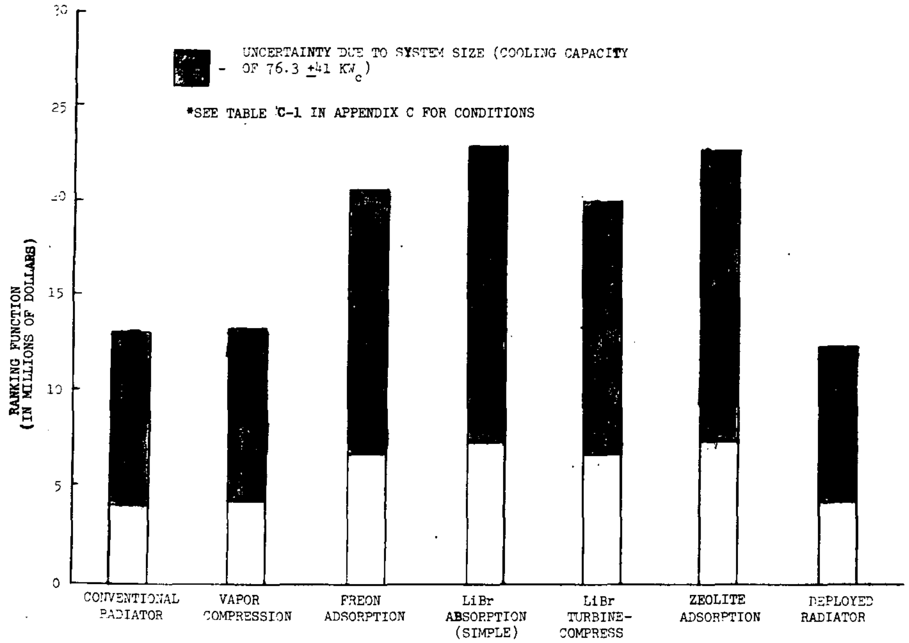


FIGURE 5-8 HEAT REJECTION SYSTEM RANKING FOR AN EARTH ORBIT SPACE BASE AT 1000 $/LB LAUNCH COST*

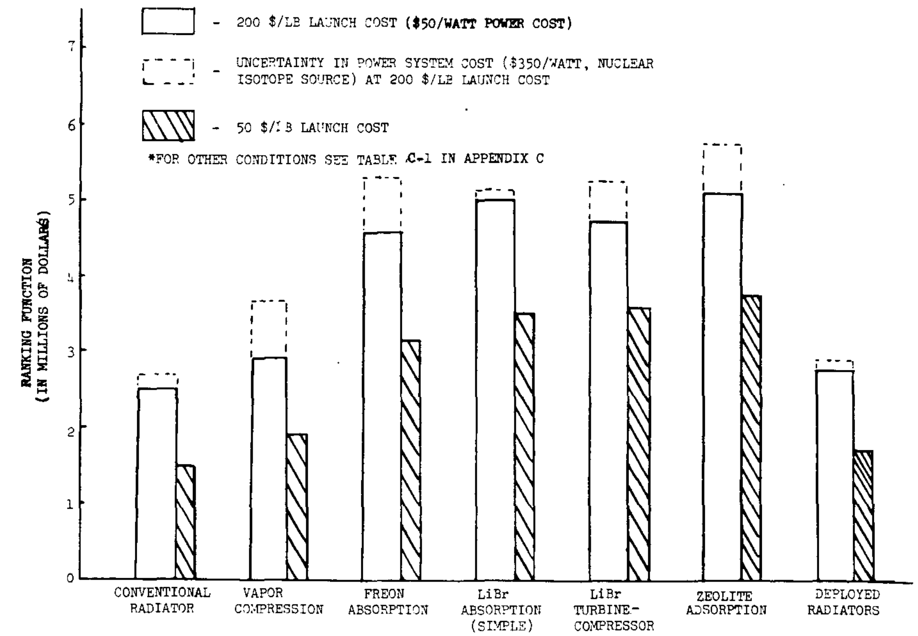


FIGURE 5-9 HEAT REJECTION SYSTEM RANKING FOR AN EARTH ORBIT SPACE BASE AT $200/LB and $50/LB LAUNCH COST*

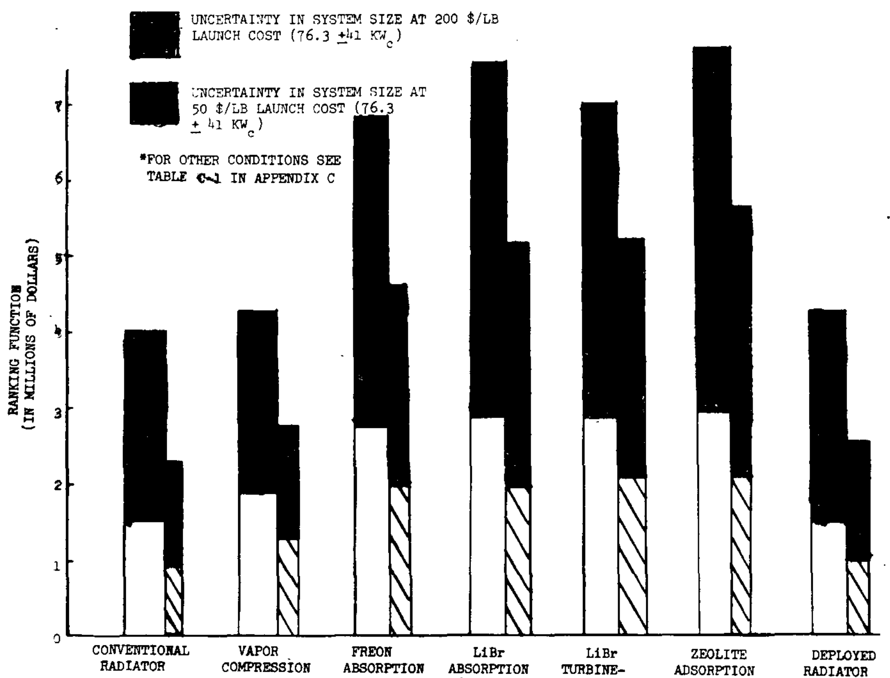
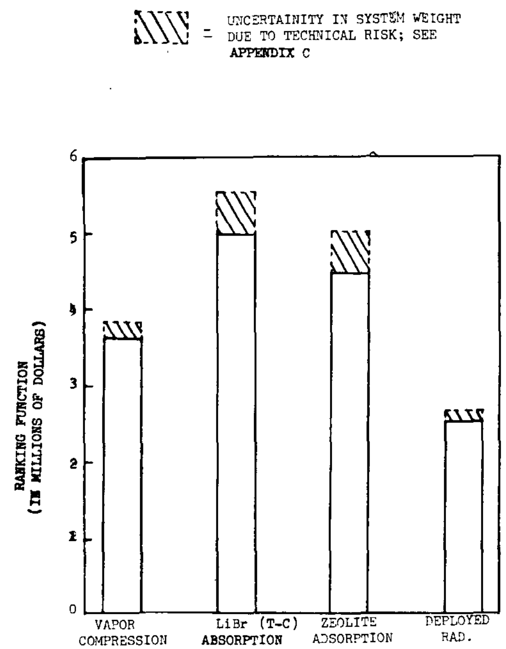
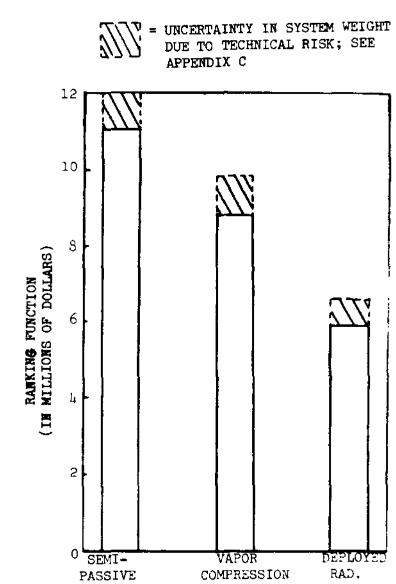


FIGURE 5-10 HEAT REJECTION SYSTEM RANKING FOR SPACE BASE WITH VARIOUS LAUNCH SYSTEM COST AND SYSTEM SIZES*



FIGURF 5-11 HEAT REJECTION SYSTEM RANKING FOR AN EARTH ORBIT SPACE STATION*

*SEE TABLE C-1 IN APPENDIX C



FIGURF 5-12 HEAT REJECTION SYSTEM RANKING FOR AN EARTH ORBITING SHUTTLE*

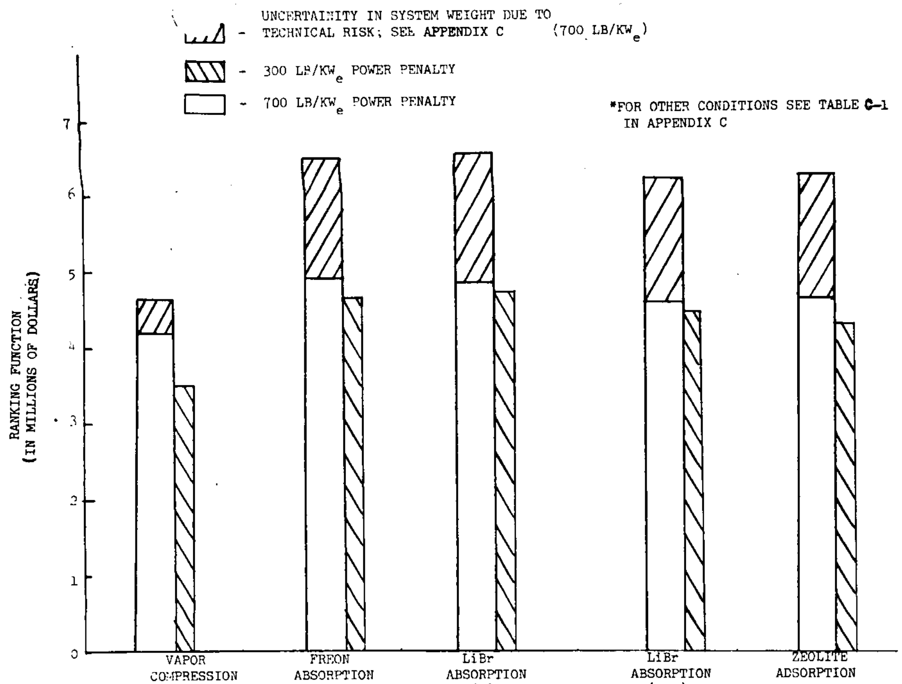


FIGURE 5-13 HEAT REJECTION SYSTEM RANKING FOR A LUNAR ORBIT SPACE STATION*

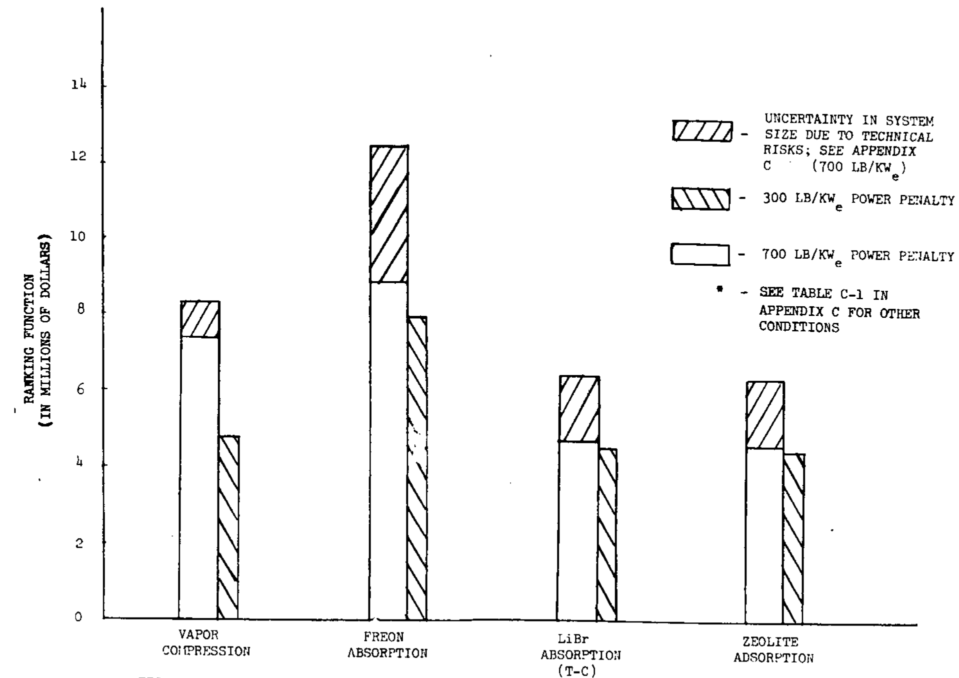


FIGURE 5-14 HEAT REJECTION SYSTEM RANKING FOR A LUNAR SURFACE BASE*

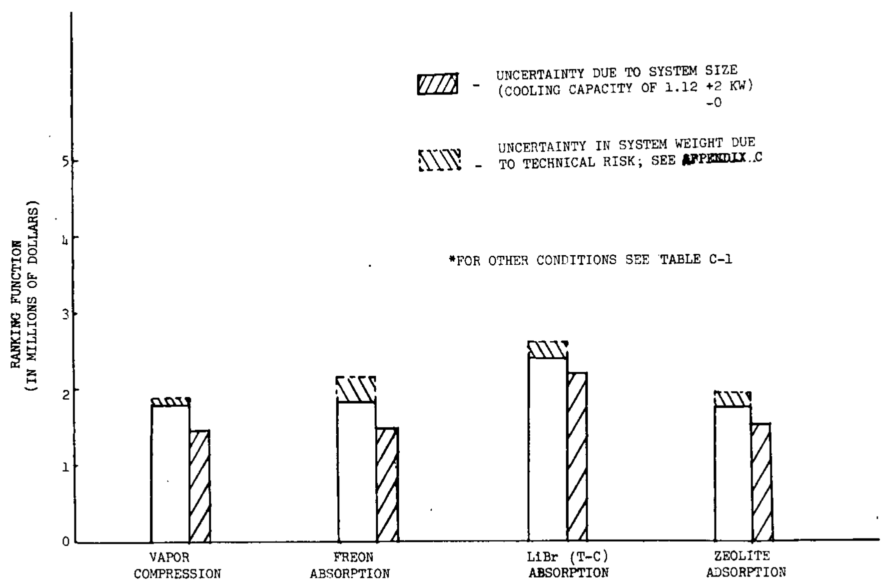


FIGURE 5-15 HEAT REJECTION SYSTEM RANKING FOR LUNAR SURFACE EVA*

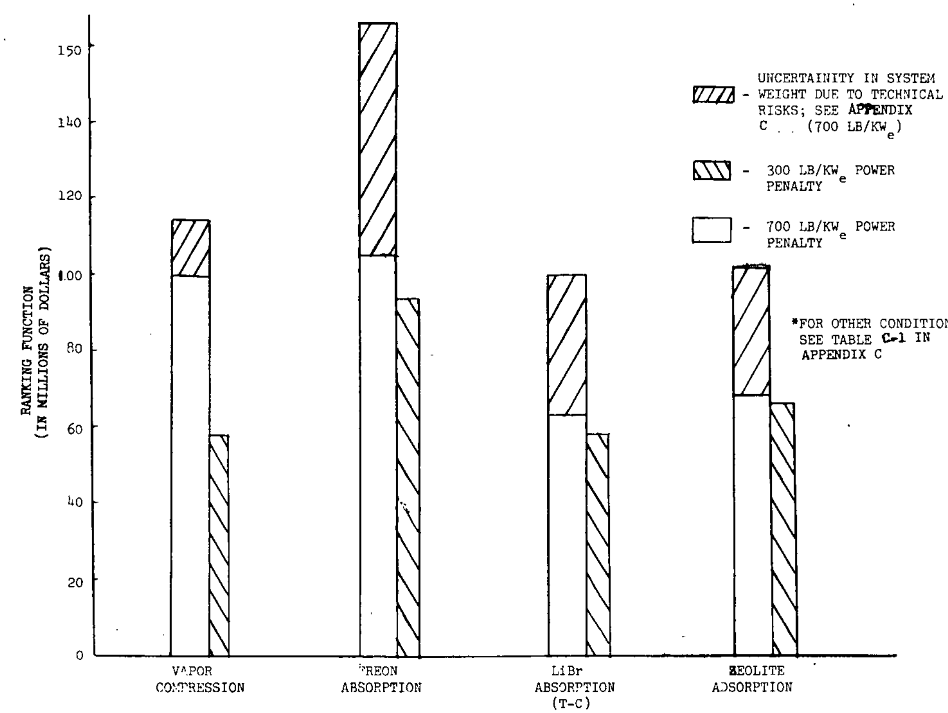


FIGURE 5-16 HEAT REJECTION SYSTEM RANKING FOR A TRANSMARTION MISSION*

This is because the conditions considered do not demand a refrigeration system, since there is a 0°F sink temperature, and the mechanical vapor compression system can be optimized with a low condenser temperature. Since the refrigeration system uses a condensing radiator, and thus has an almost constant temperature, the refrigeration system requires less radiator area than the conventional radiator systems. Figure 5-9 shows the same comparison for launch costs of \$200/lb and \$50/lb. The results are the same for both of these launch costs, with the vapor compression refrigeration system being much less costly than the other refrigeration systems.

Figure 5-10 shows the variance in ranking cost for the refrigeration systems for variance in system cooling capacity and in power system cost. The vapor compression system remains the lowest cost refrigeration system for all variances.

Figure 5-11 shows a comparison of ranking cost for three refrigeration systems and a deployed radiator system for an earth orbiting space station. In this case, the vehicle heat load is 120,000 BTU/hr (35 KW_t), the radiator sink temperature is 20°F for integral area and -78°F for deployed area, and the integral radiator area is limited to 2500 sq. ft. The results show that the vapor compression refrigeration system has the lowest cost among the refrigeration systems. The heat-driven systems were considered for this application, but are not likely candidates because the space station will probably be powered by solar cells - so no heat source will be available. It is interesting to note that the Deployed Radiator System has the lowest cost of the four candidate systems. The integral radiator system could not be used in this case because of the limited area availability and the high sink temperature.

Figure 5-12 presents a comparison of costs for a conventional radiator system, a vapor compression refrigeration system, and a deployed radiator system on an earth orbiting space shuttle. The conventional radiators and the vapor compression system radiators are located on the cargo bay door interiors, are limited in area to 900 sq. ft. and have an equivalent sink temperature of 20°F. The deployed radiator has an area of 1800 sq. ft. with an equivalent sink temperature of -78°F. Figure 5-12 shows that the vapor compression system is less costly than the conventional radiator system, and that the deployed radiator system is the least costly of the three heat rejection systems. (Heat-driven refrigeration systems were not considered because of the lack of a suitable waste heat source from the Shuttle Orbiter Power System.)

The Lunar Orbit Space Station results are given in Figure 5-13. This figure shows the effects of a change in power penalty from 300-700 lb/KW_e, and it shows the uncertainty in system cost due to technical risk. The vapor compression system has the lowest cost, even when considered under the worst circumstances, i.e., with a large power penalty (700 lb/KW_e) and with the greatest expected cost for the vapor compression system compared against the minimum expected costs for the other refrigeration systems. The heat-driven refrigeration system costs are very similar in all cases.

Results for the Lunar Base application are shown in Figure 5-14. The simple LiBr/H_2O Absorption System is not shown because it will not function under the design conditions considered. With a power penalty of 300 lb/KW_e, the vapor compression system has a slight edge over the H_2O/Zeolite Adsorption System and the LiBr/H_2O Absorption System augmented by a turbine/compressor. For a power penalty of 700 lb/KW_e, the H_2O/Zeolite Adsorption System and the LiBr/H_2O Absorption system augmented by a turbine/compressor have nearly equal costs, and are below the cost of the vapor compression system. The R22/E181 Absorption System has the highest cost in all cases.

The Lunar Surface EVA mission results are shown in Figure 5-15. These results show that the vapor compression system and the vapor adsorption system are essentially tied with the lowest costs of any of the candidate systems. This study considered the total cost of the systems; however, there was no consideration of the EVA application weight and volume constraints, resulting from the crewman's physical limitations. This is significant because the heat rejection system used in an EVA application must be portable. The vapor compression system has some advantage over the other systems in this regard because of the relatively short duration of EVA missions, which makes it feasible to use battery power as previously discussed.

Figure 5-16 shows the results for the Transmartian mission. At a power penalty of 300 lb/KW_e the LiBr/H_2O Absorption System augmented by a turbine/compressor has a narrow cost advantage over the vapor compression system, and a slightly larger advantage over the adsorption system. At 700 lb/KW_e the LiBr/H_2O Absorption System has the lowest cost followed closely by the H_2O/Zeolite Adsorption System with the vapor compression system third by a margin of $30 million. Considering system cost uncertainty due to technical risk the vapor compression system remains third, but by a much smaller margin of only $10 million.

5.3 Subjective Evaluation

The earliest potential application for a spacecraft refrigeration system is in the space shuttle orbiter. The mechanical vapor compression system is the only serious candidate (among refrigeration systems) for this application. The mechanical vapor compression system is the only candidate system which is applicable to all missions. If a shuttle orbiter refrigeration system were developed, then all of the previous effectiveness analyses would be shifted very strongly toward the vapor compression system, since it would then have lower development cost, lower technical risk, etc.

6.0 CONCLUSIONS AND RECOMMENDATIONS

6.1 Conclusions

The following significant conclusions have been reached in this refrigeration system comparison:

(1) The vapor compression refrigeration system has the lowest cost of any refrigeration system in four (earth orbiting space station, space base, shuttle, and lunar orbit station) of the seven applications presented, and is essentially tied with vapor adsorption in one other application (the Lunar Surface EVA). Other considerations, primarily the system portability requirement which limits system weight, make the vapor compression system the favored system in the EVA mission also. The vapor adsorption and vapor absorption (LiBr/H_2O with a turbine/compressor) systems have the lowest cost (and are about equal in cost) for the Lunar Base and the Transmartian Excursion.

(2) The mechanical vapor compression system is the only refrigeration system which is applicable for all missions.

(3) The earliest, and one of the most promising, applications for a refrigeration system is in the space shuttle orbiter. Even though the vapor compression refrigeration system is more costly than a deployed radiator system in earth orbit, it offers some advantages in vehicle operational simplicity (e.g., reduced orientation and maneuvering constraints). The vapor compression system offers advantages since it can be used in all pre-flight, flight, and post-flight phases in shuttle orbiter operation.

(4) Heat-driven refrigeration machines offer little advantage in earth orbital operation because of low COP's which greatly increase heat rejection requirements and thus radiator area requirements. For lunar surface operation some advantage can be gained with these machines if they are designed to operate with a very high condenser (and thus radiator) temperature. The H_2O/Zeolite Adsorption system is well suited to this type of operation; however, it has the disadvantage of requiring a very high (500°F range) heat source. The availability of waste heat at this temperature is strongly dependent on the power system. A power system using a Brayton cycle would be severely penalized if it supplied heat at this temperature, whereas a thermoelectric system would not be penalized.

(5) The vapor compression system is the only refrigeration system which can be applied to spacecraft in the near-term because the early space station will be powered by solar cells, and the space shuttle will be powered by fuel cells, so there will be no waste-heat at a temperature suitable for driving a refrigeration system on either of these vehicles.

(6) If the vapor compression system were developed for the early missions then it would be more attractive for the later systems than the results indicate because development costs would be amortized over a greater number of flight units.

(7) Without going to 200°F radiator temperatures, the vapor compression system is much more effective in reducing radiator area requirements or in performing with a radiator area limitation.

(8) For a space station with limited area availability, the vapor compression refrigeration system offers an alternative to a deployed radiator system. A refrigeration system could be used to augment a conventional radiator system during periods of high heat loads or of exposure to unfavorable external environments.

(9) The consideration of redundancy requirements (as was done with the Effectiveness Function) enhances the mechanical vapor compression system relative to the heat-driven systems.

(10) The vapor adsorption system ranks second to vapor compression in the overall comparison of refrigeration system costs.

(11) The vapor adsorption system has the advantage that it requires no liquid-gas phase separation. (The vapor compression system may involve separation of liquid lubricant and vapor refrigerant).

6.2 Recommendations

The following recommendations are made based on the results of this work:

(1) Vapor compression is the most promising refrigeration system for spacecraft applications, and development of this system should be pursued. The preliminary design to be executed as a part of this contractual effort should be on a vapor compression refrigeration system.

(2) The philosophy on assignment of power penalties for spacecraft refrigeration systems should be evaluated in light of the variance in reliability requirements between the power and environmental control systems. It appears that a power supply dedicated to a refrigeration system would not require a higher reliability than the refrigeration system itself. The assigned power penalties should also include consideration of intermittent operation over the mission lifetime and use of reserve installed power system capacity.

(3) The vapor adsorption refrigeration system should be investigated for application where a heat-driven system is desirable.

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APPENDIX A

EFFECTIVENESS METHODOLOGY

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APPENDIX A

EFFECTIVENESS METHODOLOGY

1. EFFECTIVENESS MEASURE

In the context of military operations analysis and systems evaluation, it may be difficult to arrive at a single Effectiveness Index which is both workable and comprehensive. This is ofen the case in subsystem development areas where the requirements are still general or in the formative state. This section discusses the selection and use of an Effectiveness Index of a subsystem when the overall system requirements and properties may not yet be fully known. This is done by deriving a ranking model from the constrained optimization problem, using so-called "trade factors". These are partial derivatives of the Accountable Factors, concept data deemed most likely to eventually determine overall system effectiveness. Methods of estimating these trade factors within error bounds are given with a sensitivity analysis technique for paired comparisons of subsystem candidates.

a. Accountable Factors

Accountable Factors x_i are those things deemed most pertinent to the problem. If the emphasis is on the evaluation of competing systems the Accountable Factors are usually those physical characteristics of the mission and the systems which distinguish one system from another, e.g., weight, reliability, accuracy, etc. Cost is also a factor but may be kept separate so that first a purely technical evaluation may be made before making one which includes cost.

b. Effectiveness Function

The Effectiveness Function

$$E = F(x_1, x_2, \ldots, x_m) \tag{1}$$

is defined to be a real valued function of the m Accountable Factors x_i. It may or may not have a physical interpretation such as: Dollars per pound of payload in orbit; Launch rate per year; or, Sortie rate per bomber squadron, etc. But it is essential that it have what is sometimes called the Monotone Property: In the permissible range of the variables, if it is increasing monotone, then any increase in E must signify an increase in desirability, and vice versa if it is decreasing monotone.

Examples of what we might call Mixed Indices (i.e., having no physical units of measure) might include:

$$E = \frac{(\text{launch rate per yr.}) \times (\text{reliability})}{(\text{dollars per lb in orbit})} \tag{2}$$

This might be a suitable index for comparing two space booster systems under the constraints of a fixed number of launching pads and processing facilities. It has the Increasing Monotone property. It might not be a suitable index for comparing two booster systems having widely different launch concepts and no constraints on the number of launch sites. Here additional Accountable Factors might have to be introduced, such as fixed costs and recurring costs, the total number of missions to be provided for, etc.

c. Transformation of the Effectiveness Function

The example of an effectiveness function given in equation (2) shows that the reciprocal of E would serve equally well. It would then have the decreasing monotone property. Also the logarithm of E would serve as a measure of effectiveness and would have the increasing monotone property. In fact, any one-to-one transformation of the effectiveness function or any of the accountable factors would produce an associated effectiveness function having the desired monotone property.

d. The Ranking Problem

Given a suitable Effectiveness Index E as defined in (1), and given n systems to be evaluated and compared, we may think of having an $(m \times n)$ matrix of data representing the m values of each of the m accountable factors on each of the n systems, i.e.,

$$H = (h_{ij}) \ , \quad h_{ij} = \text{the value of the } i\text{th accountable factor on system } j \ . \tag{3}$$

To rank these we would merely substitute the values in the effectiveness function in (1) and compare the resulting values of E .

This ignores two problems: (1) We may not know the Effectiveness Function completely, and (2) There may be uncertainties in our Data.

2. METHODOLOGY DEVELOPMENT (FIRST ORDER)

To deal with the two questions above, we first linearize the effectiveness function and proceed to find quantities which may yet be estimated and yet permit ranking if not absolute evaluation. Later we will extend this to a second order case. Let the effectiveness function in (1) be developed in a Taylor's Series with remainder as follows: Let

$$E = F(\underline{x}_j) = F(x_{1j}, x_{2j}, \ldots, x_{mj}) \tag{4}$$

$\underline{x}_j = x_{1j}, x_{2j}, \ldots, x_{mj}$ = the $j\underline{th}$ system parameters

$\underline{c} = c_1, c_2, \ldots, c_m$ = a reference system or coordinate origin

Let the increments from the point $\underline{c}$ be defined by

$$x_{ij} = c_i + h_{ij} \tag{5}$$

Then by use of Taylor's series with remainder we get

$$F(\underline{x}_j) = F(\underline{c}) + h_{1j} F_1^{(1)} + \ldots + h_{mj} F_m^{(1)} + R_j \tag{6}$$

where: $F_i^{(1)} = \left.\frac{\partial F}{\partial x_i}\right|_{\underline{c}}$

Assume the remainder R_j is essentially constant for all systems j. Note that it does not have to be small for our purposes, but we require that it be essentially constant in the region of interest. Then we define the First Order Ranking Index to be

$$E_j^* = \frac{F(\underline{x}_j) - F(\underline{c}) - R_j}{F_b^{(1)}}$$

$$= \sum_{i=1}^{m} h_{ij} t_i \tag{7}$$

where

$$t_i = F_i^{(1)} \Big/ F_b^{(1)} = \left.\frac{\partial F}{\partial x_i} \Big/ \frac{\partial F}{\partial x_b}\right|_{\underline{c}} = -\left.\frac{\partial x_b}{\partial x_i}\right|_{\underline{c}}$$

is defined as the First Order Trade Factor with respect to a suitable base factor x_b for which $F_b^{(1)}$ is not zero.

Now we can rank any two systems without knowing the absolute value of the Effectiveness Function. All we need is the difference in the values of the ranking index E_j^* for the two systems. Let us consider two systems, j=p and j=q, and let

$$\begin{aligned} E_{pq}^* &= E_p^* - E_q^* \\ &= \sum_{i=1}^{m} h_{ip}t_i - \sum_{i=1}^{m} h_{iq}t_i \end{aligned} \tag{8}$$

Note that E_{pq}^* as defined in (8) becomes known when the system data h_{ij} and the trade factors t_i becomes known, but that the actual difference

$$E_{pq} = F(\underline{x}_p) - F(\underline{x}_q) \tag{9}$$

is still unknown. However, knowing E_{pq}^* is sufficient to perform a ranking.

DEFINITION: As a matter of convenience in further discussion we define the smallest value of $E = F(\underline{x})$ to be best, i.e., system p ranks before system q if the "true" difference E_{pq} is negative.

3. SENSITIVITY ANALYSIS (FIRST ORDER)

Since the First Order Trade Factors t_i as developed preceding are now the key to the ranking model, we need to assess the sensitivity of the ranking to the possible errors in their estimation. Assume

$$t_i = \tau_i + \epsilon_i \tag{10}$$

where τ_i = "true" value of trade factor i

ϵ_i = "error" in estimating τ_i.

Then from (8) and (10) we get

$$E_{pq}^* = E_{pq}' + E_{pq}'' \tag{11}$$

where

$$E_{pq}' = \sum_{i=1}^{m} h_{ip}\tau_i - \sum_{i=1}^{m} h_{iq}\tau_i$$

= "true" difference in system "scores."

and

$$E''_{pq} = \sum_{i=1}^{m} h_{ip}\epsilon_i - \sum_{i=1}^{m} h_{iq}\epsilon_i$$

= "error" difference in system "scores."

Then from (11)

$$\begin{aligned} E'_{pq} &= E^*_{pq} - E''_{pq} \\ &\leq E^*_{pq} + |E''_{pq}| \\ &\leq E^*_{pq} + \sum_{i=1}^{m} |h_{ip} - h_{iq}|\,|\epsilon_i| . \end{aligned} \tag{12}$$

Hence for the "true" difference E'_{pq}

$$E'_{pq} < 0 \quad \text{if} \tag{13}$$

$$E^*_{pq} + \sum_{i=1}^{m} |h_{ip} - h_{iq}|\,|\epsilon_i| < 0 \;, \tag{14}$$

i.e., if

$$k_{pq} = \frac{-E^*_{pq}}{\sum_{i=1}^{m} |h_{ip} - h_{iq}|\,|\epsilon_i|} > 1 \;. \tag{15}$$

Note that the ratio k_{pq} as defined in (15) may be interpreted as the proportion by which all the trade factor errors ϵ_i could grow simultaneously and yet satisfy the inequality in (15), and hence in (14) and (13). Thus if $k_{pq} = 2.0$ for example, then all the errors ϵ_j could double in size and (13) would still hold.

DEFINITION: The difference E^*_{pq} in observed scores is significant whenever the inequalities in (14) and (15) hold. This implies that the inequality (13) holds and that the ranking of system p over system q is significant.

APPENDIX B

COMPARISON OF REFRIGERATION SYSTEMS BY THE EFFECTIVENESS FUNCTION

APPENDIX B

COMPARISON OF REFRIGERATION SYSTEMS BY THE EFFECTIVENESS FUNCTION

The total cost of a refrigeration system for lunar surface and spacecraft applications includes not only the development and construction costs, but also the launch costs to deliver the system to the use location and penalties for volume, maintenance, repair and redundancy. Because launch systems have been weight critical in the early years of manned spaceflight, particular attention has been given to the weight and volume of systems. However, for advanced launch systems (such as the space shuttle) all of the above factors assume significant proportions; thus, the selection of a refrigeration system to use in advanced spacecraft must quantitatively evaluate all system characteristics. An effectiveness function has been developed to provide a systematic method of relating all quantities to a common base. This function, since it is defined in such a manner as to quantitatively assess the relative or absolute importance of all factors in the performance of a specific task, has the monotone property, that is the function either increases with increasing costs and penalties or it decreases. It is not necessary that the function always increase or that it have a particular set of units. (The function could be evaluated in dollars, pounds, per cent, maintenance hours or on a non-dimensional basis); but it is required that each factor be quantitatively related to the others in proportion to its importance in the mission. An analytical discussion of general effectiveness methodology is presented in Appendix A. The application of the method to the development of refrigeration systems for lunar surface and spacecraft applications is discussed in the following sections.

1.0 THE EFFECTIVENESS FUNCTION

The effectiveness function is defined herein to assess the relative costs of competing systems for a particular application. The function is evaluated in dollars so that the system which has the minimum value of its function represents the most effective utilization of resources to perform the heat rejection task for a spacecraft environmental control system (ECS). Inherent in this method is the assumption that development costs and launch costs and any or all combinations of accountable factors can be traded at will against each other (subject to weighting factors, herein identified as trade factors and noted with small letters). This essentially assumes that weights, volumes, etc. of the systems under consideration are never so large as to require a major modification of the launch system or the spacecraft.

The equivalent cost of each accountable factor, i, is evaluated for each system, j, and summed to obtain the total cost as follows:

$$F_j = \sum_{i=1}^{n} x_i X_{ij} = x_1 X_{1j} + x_2 X_{2j} + x_3 X_{3j} + \cdots x_n X_{uj} \quad \text{(Eq. B-1)}$$

$$
\begin{aligned}
F_j = {} & \left(\text{DEVELOPMENT COSTS}\right)_j + \left(\text{CONSTRUCTION COSTS}\right)_j + \left(\text{LAUNCH COSTS}\right)_j \\
& + \left(\text{EQUIVALENT VOLUME PENALTY COSTS}\right)_j + \left(\text{MAINTENANCE COSTS}\right)_j + \left(\text{REDUNDANCY AND SPARES COSTS}\right)_j \\
& + \left(\text{EQUIVALENT COSTS FOR TECHNICAL RISKS}\right)_j + \left(\text{SYSTEM INTEGRATION AND INTERFACE COSTS}\right)_j + \left(\text{EQUIVALENT COST OF OTHER FACTORS}\right)_j
\end{aligned}
$$

where:

F_j = The cost of system j to perform the heat rejection task, in dollars ($)

x_i = The trade factor for accountable factor, i, in \$/LB, \$/Unit, \$/Ft^3, etc. (Trade factors are a function of the mission and the vehicle)

X_{ij} = The value of the ith accountable factor for system j, in LB, Units, Ft^3, etc. (Accountable factors are a function of the individual refrigeration systems)

For convenience, each of the factors are identified by letters rather than subscripts as follows:

$$
\begin{aligned}
F_j = {} & \frac{1}{n}(NR)_j + q(UC)_j + bq\,W_j + bqv\left(V_j - \frac{W_j}{v}\right) \\
& + t\,T_j + fqFWT + \frac{1}{n}\left(TR_j + SI_j + RC_j\right) \qquad \text{(Eq. B-2)}
\end{aligned}
$$

where for each refrigeration system j, the accountable factors are:

NR_j = Non-recurring (development) costs in dollars($)

UC_j = Unit cost in dollars per KW of cooling ($\$/KW_t$)

W_j = System specific weight including penalties for power and radiator panels, in pounds (LB/KW_t)

V_j = System specific volume in cubic feet (FT^3/KW_t)

T_j = Astronaut or crewman repair time in hours (hr)

FWT = The specific system fixed weight excluding penalties for power and radiator area. (A redundant loop is assumed in determining the radiator penalty). In LB/KW_t

TR = Technical risk associated with the development, in dollars ($)

SI = System integration and interface costs associated with the application of system j to a mission, in dollars

RC = Other factors which might be applicable to particular systems, in dollars

And where for all refrigeration systems the trade factors for each mission are:

n = Number of units required

b = Launch costs in $/LB

v = Launch system volume penalty in LB/Ft^3

t = Value of crewman time to performance maintenance in $/hr

f = Redundancy launch cost in $/LB (the redundancy factor times the launch cost, b)

q = The required amount of cooling in $KW_{cooling}$.

Since a bulky or irregularly shaped system limits the volume available for other systems, a penalty is assigned when a systems' density is less than the average spacecraft density (Spacecraft Weight/Spacecraft Volume). The penalty assigned is $(V_j - W_{j/v})$. There is no premium for a high density system and $(V_j - W_{j/v})$ is set to zero in that case.

The total cost for each of m systems under consideration is given by the above function so that there exists a matrix of trade and accountable factors as follows:

$$F = \begin{vmatrix} F_1 \\ F_2 \\ \cdot \\ \cdot \\ \cdot \\ F_m \end{vmatrix} = \begin{vmatrix} \sum_i^n x_i X_{i1} \\ \sum_i^n x_i X_{i2} \\ \cdot \\ \cdot \\ \cdot \\ \sum_i^n x_i X_{im} \end{vmatrix} \qquad \text{(Eq. B-3)}$$

The most preferred system in the matrix is the one with the minimum value of its effectiveness function and it can be easily discerned by a simple comparison.

Preliminary studies have shown that the relative importance of repair time, integration costs, and other subjective factors are approximately equal in all systems and are not major contributors to the total cost. The size of the matrix was reduced by assuming that the above factors were equal and defining the ranking function, E_j, (which is identified as the first order ranking index, E_j^*, in Appendix C) as follows:

$$E_j = \frac{F_j - C_1}{C_2} = \frac{F_j - (t\ T_j + SI + RC)}{\$\ 10^6} \qquad \text{(Eq. B-4)}$$

where:

E_j = The ranking function, in millions of dollars

F_j = The effectiveness function in dollars

C_1 = Subjective factors which are assumed to be equal in all systems (such as repair time, weights, interface costs, etc.)

C_2 = One million dollars ($\$10^6$). A constant to reduce the magnitude of numbers

By employing the constant C_2 at 10^6 dollars, the trade factors must be assigned in reciprocal millions of dollars. The ranking function is employed to calculate the relative cost of providing heat rejection for one particular mission. Since the missions will be conducted in the future, the operating conditions are necessarily uncertain. The effects of these uncertainties upon the results must be evaluated before firm conclusions can be drawn. A sensitivity analysis was performed and is discussed in the paragraphs below.

The ranking function as described above will indicate the preferred system for the particular set of trade and accountable factors. However, uncertainties exist in both of these factors j, therefore, by taking derivatives with respect to each factor, k, the sensitivity can be calculated as follows:

$$\Delta E_j = \frac{\delta E_j}{\delta K} \Delta K \qquad \text{(Eq. B-5)}$$

where:

ΔE_j = The change in the ranking function

K = A trade or accountable factor

ΔK = The uncertainty in the trade factor or accountable factor

For this study, only two quantities: the thermal load and the power penalty will be allowed to vary. Subdividing the unit cost into structure and power penalty at 100 \$/LB and fixed weight at 600 \$/LB, the unit cost is given by the following:

$$UC_j = (100\ \$/LB\ [W_j - FWT] + 600\ FWT)\ \$/KW_c \qquad \text{(Eq. B-6)}$$

$$= (100\ W_j + 500\ FWT)\ \$/KW_c$$

Then the ranking function equation is given as follows:

$$E_j = \frac{1}{n} NR_j + q\ (100\ W_j + 500\ FWT) + bq\ W_j$$

$$+ \ bq\ v\ (V_j - \frac{W_j}{v})^* + fq\ FWT$$

$$= \frac{1}{n} NR_j + q\ [(100 + b)\ W_j + (500 + f)\ FWT \qquad \text{(Eq. B-7)}$$

$$+ \ bv\ (V_j - \frac{W_j}{v})^*]$$

*Negative values are set equal to zero i.e., no premium is allowed for volume which is associated with a launch weight but which is not occupied due to a high system density.

Evaluating the deriveratives with respect to q and W_j yields the following equations for the sensitivity of the ranking function:

$$\triangle E_j(q) = [(100 + b)\ W_j + (500 + f)\ FWT$$
$$+ \ bv\ (V_j - \frac{W_j}{v})] \cdot \triangle q \qquad \text{(Eq. B-8)}$$

and

$$\triangle E_j(W_j) = q\ (100 + b) \cdot \triangle W_j \qquad \text{(Eq. B-9)}$$

and when both occur simultaneously:

$$\triangle E_j\ (q, W_j) = (100 + b) \cdot \triangle W_j \cdot \triangle q \qquad \text{(Eq. B-10)}$$

The above equations are employed using the expected launch penalties and operating conditions presented in Section 2.0 of this Appendix.

2.0 MISSION DEPENDENT PARAMETERS (TRADE FACTORS)

The trade factors given in Equation B-2 (represented by the lower case letters b, q, v, etc.) are primarily a function of the specific mission, and the vehicle and launch system used for accomplishing that mission. This section describes the baseline missions and vehicles considered in this study, and the pertinent characteristics of these systems, such as weight penalty, volume penalty, etc., which establish the trade factors.

The baseline missions along with significant parameters of launch costs, power penalty, radiation sink temperature, and vehicle heat load are given in Table B-1. Several different values of the significant parameters are given for the earth orbital missions; however, only one set of parameters was used for the other missions. This approach was taken because there are several specific launch systems and vehicles which can currently be identified for the earth orbital mission, but for the advanced missions there are no definite launch systems which can be identified.

2.1 Launch Costs

Derivation of launch costs used for each baseline mission is given below.

2.1.1 Earth Orbit

The cost of the Titan/Advanced Orbiting Launch (AOL) system was identified as $729/lb of payload with an average allowable density of 7.5 lb/cu. ft in Reference [27]. This reference also gives a cost of $630/lb of payload for the Saturn/S-IVB with an average allowable density of 12 lb/cu. ft. Reference [28] shows launch cost as a function of orbital vehicle weight; the cost is $1000/lb for a 100,000 lb vehicle and $700/lb for a 200,000 lb vehicle. Reference [29] states that the cost of payload to orbit is $1000/lb for the Saturn 5 launch system. Reference [34] gives a Saturn 5 launch cost of $570 million with a 306,000 lb payload put into a 289 n.m. orbit; this is a unit cost of $1800/lb. It is not clear as to what basis is used in each reference for the establishment of unit costs; in particular, it is unclear as to whether the costs include amortization of launch system development costs or not, and it is unclear whether the unit weight includes the entire vehicle which is put into the orbit, or if it only includes the actual useful cargo put into orbit. For the purposes of this study a single value was chosen for a conventional launch system as shown in Table B-1; $1000/lb of useful cargo with an average allowable density of 7.5 lb/cu. ft.

NASA is currently pursuing the development of space shuttle system which would employ a reuseable booster, a reuseable orbiter, and which would deliver cargo to low earth orbit at a cost roughly an order-of-magnitude lower than current costs. Reference [29] states that the goal of the space shuttle system will be to achieve launch costs of $20 - 50/lb. Spieth and Woods presented projected space shuttle launch costs of $200/lb for a 12,500 lb payload shuttle, and $50/lb for a 50,000 lb payload vehicle in Reference [30]. Milton and Schramm presented a projected shuttle operating launch cost of $50/lb and a total amortized launch cost of $170/lb in Reference [31]. All of these costs appear to represent a "one-way" cost in which cargo is transported to earth orbit in the shuttle; but the shuttle returns to earth empty. If there were a

TABLE B-1

MISSION DEPENDENT PARAMETERS

BASELINE MISSIONS	LAUNCH COSTS WEIGHT $/LB	LAUNCH COSTS VOLUME $/CU.FT.	VEHICLE HEAT LOAD BTU/HR	VEHICLE HEAT LOAD KW_t	DESIGN ENVIRONMENT SINK TEMP. °F	RADIATOR LIMITATION (AREA OR ORIENTATION)	POWER PENALTY LB/KW_e	POWER SYSTEM COST $\$/W_e$	CREW SIZE	NO. OF VEHICLES BUILT
Earth Orbit Space Shuttle Orbiter (Based on 100 launches)	5,000	25,000	35,000	10.2	20 (Integral Area) -78 (Deployed Radiator)	900 Sq. Ft. Inside Cargo Bay Doors	300	50	4	10
Space Station (Solar Cell Power) With High Cost Shuttle Launch Vehicle	200	472	120,000	35	20 (Integral Area) -78 (Deployed Radiator)	2500 Sq. Ft. Integral area	500	50-150	12	10
Space Base or Station With Saturn V Launch	1,000	7,500	260,000 ± 120,000	76 ± 41	0	None	300 500 700	50	60(Base)	10
Low Cost Shuttle Launch	50	236								
High Cost Shuttle Launch	200	472								
Lunar Orbit Station (Medium Cost Shuttle to Earth Orbit and Nuclear Tug to Lunar Orbit)	500	2,500	120,000	35	100	None	700	50	12	5
Lunar Base (Carried to Lunar Surface by Reusable Vehicle from Above Lunar Orbit)	1,000	5,000	50,000	14.6	100	None	700	50	6	5
Lunar Surface EVA (Originating from Lunar Base Described Above)	1,000	20,000 (Based on PLSS* Package Density of 20 lb/cu. ft.)	3,500	1	100	None	700	50	1	10
Mars Excursion (Based on Shuttle and Tug on Earth Orbit, Orbital Assembly and Launch)	8,000	40,000	120,000	35	100	None	700	50	12	3

*Apollo Portable Life Support System

payload which could be transported from earth orbit to earth, these costs could be correspondingly reduced. For purposes of this study, the two extremes of the projected shuttle costs were assumed. A 25,000 lb payload shuttle with a total amortized launch cost of $5 million, and a 50,000 lb payload shuttle with a launch cost of $2.5 million were considered; each with an empty return from earth orbit. These numbers give unit costs of $200/lb and $50/lb, respectively. The cargo space for each of these assumed shuttle vehicles was a 15 ft. diameter cylinder which is 60 ft. in length (References [29], [32] and [36]). This gives a total cargo volume of 10,600 cu. ft., or an average allowable density of 2.36 lb/cu. ft. for the high cost shuttle ($200 lb), or 4.72 lb/cu. ft. for the low cost shuttle ($50/lb).

2.1.2 Lunar Orbit

For the lunar orbit mission it is assumed that there is an operational space shuttle system for delivering cargo to earth orbit, and that there is an operational nuclear powered space tug for transferring cargo from earth orbit to lunar orbit. Johnson presented such a projected system in Reference [33]. Assuming a one-way payload to lunar orbit, and a useful life of ten round-trips for the tug, the unit cost to lunar orbit would be $500/lb if it cost $100/lb for placing the cargo in earth orbit, and $850/lb if it cost $200/lb to put the cargo in earth orbit. For this mission, a nominal cost to earth-orbit of $100/lb was assumed, so the total cost to lunar orbit is then $500/lb. A volume penalty of 5 lb/cu. ft. was assumed for this mission.

2.1.3 Lunar Surface Base

An arbitrary transportation cost of $1000/lb was assumed for transporting cargo to the lunar surface. This is based somewhat on the cost to lunar orbit of $500/lb. A volume penalty of 5 lb/cu. ft. was assumed for this mission.

2.1.4 Lunar Surface EVA

The same transportation cost as for the base was assumed, that is, $1000/lb. The volume penalty is based on EVA equipment packaging rather than the launch vehicle. The Portable Life Support System packaging density was used as a baseline. The approximate volume is 4.25 cu. ft. (Reference [37]) and the approximate weight is 85 lb (Reference [38]), for a density of 20 lb/cu. ft.

2.1.5 Mars Excursion

For the Mars Excursion, it was assumed that a space shuttle launch system and a nuclear powered space tug will be operational. It was assumed that an orbital launch system would be employed for the Mars Excursion vehicle. The vehicle would be transmitted to earth orbit piecemeal; there it would be assembled, checked out, and launched. Launch systems such as this are described in References [28], [34] and [35]. The cost per lb for launching a vehicle to Mars was calculated based on information given in Reference [34], which is below:

Mars Vehicle Wt. in Earth Orbit	=	2,158,000 lb
3rd Stage (which escapes from Mars) Wt.	=	624,000 lb
4th Stage (which returns to Earth) Wt.	=	160,000 lb
Vehicle Cost (including development)	=	$1100 million

If a cost of $100/lb is used for transporting the vehicle to Earth-Orbit, and the unit cost is based on the 4th Stage Weight, then the cost per lb to Mars is

$$\text{Cost} = \frac{(2{,}158{,}000 \text{ lb})\ (\#100/\text{lb}) + \$1.1 \times 10^9}{160{,}000 \text{ lb}} = \$8{,}000/\text{lb}$$

This cost is very speculative, but then so is the entire concept of a Manned Mars mission at this time. The volume penalty for this mission was estimated at 5 lb/cu. ft.

2.2 Vehicle Heat Loads and External Environments

Three separate classes of vehicles were considered for the earth orbital missions; the space station; the space base, and the space shuttle. North American Rockwell has estimated the maximum vehicle heat load for a 12 man space station to be 120,000 BTU/hr (Reference [39]), and for a 60 man space base to be 400,000 BTU/hr (Reference [40]). VMSC used 35,000 BTU/hr for the space shuttle orbiter internal heat load in Reference [41]. For the lunar orbit space station a heat load of 120,000 BTU/hr was assumed. The Lunar Base was assumed to have a heat load of 50,000 BTU/hr, the Lunar EVA heat load was assumed to be 10,000 BTU/hr, and the Mars Excursion Vehicle was assumed to have a heat load of 120,000 BTU/hr. These heat loads were based on the vehicle crew size to a certain extent; it was estimated (based on the space station and space base data) that a vehicle with a closed life support system and internal electrical equipment such as Guidance and Navigation will have a maximum heat load of around 10,000 BTU/hr-man. For the EVA, it was assumed that the single crewman could have a high metabolic load (up to 3500 BTU/hr) plus a very significant heat leak from the environment, so a value of 10,000 BTU/hr was used even though the EVA system probably would not have a closed life support system.

The external environment is characterized by a single maximum radiation sink temperature in this work. (Sink temperature is based on surface optical properties of emittance = 0.9 and solar absorptance = 0.3). This sink temperature is used to size the individual systems, as was discussed previously in Section 3.2. For the nuclear powered space station and space base systems in earth orbit, a sink temperature of 0°F was assumed for radiators integral with structure. A conventional radiator system was considered in this case for comparative purposes. The solar cell powered space station in earth orbit was assumed to have a sink temperature of 0°F for integral vehicle surface area, while the deployed radiator system was assumed to have a sink temperature of -78°F (Reference [42]). The integral area available for radiators is limited to 2500 sq. ft. For this vehicle, a vapor compression refrigeration system with integral radiators is compared to a conventional radiator system using deployed panels. The same comparison was made for a space shuttle system in orbit. For the shuttle, the integral radiators are assumed to be on the inside of the cargo bay doors, and to be limited to 900 sq. ft. (Reference [41]). The sink temperature is 20°F for integral radiators, and -78°F for deployed radiators. The lunar orbit space station was assumed to have a sink temperature of 20°F. For the lunar surface base and lunar surface EVA, a sink temperature of 100°F was assumed. As described in Reference [37] this represents an element with $\epsilon = 0.9$ and $\alpha = 0.3$ facing up in a 10:1 (diameter to depth ratio) lunar crater on the lunar equator at lunar noon. For the Mars mission, a maximum sink temperature of 100°F was assumed based on a spinning cylinder at a distance of 0.8 AU from the sun. This distance represents the closest to the sun that a vehicle in an opposition mission to Mars would come (Reference [41]).

2.3 Power Penalty (Weight)

The specification in the contract (Reference [43]) under which this work was accomplished calls for a power penalty of 600 lb/KW_e for regulated d.c., and 700 lb/KW_e for regulated a.c. These values are similar to the North American Rockwell (NAR) Space Station Study results (References [39] and [40]) for nuclear power systems. The consideration of nuclear power systems is consistent with the study objective of using waste heat to drive the heat rejection system. The NAR space station studies project solar cell power system weight at a weight of 350 lb/KW_e. Borentz (Reference [44]) gives a solar cell power system weight of 325 lb/KW_e, for a silicone cell array. Borentz also gives weights for Isotope power systems (in the 20 KW_e size range) of 375 lb/KW_e down to 340 lb/KW_e, for Reactor Thermoelectric systems of 1025 lb/KW_e, and for Reactor Rankine Cycles of 680 lb/KW_e to 900 lb/KW_e. Barker and Nicol (Reference [45]) recommended a power penalty of 500 lb/KW_e in a recent paper on spacecraft thermal control systems. Gaddis et al (Reference [41]) have recommended a Fuel Cell Weight Penalty (FCWP) of

$$\text{FCWP} = 120\ \text{LB/HP} + 0.8\ \text{LB/HR-HP}\ [\text{HRS OF OPERATION}]$$

For a space shuttle on a 30 day mission with 5 days of refrigeration system operation, this penalty becomes

$$\begin{aligned}\text{FCWP} &= 120\ \text{LB/HP} + 0.8\ \text{LB/HP-HR}\ [5\ \text{DAYS}]\ [24\ \text{HRS/DAY})\\ &= 120\ \text{LB/HP} + 96\ \text{LB/HP} = 216\ \text{LB/HP}\\ &= (216\ \text{LB/HP})\ (1.34\ \text{HP/KW}) = 290\ \text{LB/KW}\end{aligned}$$

This number is based on the Apollo Fuel Cells; NASA is attempting to develop a fuel cell that has a fixed weight of 45 LB/HP for the space shuttle (as opposed to 120 LB/HP) according to Reference [51], and so 290 LB/KW should give an adequate margin for radiators and plumbing.

For purposes of this study, weight penalties of 300 LB/KW_e, 500 LB/KW_e, and 700 LB/KW_e were all considered for the earth orbital space station and base. For the space shuttle a penalty of 300 LB/KW_e was used. The solar cell powered space station was assumed to have a power penalty of 350 LB/KW_e. All other vehicles were assumed to have power at a penalty of 700 LB/KW_e.

2.4 Maintainability and Reliability

Maintainability and reliability for the purposes of this work resolve down to the amount of crew time necessary to repair the system, plus the amount of redundancy which must be designed into the system and the weight of spare components which must be carried. Yakut et al (Reference [49]) presented the projections given in Table B-2 which estimate the weight of spares which must be carried on a mission to achieve the indicated reliability. Jennings (Reference [50]) presented results of a study on "Maintainability and Reliability of Environmental Control/Life Support Systems", which estimates the time required for unscheduled maintainance of an entire regenerable EC/LSS as 10 minutes per day, and for scheduled maintainance at 36 minutes per day.

The assumptions made for this study were based on the above mentioned work. The approach is to provide redundant systems plus spares and to account for crew time required for scheduled and unscheduled maintenance. Table B-3 presents the redundancy and spares considered for each mission. The systems are assumed to have a parallel redundant system with cross-over capability plus spares. The scheduled and unscheduled maintenance time were estimated as 5 minutes and 15 minutes per day, respectively, for all missions except lunar surface EVA. Preliminary results indicated that maintenance time was not a significant parameter in system selection; and thus more accurate figures for each system were not developed.

2.5 Crewman Time Cost

The cost of crewman time can be estimated in several ways, all of which may be debatable. For the purposes of this study, the cost of crewman time is taken as the mission total cost divided by the number of crewman hours available for useful work. Establishing either of these numbers is difficult; and the latter is particularly difficult on missions with small crews and a large amount of housekeeping tasks, such as the Apollo flights. It could also be argued that, at times, the Apollo crew had nothing much to do. Determining the cost basis is also problematical because of questions such as whether or not vehicle development costs should be included. On Apollo, the total development cost of over $20 billion could be used along with the roughly 20 hours of crew time which have been spent in EVA on the moon to date; for an Apollo crewman EVA time cost on the lunar surface of $1 billion/hr.

In this work, the useful crew time is taken as 75% of the total available based on an 8 hour day, or 6 hr/day per crewman. The value for Crewman Time for the various missions is given in Table B-4, which also shows mission duration, mission cost, and crew size. Vehicle development costs have been excluded. These results are very primitive, and give only a rough approximation of the value of crewman time. The assumption is also made that there be sufficient useful work (other than in maintaining the vehicle) required to expend 1.3 million manhours over a ten year period, as in the case of Earth Orbit Space Base.

2.6 Trade Factor Summary

Table B-5 presents a summary of the specific Trade Factors used in this work, based on the preceding discussion.

TABLE B-2

SPARES FOR THERMAL CONTROL
SUBSYSTEM (FROM REFERENCE [49])

	MISSION DURATION, DAYS				
	90	180	400	800	2000
All Systems Independent of Crew Size	0.5(0.9) 1.8(0.9999)		1.13(0.9) 2.74(0.9999)		2.2-(0.9) 4.72(0.9999)

NOTE: Pairs of points denote ratio of spares weight to subsystem Weight Corresponding to subsystem reliability in parentheses.

TABLE B-3

REFRIGERATION SYSTEM REDUNDANCY
AND SPARES REQUIREMENT

MISSION	REDUNDANCY	SPARES (NOT INCLUDING RADIATOR)
Earth Orbit	100%	25%
Lunar Orbit	100%	25%
Lunar Surface Base	100%	30%
Lunar Surface EVA	50% for Emergency System	
Mars Excursion	200% (Except 100% on Radiator)	50%

TABLE B-4

CREWMEN TIME COST

MISSION	MISSION DURATION DAYS	CREW SIZE	USEFUL CREWMAN TIME AVAIL. HRS. (6 HRS PER DAY)	RESUPPLY INTERVAL DAYS	LAUNCH COST $ MILLION	VEHICLE* COST $ MILLION	RESUPPLY** COST $ MILLION	TOTAL COST $ MILLION	CREWMAN TIME COST, $/HR
EARTH ORBIT									
Space Station ($1000/lb Launch, Cost)	90	12	6,500	-	100	100	-	200	31,000
Space Station ($1000/lb Launch Cost)	3600	12	260,000	180	20	100	285	405	1,550
Space Base (200/lb Launch Cost)	3600	60	1,300,000	180	50	500	1,300	1,850	1,400
LUNAR ORBIT STATION									
($600/lb Launch Cost)	360	12	26,000	-	100	200	-	300	11,500
LUNAR SURFACE BASE									
($1000/lb Launch Cost)	360	6	13,000	-	100	100	-	200	15,400
LUNAR SURFACE EVA	Same as Lunar Surface Base								
MARS EXCURSION	360	12	26,000	-	200	1,100	-	13,000	50,000

* Based on CSM cost of $75,000,000 ea - Development cost not included
** Based on launch cost plus supplies cost

TABLE B-5

TRADE FACTORS (LOWER CASE LETTERS IN EQ. B-2)

	NO. OF UNITS n DIMENSIONLESS	LAUNCH COSTS, b $/LB x 10^{-3}	VOLUME PENALTY, v LB/FT^3	CREWMAN TIME COST, t $/HR	REDUNDANCY FACTOR, (f) (b) (REDUNDANCY IN TABLE B-3)xb/100% $/LB	THERMAL LOAD q KW_t
Earth Orbit						
Space Shuttle (100 launches)	10	5	5		250	10.2
Space Station	10	0.2	2.36	14,000	250	35
Space Base (launch system)						
Saturn V	10	1	7.5	31,000	1,250	76 ± 41
Low Cost Shuttle ($50/lb)	10	0.05	2.36	1,550	63	
High Cost Shuttle (200/lb)	10	0.2	2.36	1,400	250	
Lunar Orbit	5	0.5	5	11,500	625	35
Lunar Base	5	1	5	15,400	1,250	14.6
Lunar Base EVA	10	1	20	15,400	500	$1 {}^{+2}_{-0}$
Transmartian	3	8	5	50,000	10,000	35

3.0 REFRIGERATION SYSTEM (ACCOUNTABLE FACTORS)

The Accountable Factors, which are represented in Equation B-2 by capital letters (V, NR, UC, W, T, etc) represent such factors as refrigeration system cost, weight, and required crewman time for repairs. These accountable factors are related to the individual refrigeration system, rather than being related to the mission or the vehicle. Of course, the individual refrigeration systems may have different characteristics for the different missions. The accountable factors used in this study are discussed in the following sections.

3.1 Refrigeration System Weight - W

The optimum refrigeration system fixed weight was found to be a linear function of heat load regardless of mission (for missions considered). The fixed weights used were:

(1) Vapor Compression (R12): 7.6 lb/KW_t

(2) Vapor Absorption (R22/E181): 24 lb/KW_t

(3) Vapor Absorption (LiBr/H_2O): 35 lb/KW_t

(4) Vapor Absorption (LiBr/H_2O-Turbine/Compressor): 40 lb/KW_t

(5) Vapor Adsorption (H_2O/Zeolite): 45 lb/KW_t

(6) Radiator System (R21): 5.6 lb/KW_t

The radiator weight was based on the heat load of the system, the optimum radiator temperature given in the RSPLAT run (Appendix C) and the radiator weight penalty assumed for the particular mission.

The RSPLAT routine is discussed in Appendix C, with the equations used in the routine being given in Appendix C and the results of all runs used being given in Appendix D. The refrigeration systems were optimized for minimum weight based on radiator (i.e., condenser) temperature for each mission except in the case of the area limited radiators for the space shuttle and space station where the minimum radiator temperature high enough to obtain the desired heat rejection was used. The required power system weight is the difference between the total system weight and the sum of the fixed weight and radiator weight for each system. It was necessary to establish the fixed weight, radiator weight and power system weight of each refrigeration system because the costing technique evolved in the next section required that each of these be known.

3.2 Development and Unit Costs - NR and UC

3.2.1 Mechanical Component Costs

The NASA-MSC recently reported non-recurring and recurring costs for spacecraft and aircraft thermal control systems in Reference [52] are shown in Table B-6.

TABLE B-6 AEROSPACE ECS COSTS

	NON-RECURRING COST $ MILLION	NON-RECURRING COST $/lb	RECURRING COST $ MILLION	RECURRING COST $/lb
Apollo ECS (LM + CM)	100	100,000*	7.5	7,500*
Gemini	34	65,000*	1.65	3,150*
Mascot "90"***	120		7.5	
B-70	12.48	2,270**	1.6	291**
X-15	1.46		Unknown	

* Weight from Reference [46]: 980 lb total for LM and CSM ECS; 523 lb for Gemini

** Weight from Reference [47]: 5500 lb (Dry) for the entire ECS

*** From Reference [52]: For the Apollo Program

In 1967, Mandell (Reference [46]) reported the Spacecraft ECS costs given in Table B-7.

TABLE B-7 SPACECRAFT ECS COSTS

	LM	CSM	Gemini	Mercury
Weight, lb	380	600	523	135
Non-Recurring Cost $ Million	15	75	13	-
1st Item Cost $ Million	1.7	1.18	1.65	-
Non-Recurring Cost $/lb	39,500	124,000	25,000	50,000
1st Item Cost $/lb	4,500	2,000	3,150	1,000

Based on the cost of 10 flight and 7 training Portable Life Support Systems (PLSS) in Reference [48], and weights given in Reference [38], the PLSS unit cost is $10,000/lb. Mandell recommends a graphical costing procedure in Reference [46] which uses the cost per pound of a specific type of component as a basis. This approach is used throughout industry for estimating costs of future systems based on previous experience, and it can be highly accurate under ideal circumstances. The early spacecraft ECS components which were developed are probably a typical for several reasons; (1) the techniques used for accomplishing environmental control had been little used previously, (2) the limited weight capabilities of the launch systems in use required unusual optimization of component design, (3) expensive component and system qualification testing requirements were imposed by safety considerations, (4) components and systems were required to function successfully for relatively long periods without inspection or maintenance, and (5) the vehicles were used only once, and each design was used for only a few flights, so there is little opportunity for gradually uprating the vehicle capabilities. It is believed that future spacecraft development and unit costs will more nearly parallel those of advanced aircraft than the early spacecraft. The B-70 air vehicle, developed in the late 1950's, contained a vapor compression refrigeration system. This was the first high performance aircraft to contain a vapor cycle refrigeration system to the author's knowledge. The cost of an advanced spacecraft, or a shuttle Orbiter Aerospace Vehicle, should parallel the B-70 ECS costs much more closely than spacecraft ECS costs, in the author's opinion. The restrictions on the NASA budget will probably be reflected in a reduction in development costs, and perhaps in an increase in recurring costs. The development of lower cost launch systems will reinforce this trend since heavier components will be permissible, thus reducing the need to optimize component weight, increasing component inherent reliability, and reducing the need for extensive testing. Krantweiss (Reference [47]) has given the data on the B-70 vapor compression refrigeration system shown in Table B-8.

TABLE B-8 B-70 AIR CONDITIONING SYSTEM

Working Fluid	=	Freon 11	
Evaporator Heat Load	=	12.5 Tons	
Power Required	=	106 Shp.	
Refrigeration System Package Wt.	=	610 lb (including a 195 lb Frame)	
Turbine Driven Compressor Wt.	=	91 lb	
Component Costs:			
Refrigeration Package	=	$255,400	
Turbine - Driven Compressor	=	$ 43,500	
Unit Costs:		1960 $'s	1970 $'s*
Refrigeration Package $/lb	=	418	620
Turbine Driven Compressor $/lb	=	480	710

* Assumes 4% per year inflation

By way of comparison, a 15 ton Chrysler Air-Temp railroad car air conditioning unit costs $1600 (not including air ducts or installation) and weighs 1100 lb, for a unit cost of $1.46/lb. The compressor costs $1.75/lb. The rule of thumb for cost of an installed commercial air conditioning unit is about $2/lb. The authors believe that the B-70 refrigeration system unit cost of $620/lb (in 1970 dollars) is more representative of the costs to be expected for future spacecraft thermal control system (TCS) components than early spacecraft TCS components. Similarly, the development costs should be more in line with the $3360/lb (in 1970 dollars: $2270/lb in 1960 dollars) than previous spacecraft development. Thus, a component recurring cost of $600/lb has been assumed for this work. The development cost of the vapor compression system is estimated at $6000/lb (which is consistent with the order of magnitude greater recurring cost over non-recurring costs indicated in Tables B-6, B-7, and B-8). It is assumed for this work that the development cost of a refrigeration unit would be about the same for any capacity unit over the range of capacities being considered (35,000 BTU/hr - 400,000 BTU/hr). This 3 to 30 ton range represents neither extremely small nor extremely large units. The specific costs for each system assumed are given in Table B-9.

TABLE B-9 REFRIGERATION SYSTEM DEVELOPMENT COSTS

	DEVELOPMENT COST ($ MILLION)	APPROX.* WEIGHT LBS	UNIT DEVELOPMENT COST ($/LB)
Vapor Compression	3.5	580	$6000
Vapor Absorption (R22/E181)	5.5	1830	3000
Vapor Absorption (LiBr/H_2O)	4.5	2660	1700
Vapor Absorption (LiBr/H_2O + Turbine/Compressor)	7.0	3040	2300
Vapor Adsorption (H_2O/Zeolite)	4.5	3440	1300

*Based on a 260,000 BTU/HR (76 $KW_{cooling}$) system

The heat-driven refrigeration systems have been assigned lower development costs on a per pound basis because they tend to be heavier and to have fewer rotating parts such as the compressor. The LiBr/H_2O vapor absorption system with the turbine/compressor has rotating parts, and so has the highest development costs. The R22/E181 system has a relatively higher cost because it requires development of a direct absorber radiator (for the minimum weight system considered). Development costs are considered to be amortized over the rather arbitrary number of vehicles shown in Table B-1.

All of the refrigeration systems require a radiator for heat rejection. The radiator cost is considered separately for each system. The costs assumed are:

Simple radiator system development: $2 Million
Condensing radiator system development: $2.5 Million
Radiator recurring cost: $100/lb (This is based on aircraft fabrication costs of $100/lb-see Reference [36]).

3.2.2 Power Costs

Systems which require a large amount of power for operation relative to the power capacity of the entire vehicle, as some of the refrigeration systems do, should be charged with the appropriate portion of the power system equipment cost. In the units which have only small power requirements, this may not be true since power systems usually are packaged in a modular fashion, and so some excess capacity is usually available. In this study, all refrigeration systems with a power requirement are charged with a recurring cost penalty as well as with a weight penalty. The power system development cost is not charged to the individual systems. The recurring costs for typical spacecraft power systems (not including launch costs) are given in Table B-10. The additional recurring costs are very speculative. The solar cell system cost includes a deployment mechanism, structure, and a propulsion system with propellant for one year, in addition to the solar cells. The propulsion system is required for station keeping to off-set drag on the solar cells which tend to slow down the vehicle, and as a result to change the vehicle's orbit. It is estimated that only a fraction of the original cost will be required to enlarge the power system. This may not be true after some critical size of the solar cell power system is reached. The add-on costs for the nuclear system are estimated to be much closer to the original cost because such a large portion of the cost is represented by fuel cost (Reference [44]). Because of the uncertainty in these numbers, various values from $\$50/W_e$ to $\$350/W_e$ were considered.

TABLE B-10

POWER SYSTEM RECURRING COSTS	RECURRING COST* (NOT INCLUDING LAUNCH COST) $/WATT	RECURRING COST PENALTY FOR INCREASING POWER SYSTEM SIZE $/WATT
Solar Cells (Silicon Cells)	350	150
Solar Cells (C_dS Thin Film)	125	50
Isotope Brayton (P_u238 Heat Source)	1800	1750
Radioisotope Thermoelectric	58	50
Reactor Rankine	42	40

* From Reference [44]

3.3 Refrigeration System Maintenance Time Requirements (T)

Refrigeration system maintenance times were discussed in Section 2.4 of this Appendix. It was assumed that the scheduled and unscheduled repair times for all systems are essentially the same. For the earth orbital missions the total cost of repair time for each system in a long duration mission would be:

$$(1/3 \text{ hr/day})\ (360 \text{ day/yr})\ (10 \text{ yr})\ (\$1550/\text{hr}) = \$186{,}000$$

A variance of $\pm$ 50% on this value would not be highly significant compared to the total system costs; and therefore, no attempt was made to further refine the estimates of required maintenance time for each of the individual systems.

3.4 Refrigeration System Volume - V

The volume of the refrigeration systems were estimated as follows:

(a) Fixed Volume

The volume of the refrigeration system hardware was based on the somewhat arbitrary packaged density of 12 lb/ft^3.

(b) Radiator Volume

The volume of the required R21 radiator system was based on a density of 0.016 cu. ft. per sq. ft. of radiator area. For the R11 condensing radiator the volume was 0.032 cu. ft. per sq. ft. These densities are based on 3/16" and 3/8" diameter tubes, respectively.

(c) Power System Volume

The power system density was estimated as 48.3 cu. ft./KW_e. This is based on the density of a fuel cell power system.

3.5 Technical Risk - TR

The technical risk for each system was assigned as a percentage of total system weight. The numbers used, which were based on judgment of complexity and development experience with similar systems, were:

Vapor Compression: 15%
R22/E181 Absorption: 50%
LiBr/H_2O Absorption: 50%
LiBr/H_2O Absorption With a Turbine/Compressor: 60%
Vapor Adsorption: 50%

3.6 Waste Heat Cost

There was no penalty associated with heat driven refrigeration systems for the cost of heat. There would be such a cost in an actual system for:

(1) Plumbing and heat transfer equipment required to transport heat from the power generation equipment to the refrigeration equipment.

(2) Degradation in power system performance if waste heat were required at a temperature above the optimum heat rejection temperature, which is around 250°F. (For example, waste heat at 600°F supplied to an adsorption refrigeration machine would greatly reduce the thermodynamic efficiency of a Brayton Cycle Power System).

(3) Higher integration costs associated with interfacing a heat-driven machine with the power system heat rejection system.

3.7 Vehicle Integration Project and Administrative Costs - SI

It has been assumed that the integration costs for all refrigeration systems would be the same. This is probably a good assumption, except for the heat-driven refrigeration machine interface with the heat rejection system of the power plant. The project and administrative costs are assumed to be the same for all systems. That is, the time thus spent by NASA and support contractor personnel would be about the same for any system.

3.8 Accountable Factor Summary

The Accountable Factors considered in this study are summarized in Table B-11.

TABLE B-11

ACCOUNTABLE FACTORS

SYSTEM	NON-RECURRING** COSTS NR	FABRICATION UNIT** COSTS, UC	SPECIFIC WEIGHT, W	SPECIFIC** VOLUME, V	UNIT WEIGHT ENCL. POWER AND RADIATOR AREA	TECHNICAL RISK, TR
	$ MILLION	$/$KW_c$	LB/KW_c	FT^3/KW_c	LB/KW_c	%
Vapor Compression (R12)	3.5	4,560	Obtained From RSPLAT Run for Each Mission Along with Power Reqm't	91	7.6	15
Vapor Absorption (R22/ E-181)	5.5	14,400		288	24	50
Vapor Absorption (LiBr/ H_2O)	4.5	21,000		429	35	50
Vapor Absorption (LiBr/H_2O - Turb/Comp.)	7.0	24,000		480	40	60
Vapor Adsorption (H_2O/ Zeolite)	4.5	27,000		540	45	50
Pumped Fluid Radiator (R 21)	2	560		0.016 $\frac{FT^3}{FT^2}$ of Radiator Area	5.6	0
Power	Not Considered	See Table B-10		$48.3 FT^3/KW_e$	See Table B-1	

* Crewman Time, T; System Integration, SI; and other factors, RC were assumed to be similar for all systems

** Power and Radiator not included in Refrigeration System

4.0 EXAMPLE TRADE STUDY

An example ranking of a vapor compression heat rejection system with a deployed radiator system in a space station, area limited application is presented herein to clarify the application of the trade and accountable factors in the ranking function. A brief discussion of the procedures employed in estimating each factor and of the design philosophy is included.

A vapor compression refrigerator with a condensing radiator is compared with a deployed radiator system for an earth orbiting space station. Studies of the space station prototype, SSP, (Reference [42] have shown that a deployed system will be required because the available integral vehicle area is too limited with the effective sink temperatures in near earth orbit.

The operating environments for the two systems are assumed to be different, since the deployed radiator may be oriented so that it always faces a cold sink (no vehicle orientation penalty is incurred since the position of the space station is already constrained by the solar cell power generation system). Although it is recognized that the environment is characterized by basically a transient condition with a distributed sink temperature, steady state conditions are assumed to apply, with only one sink temperature. This assumption will penalize the vapor compression system; however, the uncertainties in the effective sink temperature of integral area and in its availability are sufficiently large that a conservative assumption is justified. The effective operating conditions for the two systems which are assumed to be applicable are as follows:

	VAPOR COMPRESSION	DEPLOYED RADIATOR
TSINK	20°F	-78°F
AREA	2500 FT^2	Not Limited
FEPP	500 lb/KW_e	700 lb/KW_e
FRAP	0.6 lb/FT^2	1.5 lb/FT^2
TEVAP	35°F	35°F(T_{AVG} = 45°F)

Since solar cell power incurs a penalty of 700 lb/KW_e for full orbit (with shadow) but only 300 lb/KW_e for sun side only power, the power penalty was set at 500 lb/KW_e for vapor compression due to the fact that active use is required for only part of the dark portion of the orbit (assuming a shadowed orbit). One redundant radiator loop is included in the radiator penalty but the fin weight is assumed to be existing structure for integral area. A minimum temperature of 35°F for both systems is required; the fact that the heat load exists at temperatures above 35°F is utilized with the deployed radiator system and results in an average heat rejection temperature of 45°F. However, a single evaporator and compressor are assumed for the vapor compression so that the thermodynamic advantage of a distributed heat source is not utilized. Obviously, the vapor compression system has not been optimized, but the operating conditions are at least representative of a space station application. (The intent of this study is to select a system for development for a future application. A detailed optimization for a particular application cannot be performed until all factors in the trade as well as the implications and effects on other systems are known).

The above conditions were used to evaluate the weight of each system with the RSPLAT data in Appendix C. The weights are as follows :

	TOTAL WEIGHT (W_j)	FIXED WEIGHT (FWT)
VAPOR COMPRESSION	135 lb/KW_c	7.6 lb/KW_c
DEPLOYED RADIATORS	85 lb/KW_c	11.2 lb/KW_c

The fixed weights were calculated from individual system analysis (References [18] and [53]), and are not available from RSPLAT.

The volume attributal to a heat rejection system is the sum of the volume occupied by the fixed components (pumps, compressors, motors, controls, lines, etc), the radiators, and the power system penalty. The specific volume, V_j, is given by the following equation:

$$V_j = \frac{FWT}{12\ LB/FT^3} + (DR)\ .\ (AR) + \frac{48.3\ ft^3}{EPUF\ KW} \qquad \text{(B-11)}$$

where:

V_j = The specific volume in FT^3/KW_c

FWT = The fixed weight in lb/KW_c assumed to be installed at an average packing density of 12 LB/FT^3

DR = The diameter of the radiator tubes, 3/16 inch for deployed radiators and 3/8 inch for condensing radiators

AR = The area required or available per unit thermal load in FT^2/KW_c [71 ft^2/KW_t (20°F sink) for V-C radiator, 49.3 FT^2/KW_c (-78°F sink) for deployed radiators].

EPUF = The electrical power utilization factor in KW_c/KW_e (59 for radiator, COP = 6.6 for vapor compression).

The equivalent volumes for the two systems are as follows:

$$V_j(V\text{-}C) = \left[\frac{7.6}{12} + \frac{(3)}{(8)(12)}\ \frac{(71)}{1} + \frac{48.3}{6.6}\right] \frac{FT^3}{KW_t}$$

$$= 10.17\ FT^3/KW_c$$

$$V_j(D\text{-}R) = \left[\frac{11.2}{12} + \left(\frac{3}{16}\right)\left(\frac{1}{(12)}\right) \frac{(49.3)}{1} + \frac{48.3}{59}\right] \frac{FT^3}{KW_t}$$

$$= 2.52\ FT^3/KW_c$$

The above accountable factors complete the data required to describe the system as launched. However, since the number of missions is small, the development costs are significant and were estimated in Section 3.2 of this Appendix, as follows:

VAPOR COMPRESSION	NR
CONDENSING RADIATOR	2.5 MILLION $
COMPRESSOR, EVAPORATOR, CONTROLS	3.5 MILLION $
TOTAL	6.0 MILLION $
DEPLOYED RADIATOR	
CONVENTIONAL RADIATOR	2.0 MILLION $
FLUID SWIVELS, SUN SENSORS, CONTROLS	2.0 MILLION $
TOTAL	4.0 MILLION $

The trade factors must be assigned considering the number of and requirements of missions, construction costs, and redundancy as well as the launch system. For this example, it is assumed that three missions with a heat load of 35.2 KW_c are to be conducted. Redundancy is set at 125% of the fixed weight with one redundant radiator loop (this loop has been included in the radiator penalty factor). No redundancy for power is assigned since this factor is included in evaluating the power penalty factor. Construction costs were set at 100 $/LB for structure and power but 600 $/LB for the valves, pumps, compressors, controls, etc., in the fixed weight. The launch system is assumed to be a shuttle with a launch penalty of 200 $/LB and a volume penalty of 4.72 LB/FT^3. The trade factors required in Equation B-7 are summarized as follows:

TRADE FACTOR	VALUE*
Number of missions, n	3
System Size, q	35.2 KW_c
Launch Cost, b	0.00020 MB/LB
Launch system specific volume v	4.72 $\#/FT^3$
Redundancy Factor, f	1.25b = 0.00025 MB/LB

* One MB is a Mega Buck ($1,000,000)

These factors are applied with the accountable factors for each system employing Equation B-7 as follows:

$$E_j \quad = \quad \frac{1}{n} NR_j + q \, [(0.0001 + b) \, W_j + (0.00050 + f) \, FWT_j$$

$$+ \quad bv \, (V_j - \frac{W_j}{v})]$$

<u>For the Vapor Compression System</u>

$$E(V\text{-}C) = \quad \frac{1}{3}(6) + 35.2 \, [(0.00010 + 0.00020) \, 135$$

$$+ (0.00050 + 0.00025) \, 7.6$$

$$+ (0.00020)(4.72)(10.17 - \frac{135}{4.72})]$$

$$= \quad 2.0 + 35.2 \, [(0.00030) \, 135 + (0.00075) \, 7.6$$

$$+ (0.000944)(\cancel{10.17 - - - - 28.5})^{*}]$$

$$E(V\text{-}C) = \quad 3.62 \text{ MB}$$

*This term set equal to zero since no premium is allowed for volume not used.

<u>For Deployed Radiators</u>

$$E(D\text{-}R) = \quad \frac{1}{3} (4) + 35.2 \, [0.00010 + 0.00020) \, 85$$

$$+ (0.00050 + 0.00025) \, 11.2$$

$$+ (0.00020) \, (4.72) \, (2.52 - \frac{85}{4.72})]$$

$$= \quad 1.333 + 35.2 \, [(0.00030) 85 + 0.00075) \, 11.2$$

$$+ 0.000944 \, (\cancel{2.52 - - - - 17.96})^{*}]$$

$$E(D\text{-}R) = \quad 2.53 \text{ MB}$$

For the conditions established for this example, it is apparent that a deployed radiator would be more effective than a vapor compression system employing a limited amount of integral area. Since the ranking function is quantitative, the savings will amount to about one million dollars a mission. Subjective factors (such as interference with a docked or approaching shuttle, or the possible use of an artificial gravity field) have not been considered. Assuming that such factors can be classified as either a procedural problem or a design requirement, those systems which would be affected have previously been eliminated.

The uncertainty in the actual system performance could be significant to the ranking of the systems. Assuming that the technical risk with both systems is + 15%, i.e., that the actual system cost is 15% more than calculated herein, the sensitivity to the ranking function to this uncertainty is given by Equation B-9 as follows:

$$\Delta E_j\ (W_j) = q\ (100 + b)\ \Delta W_j$$

For Vapor Compression

$$E\ (V\text{-}C) = 35.2\ (0.00010 + 0.00020)\ 135\ (0.15)$$

$$= 0.21\ MB$$

For Deployed Radiators

$$E\ (D\text{-}R) = 35.2\ (0.00010 + 0.00020)\ (85)\ (0.15)$$

$$= 0.13\ MB$$

Clearly, for the specified conditions, the uncertainty levels are sufficiently low so that the selection of deployed radiators over vapor compression refrigeration can be made with assurance.

APPENDIX C

REFRIGERATION SYSTEM COMPARISON
PLOT ROUTINE

APPENDIX C

REFRIGERATION SYSTEM COMPARISON PLOT ROUTINE

1.0 METHOD OF SYSTEMS' COMPARISON

The computer routine described in this appendix was written to facilitate optimization of the weight of the candidate refrigeration systems under the various anticipated operating conditions. This routine, called the Refrigeration Systems Plotting and Linearized Analysis Technique (RSPLAT) employs algorithms for calculating the equivalent weight of each candidate system as a function of the operating conditions and penalty factors. The weight estimates which are calculated by this routine are functions of four quantities which can be defined to characterize any probable application:

- TEVAP, evaporator temperature (^{o}F)
- TSINK, sink temperature (^{o}F)
- FEPP, electrical power weight penalty ($\#/KW_{elec}$)
- FRAP, radiator area weight penalty ($\#/FT^2$)

Each refrigeration system is then described by four parameters which are input as functions of the difference between the effective system radiating temperature and the evaporator temperature. These parameters are:

- COP, coefficient of performance
- FWT, fixed system weight (components) ($\#/KW_{cooling}$)
- EPUF, electrical power utilization factor ($KW_{cooling}/KW_{elec}$)
- EFF, ratio of actual system COP to Carnot COP between the same temperatures (work driven systems only).

The output of this routine is a set of computer plotted curves. Total system weight (including assigned penalties) is computed and plotted as a function of radiator temperature (TRD). For a single value of radiator weight penalty, individual curves are computed for every refrigeration system. So that, for each value of FRAP, a complete set of curves are generated per grid. Each grid output by the program then represents an application characterized by the values of TEVAP, TSINK, and FEPP.

2.0 COMPUTER PROGRAM DESCRIPTION

The total system weight (WT) calculated for each system at any particular set of environment conditions is the sum of three terms. These include the weight quantities contributed by the electrical power penalty (EPPT), the radiator area penalty (FPT), and the fixed weight quantity (FWT).

$$WT = EPPT + FPT + FWT \qquad (LB/KW_t)$$

Each of these terms is computed using the four parameters that describe the system. The electrical power term is found by:

$$EPPT = FEPP \left(\frac{1}{EPUF}\right) \qquad (lb/KW_t)$$

where EPUF is the dimensionless ratio of heat load removed by the system to the total electrical power required. The radiator penalty is given by:

$$RPT = \left(1 + \frac{1}{COP}\right)\left(\frac{FRAP}{QREJ}\right) \qquad (lb/KW_t)$$

where heat rejected through the radiator per unit area is:

$$QREJ = .0000452 \left[\left(\frac{TRAD + 460}{100}\right)^4 - \left(\frac{TSINK + 460}{100}\right)^4\right] \left(\frac{KW}{ft^2}\right)$$

The fixed weight quantity is the sum of the individual system component weights

$$FWT = \sum_i \omega_i \text{ components} \qquad (lb/KW_t)$$

The four parameters (COP, FWT, EPUT, and EFF) which serve to describe the individual systems are input to the program as functions of temperature difference (TRAD-TEVAP). These functions are approximated by straight line segments which are applicable over a limited temperature range. Two points on each line and the maximum temperature difference to which the line pertains are input to define the equation of the individual segments. For the fixed weight term the form of the equation is:

$$FWT = \left(\frac{Y1-Y2}{T1-T2}\right)(TDIF-T1) + Y1 \qquad (lb/KW)$$

where

Y1 & Y2 are ordinate values of the two points on the line
T1 & T2 are abscissa values of the two points on the line
TDIF is the temperature difference (TRAD-TEVAP)

Written in the form used in the computer routine, the basic weight equation is:

$$W(I) = \frac{FEPP}{EPUF(I)} + FWT(I) + RPT(I)$$

where:

$$RPT(I) = \left(1 + \frac{1}{COP(I)}\right) \times \left(\frac{FRAP}{\sigma \in [TRAD^4 - TSINK^4]}\right)$$

and:

W(I)	=	The equivalent weight of system "I", pounds per kilowatt of cooling.
FEPP	=	The factor for electrical power penalty (LB/KW) (electrical))
EPUF(I)	=	The electrical power utilization factor [KW(cooling)/ KW (electrical)], a function of TRAD - TEVAP for system "I", and including component inefficiencies
FWT(I)	=	The fixed weight total in LB/KW (cooling) including any thermal energy penalty, a function of TRAD - TEVAP for system "I"
RPT(I)	=	The radiator penalty total, in LB/KW (cooling), a function of COP, TRAD, FRAP, and TSINK given by the second equation above for system "I"
COP(I)	=	The coefficient of performance for system "I", KW (cooling) per KW of driving source (thermal or mechanical). A function of (TRAD - TEVAP). For a mechanical work system, $\mathrm{COP(I)} = \mathrm{EFF(I)}\,\dfrac{(\mathrm{TEVAP})}{(\mathrm{TRAD} - \mathrm{TEVAP})}$
EFF(I)	=	The net efficiency of the mechanical equipment in the vapor compression refrigerator
FRAP	=	The factor for radiator area penalty, (LB/ft^2)
ϵ	=	The emittance of the radiator, 0.9 was used in this analysis
σ	=	The Stephan-Boltzmann constant
TRAD	=	The effective temperature for radiation heat rejection, ^{o}R - this was taken as arithemetic average fluid temperature minus $10^{o}F$
TSINK	=	The effective temperature for the radiation environment. A function of the orientation and spectral properties of the radiator
TEVAP	=	The effective temperature of the evaporator or mean temperature at which the thermal load is transferred, ^{o}R

In computing the system weight the routine begins at a radiator temperature equal to the evaporator or sink temperature, whichever is larger, and calculates weight totals at fixed increments of radiator temperature until a maximum temperature value (TMAX) is exceeded. Each point for a system curve is then plotted. Should the value of (TRAD-TEVAP) exceed the maximum temperature difference of the last line segment for any of the fixed weight parameters, the routine provides that the iteration stop and the weights calculated for the system be plotted.

For each plot generated, the weight totals for every system are computed with one or more values of radiator area weight penalty. A separate curve is plotted for each area penalty value.

For the five systems considered to be most applicable to space environment use, Table C-1 presents the system parametric data as input to the routine. In the table, the first row of each parameter column corresponds to the equation of the line valid from TDIF = 0°F to TDIF = TMAX. In the cases where more than one equation is needed to approximate the parameter versus temperature function, the succeeding rows of data are applicable from temperatures greater than the maximum temperature of the first row to the next TMAX value.

The vapor compression system, unlike the other systems considered, has a COP value that is highly dependent on evaporator and radiating temperatures. To provide for this characteristic, the EFF data is input as a function of temperature difference like the other parameters, but is only used if the TMAX value is a quantity greater than zero. A test is provided in the routine for this case. When the EFF value of any system is to be used, a set of EPUF and COP values is computed as follows:

$$\mathrm{COP} = \mathrm{EFF}\ (\mathrm{CCOP})$$

and

$$\mathrm{EPUF} = \mathrm{COP}$$

where

CCOP is the COP value of a Carnot type refrigeration system operating between the evaporator temperature and the effective radiator temperature. It is found by:

$$\mathrm{CCOP} = \left(\frac{\mathrm{TEVAP}+460}{\mathrm{TRAD}-\mathrm{TEVAP}}\right)$$

If these values of COP and EPUF are computed, the data input for the system for these parameters is not used by the routine. Because this routine computes and plots total system weights as a function of radiator temperature, lowest system weight for any application corresponds to an optimum radiating temperature. For individual systems, this minimum point can occur at different temperature values depending on the radiator penalty, and electrical power penalty. The plot comparison then defines not only the lightest weight system, but the optimum design radiator temperature for each type of system.

Data for each refrigerator were obtained from individual cycle analyses and correlated for a large system in steady-state operation as a function of (TRAD - TEVAP), since theoretical considerations (as an example, the COP of a Carnot refrigerator) demonstrate that this ΔT is the primary variable in evaluating the performance of a refrigerator. The routine employs linear interpolation between input data points which are presented in Table C-1 for the five active systems presented above. The data in Table C-1 are supplied to the routine as two points on a linear segment of the curve describing the function and a maximum temperature (TMAX), unless otherwise specified, the range of validity is from 0°F to TMAX or the previous TMAX to the next value. The routine performs an internal check so that heat-driven or shaft-driven refrigerators can be compared simultaneously on machine plotted graphs. The

RSPLAT routine was run to generate all combinations of the values for FRAP, TEVAP, TSINK, and FEPP given on Table C-1 resulting in 120 separate plots. Each plot is presented in Appendix D and provides the specific weight for all refrigeration systems simultaneously as a function of TRAD; thus, in performing a trade study, the optimum temperature for each system can be selected for a given operating condition. No attempt was made to evaluate other refrigeration systems or modifications to the basic systems, since detailed knowledge of the particular application would be evaluated with the data in Appendix B by adjusting the penalty factors. For example, in an orbital application with solar cell power, sun side only operation reduces the power penalty factor (FEPP) by about one-half, since storage batteries are not required; or two separate radiator panels which can shadow each other would be equivalent to doubling the radiator area penalty factor (FRAP) but reducing the effective sink temperature (TSINK). Obviously, other variations are possible, but all of these require knowledge of the particular application.

TABLE C-1

ALGORITHMS EMPLOYED IN COMPUTER ANALYSES*

SYSTEM	COP(I)					EPUF(I)					FWT(I)					EFF(I)				
	COP (1) -	COP (2) -	TDIF (1) °F	TDIF (2) °F	TMAX °F	EPUF (1) -	EPUF (2) -	TDIF (1) °F	TDIF (2) °F	TMAX °F	FWT (1) lb/KW	FWT (2) lb/KW	TDIF (1) °F	TDIF (2) °F	TMAX °F	EFF (1) -	EFF (2) -	TDIF (1) °F	TDIF (2) °F	TMAX °F
R-22/E-181 Absorption	.68	.19	10.	150.	200.	375. 160. 40. 28. 5.15	0. 0. 0. 0. 0.	15. 20. 60. 60. 111.	48. 75. 103. 119. 160.	39. 62. 86. 107.5 160.	17. 10. 0.	27. 55. 105.	0. 45. 80.	105. 140. 140.	75. 101. 160.	0.	0.	0. (Not Applicable)	0.	0.
LiBr/H_2O Absortion (Conventional System)	.82 1.05 1.20	.62 .04 0.	0. 20. 57.	90. 110. 83.	51. 73. 82.	80. 140.	54. 0.	7. 45.	115. 83.	65. 82.	33.	41.	10.	130.	82.	0.	0.	0. (Not Applicable)	0.	0.
LiBr/H_2O Absorption Plus Turbine Compressor	.78 .875	.54 .165	0. 10.	150. 260.	98. 310.	46.	30.	20.	255.	310.	35. 42.	47. 82.	50. 180.	220. 290.	190. 310.	0.	0.	0. (Not Applicable)	0.	0.
H_2O Zeolite Adsorption	.58 .37 .63	.16 .37 0.	0. 80. 110.	180. 200. 242.	90. 164. 241.	48. 96.	42. 0.	0. 90.	220. 242.	172. 241.	33.	69.	0.	260.	241.	0.	0.	0. (Not Applicable)	0.	0.
Vapor Compression	0.	0.	0. (Not Applicable)	0.	0.	0.	0.	0. (Not Applicable)	0.	0.	7.6	7.6	10.	20.	310.	.68	.34	0.	280.	310.

* Linear line segments with points (1) and (2) on the line. The range of applicability is from 0°F to T_{MAX} or from the previous T_{MAX} to present value.

3.0 COMPUTER PROGRAM

A flow chart of the computer routine is shown in Figure C-1. The program calculates the weight of the various system components, sums them, and plots them as a function of effective radiating temperature. A complete listing of the routine is given in Table C-2.

Any number of refrigeration systems can be considered. The ones which have been characterized for use in this routine are:

- R22/E181 Vapor Absorption
- LiBR/H_2O Vapor Absorption
- LiBR/H_2O Vapor Absorption plus Turbine/Compressor
- H_2O and Type 13X Zeolite Vapor Adsorption
- Mechanical Vapor Compression Using R12

Preliminary analyses indicated that, for the range of conditions being considered, these systems are the most competitive from a weight standpoint.

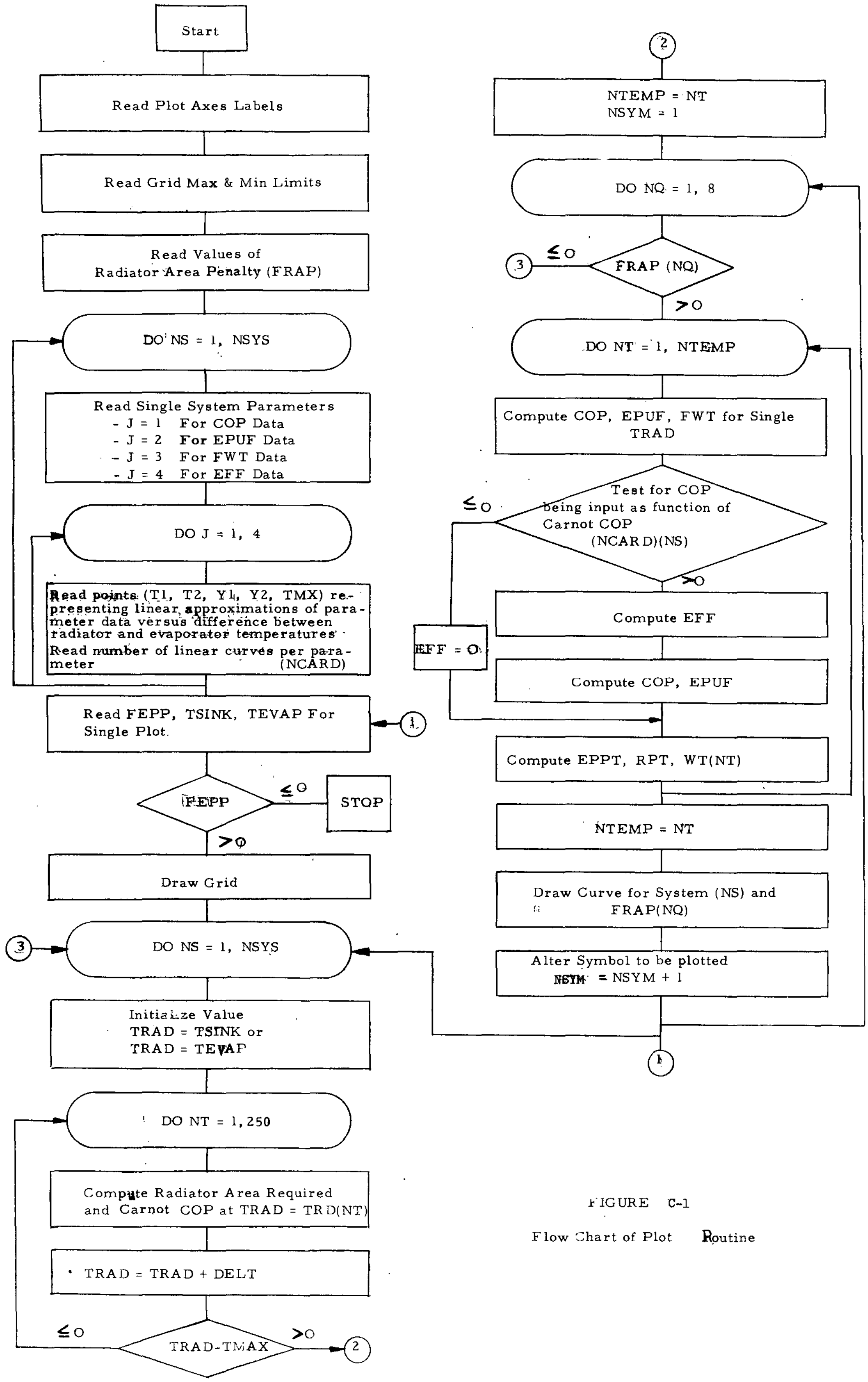

FIGURE C-1

Flow Chart of Plot Routine

TABLE C-2

PLOT PROGRAM

```
      DIMENSION TTEVP(3),TTSK(3),TFPP(3)
      DIMENSION BCDWD(3),FDV(3)
      DIMENSION T1(8,4,8),T2(8,4,8),Y1(8,4,8),Y2(8,4,8),TMX(8,4,8)
      DIMENSION FRAP(8),FWT(250)
      DIMENSION COP(250),WT(250),TRD(250),CCOP(250),QREJ(250),EPUF(250)
      DIMENSION DPX(2),DPY(2),VAR(4),NCARD(8,4)
      DIMENSION TTLSYS(12,8)
      DIMENSION BCDX(12),BCDY(12)
      DIMENSION SYM(15)
      DATA (BCDX(I),I=1,5)/30HRADIATOR TEMPERATURE - DEG F   /
      DATA (BCDY(I),I=1,5)/30HTOTAL SYSTEM WEIGHT - LB/KW    /
      DATA (TFPP(I),I=1,3)/18HFERP =        LB/KW/
      DATA (TTSK(I),I=1,3)/18HTSINK =        F   /
      DATA (TTEVP(I),I=1,3)/18HTEVAP =        F   /
      DATA SYM(1)/1H0/SYM(2)/1H1/SYM(3)/1H2/SYM(4)/1H3/SYM(5)/1H4/
      DATA SYM(6)/1H5/SYM(7)/1H6/SYM(8)/1H7/SYM(9)/1H8/SYM(10)/1H9/
      DATA SYM(11)/1H+/SYM(12)/1HX/SYM(13)/1H*/SYM(14)/1H=/SYM(15)/1H$/
   60 FORMAT(8F10.2)
   61 FORMAT(8I10)
   62 FORMAT(1P8E14.3)
   63 FORMAT(12A6)
   64 FORMAT(1H1,'THERE IS A PROBLEM WITH SOME INPUT PARAMETER')
   65 FORMAT(1H1)
   66 FORMAT(3X,'FRAP =',F7.3,2X,'LB/FT2',16X,'TEVAP =',F8.1,'  F',10X,
     1  'FEPP =',F8.1,2X,'LB/KW',10X,'TSINK =',F8.1,' F ')
   68 FORMAT(1H0,6X,'TRD(NT)',7X,'WT(NT)',7X,'COP(NT)',7X ,'CCOP(NT)',
     1  8X,'EPPT',10X,'RPT',9X,'KWT(NT)',7X,'EPUF(NT)',//)
   69 FORMAT(I5,F15.2,4F10.2,18X,I2)
   70 FORMAT(3X,'SYMBOL PLOTTED',3X,5A1)
      WRITE(6,65)
      READ( 5,60) YMIN,YMAX
      WRITE(6,62) YMIN,YMAX
      READ( 5,60) TMIN,TMAX,DELT
      WRITE(6,62) TMIN,TMAX,DELT
      READ( 5,61) NSYS
      NPLT = 0
C READ SYSTEM PARAMETERS
C NS IS SUBSCRIPT IDENTIFYING SYSTEMS
C J = 1 FOR COP DATA
C J = 2 FOR EPUF DATA
C J = 3 FOR FWT DATA
C J = 4 FOR EFF DATA
      DO 102 NS=1,NSYS
      READ(5,63) (TTLSYS(I,NS),I=1,12)
      DO 102 J=1,4
  101 READ(5,69) NCD,A,B,C,D,E,NLAST
      T1(NS,J,NCD) = A
      Y1(NS,J,NCD) = B
      T2(NS,J,NCD) = C
      Y2(NS,J,NCD) = D
      TMX(NS,J,NCD) = E
      FCD = NCD
      WRITE(6,62) FCD,A,B,C,D,E
      NCARD(NS,J) = NCD
      IF(NLAST) 102,101,102
```

```
      102 CONTINUE
C READ GRID PARAMETERS
      161 CONTINUE
          READ(5,60) FEPP,TSINK,TEVAP,(FRAP(NQ),NQ=1,5)
          IF(FEPP) 6,6,112
      112 CONTINUE
C DRAW GRID
          NSYM = 1
          DPX(1) = TSINK
          DPX(2) = TMAX
          DPY(1) = YMIN
          DPY(2) = YMAX
C DRAW AND LABEL GRID AXES
          CALL FILMAV(1)
          CALL GRID(100,1020,95,972,TSINK,TMAX,YMIN,YMAX)
C DETERMINE CHARACTER SIZE
          EXTERNAL TABLIV
          CALL CHSIZV(2,2)
          CALL RITSTV(12,18,TABLIV)
C WRITE AXES LABELS
          CALL RITE2V(385,15,1023,90,1,30,1,BCDX,NXL)
          CALL RITE2V(30,350,1023,180,1,30,1,BCDY,NYL)
C WRITE GRID IDENTIFYING INFO
          CALL RITE2V(200,1000,1023,90,1,18,1,TTEVP,NTL)
          CALL RITE2V(200,980,1023,90,1,18,1,TTSK,NTL)
          CALL RITE2V(200,960,1023,90,1,18,1,TFPP,NTL)
          FDV(2) = TSINK
          FDV(1) = TEVAP
          FDV(3) = FEPP
          DO 190 I=1,3
          CALL BNBCDV(FDV(I),BCDWD(I),NDS)
          IF(NDS) 189,189,160
      189 NDS = 1
          BCDWD(I) = 1H0
      160 NST = 280 + (6-NDS) * 12
          NVERT = 1020 -I* 20
          NBCD = I*6-5
          CALL RITE2V(NST,NVERT,1023,90,1,NDS,NBCD,BCDWD,NTL)
      190 CONTINUE
          WRITE(6,62) TSINK,TMAX,YMIN,YMAX,DPX(1),DPY(1)
          NPLT = NPLT + 1
C CALCULATE CURVES FOR SINGLE SYSTEM
          DO 86 NS=1,NSYS
C CALCULATE RADIATOR AREA AND CARNOT COP
          IF(TEVAP-TSINK) 123,123,124
      124 TRAD = TEVAP + .004
          GO TO 125
      123 TRAD = TSINK + .004
      125 CONTINUE
          DO 82 NT=1,250
          TRD(NT) = TRAD
          QREJ(NT) =.0000452*(((TRD(NT)+460.)/100.)**4-((TSINK+460.)/100.)
         1  **4)
          IF(TRD(NT)-TEVAP) 120,121,120
      121 CCOP(NT) = 0.
          GO TO 122
      120 CCOP(NT) = (TEVAP+460.)/(TRD(NT)-TEVAP)
      122 CONTINUE
```

```
      TRAD = TRAD+DELT
      IF(TRAD-TMAX) 82,82,83
   82 CONTINUE
   83 CONTINUE
      NTEMP = NT
C CALCULATE CURVES FOR SINGLE VALUE OF FRAP
      DO 90 NQ=1,5
      FQ = NQ
      FS = NS
      IF(FRAP(NQ)) 5,86,85
   85 CONTINUE
      WRITE(6,65)
      WRITE(6,67) NPLT
   67 FORMAT(2X,'PLOT NO. ',I4)
      WRITE(6,63) (TTLSYS(I,NS),I=1,12)
      WRITE(6,66) FRAP(NQ),TEVAP,FEPP,TSINK
      WRITE(6,70) SYM(NSYM),SYM(NSYM),SYM(NSYM),SYM(NSYM),SYM(NSYM)
      WRITE(6,68)
      DO 84 NT=1,NTEMP
      TDIF = TRD(NT)-TEVAP
      DO 103 J=1,3
      MM = NCARD(NS,J)
      DO 104 NCD=1,MM
      IF(TDIF-TMX(NS,J,NCD)) 110,110,104
  104 CONTINUE
  110 VAR(J) = (Y1(NS,J,NCD)-Y2(NS,J,NCD))/(T1(NS,J,NCD)-T2(NS,J,NCD))
     1 *(TDIF-T1(NS,J,NCD))+Y1(NS,J,NCD)
  103 CONTINUE
      IF(TDIF-TMX(NS,3,NCD))140,140,150
  150 NTEMP = NT - 1
      GO TO 290
  140 CONTINUE
      COP(NT) = VAR(1)
      EPUF(NT) = VAR(2)
      FWT(NT) = VAR(3)
C CALCULATE ALL POINTS FOR SINGLE SYSTEM
C TEST FOR COP BEING INPUT AS FUNCTION OF CARNOT COP
      IF(TMX(NS,4,1)) 106,106,107
  106 EFF = 0.
      GO TO 108
  107 MM = NCARD(NS,4)
      DO 209 NCD=1,MM
      IF( TDIF  -TMX(NS,4,NCD)) 210,210,209
  209 CONTINUE
      NTEMP = NT - 1
      GO TO 290
  210 EFF = (Y1(NS,4,NCD)-Y2(NS,4,NCD))/(T1(NS,4,NCD)-T2(NS,4,NCD))
     1 *(TDIF-T1(NS,4,NCD))+Y1(NS,4,NCD)
      COP(NT) = EFF*CCOP(NT)
      EPUF(NT) = COP(NT)
  108 CONTINUE
      EPPT = FEPP/EPUF(NT)
      RPT = (1.+1./COP(NT))*FRAP(NQ)/QREJ(NT)
      WT(NT)  =  EPPT+RPT+FWT(NT)
      WRITE(6,62) TRD(NT),WT(NT),COP(NT),CCOP(NT),EPPT,RPT,FWT(NT),
     1  EPUF(NT)
   84 CONTINUE
      NTEMP = NT
```

TABLE C-2 CONTINUED

```
      290 CONTINUE
          IF(NTEMP)90,90,92
       92 CONTINUE
C DRAW CURVE FOR FRAP(NQ) AND SYSTEM(NS)
          CALL PLOTIV (1,1,TRD(1),WT(1),NTEMP,1,SYM(NSYM))
          NSYM = NSYM+1
          IF(NSYM-15) 90,90,91
       91 NSYM = 1
       90 CONTINUE
       86 CONTINUE
          GO TO 161
        5 WRITE(6,64)
        6 CONTINUE
          STOP
          END
```

END OF UNIVAC 1108 FORTRAN V COMPILATION;   0 *DIAGNOSTIC* MESSAGE(S)

APPENDIX D

REFRIGERATION

SYSTEMS SPECIFIC WEIGHT COMPARISONS

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APPENDIX D

REFRIGERATION SYSTEMS SPECIFIC WEIGHT COMPARISON

This section presents the specific weight data generated in this study. Conventional radiator specific weights are presented in Table D-1. The specific weights for the 5 refrigeration systems described in Section 4.0 are presented on pages D-4 through D-124. The computer routine discussed in Appendix C was used in calculating these results. The results are presented as specific weight as a function of radiator (i.e., condenser) temperature for various values of evaporator temperature (TEVAP), radiation sink temperature (TSINK), Power Penalty (FEPP), and radiator area penalty (FRAP). The waste-heat utilization penalty (FTHER) is zero in all cases presented.

TABLE D-1
CONVENTIONAL RADIATOR SYSTEM TOTAL WEIGHT

WT = FWT + (FRAP/QREJ)
WT = 8.5 lb/KW +(FRAP/QREJ)

TSINK = 0°F

TEVAP* °F	SPECIFIC SYSTEM WEIGHT WT (lb/KW) FRAP = .1	FRAP = .6	FRAP = 1.0
40	20.8 **	82.3 **	131.5 **
45	19.4	73.9	117.5
50	18.2	66.5	105.2
65	15.6	50.9	79.3

TSINK = 20°F

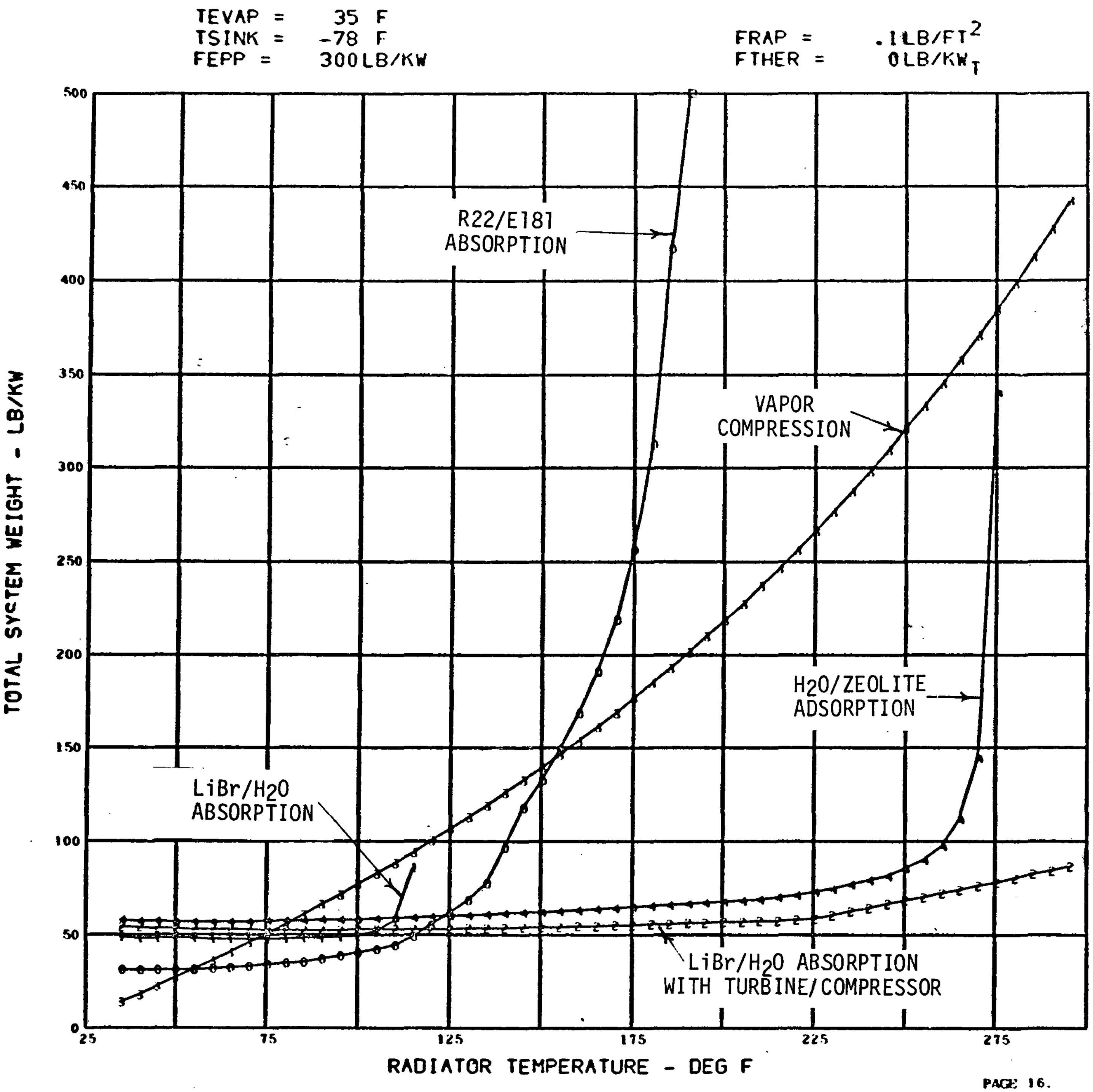
40	32.0	149.5	243.5
45	27.0	119.5	193.5
50	23.7	99.6	160.5
65	18.2	66.6	105.3

TSINK = -15.9°F

40	17.9	64.6	102.1
45	17.0	59.2	93.1
50	15.6	54.8	85.6
65	14.5	44.3	68.2

* TEVAP is the mean of the inlet and outlet temperatures of the heat exchanger and is to be equal to the effective radiator temperature.

**Plotted in Figure 5-6, with $Q_{REJ} = 0.008$ KW/ft^2


TEVAP = 35 F
TSINK = -78 F
FEPP = 300LB/KW
FRAP = .1LB/FT2
FTHER = 0LB/KWT
R22/E181 ABSORPTION
VAPOR COMPRESSION
H2O/ZEOLITE ADSORPTION
LiBr/H2O ABSORPTION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEIGHT - LB/KW
RADIATOR TEMPERATURE - DEG F
500
450
400
350
300
250
200
150
100
50
0
25
75
125
175
225
275
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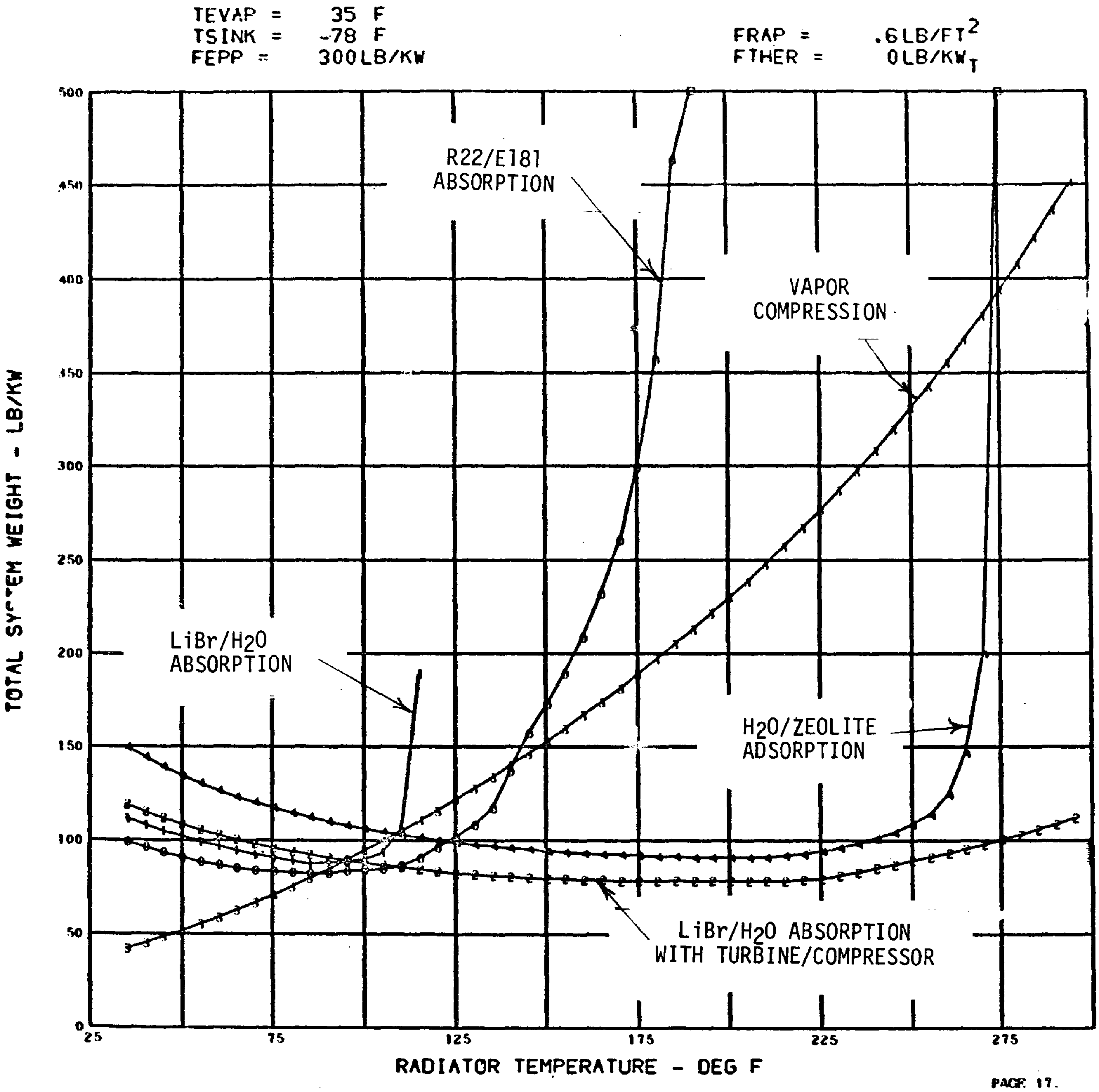

TEVAP = 35 F
TSINK = -78 F
FEPP = 300LB/KW
FRAP = .6LB/FT2
FTHER = 0LB/KWT
R22/E181
ABSORPTION
VAPOR
COMPRESSION
LiBr/H2O
ABSORPTION
H2O/ZEOLITE
ADSORPTION
LiBr/H2O ABSORPTION
WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEIGHT - LB/KW
RADIATOR TEMPERATURE - DEG F
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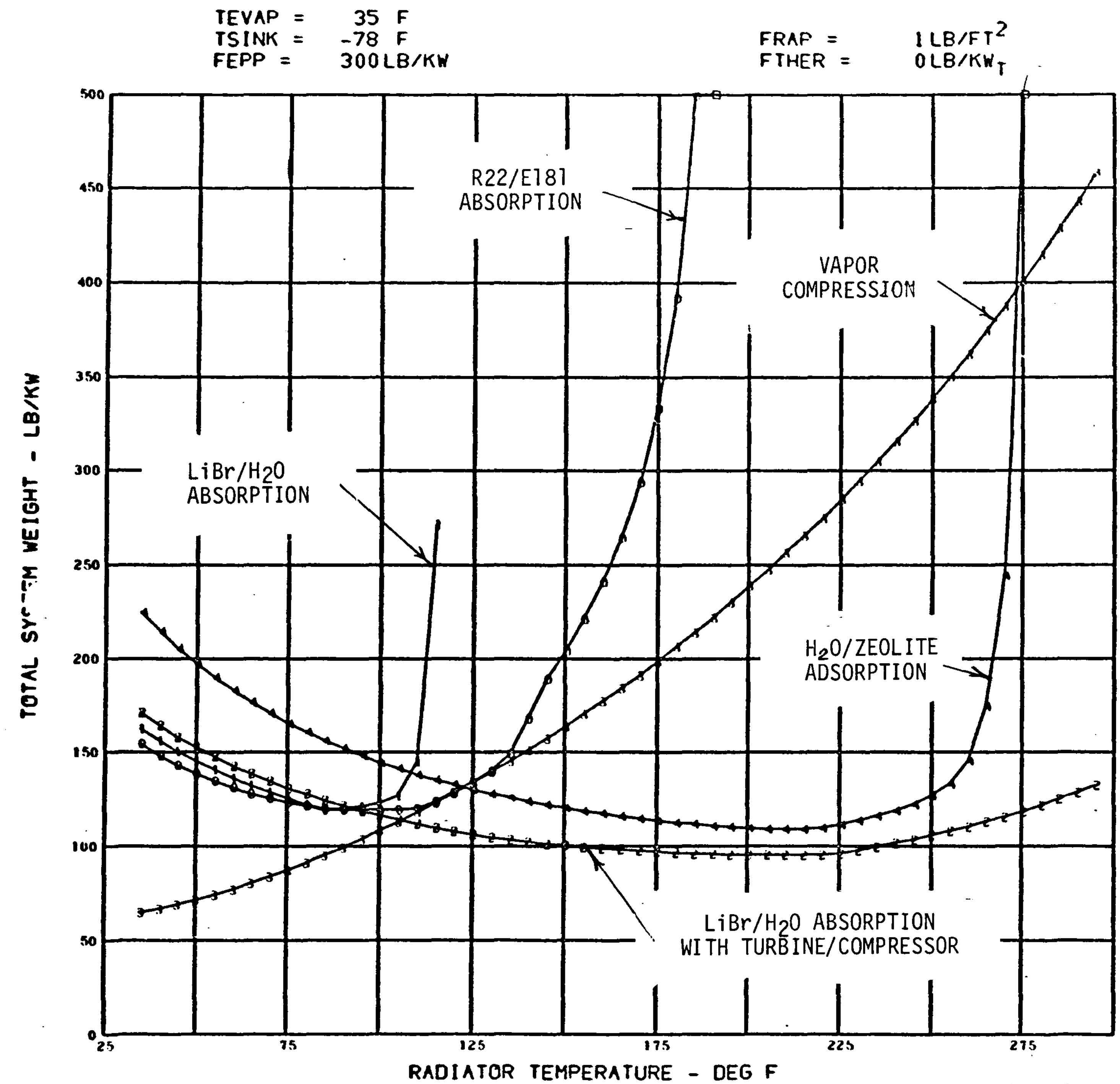

TEVAP = 35 F
TSINK = -78 F
FEPP = 300 LB/KW
FRAP = 1 LB/FT2
FTHER = 0 LB/KWT
R22/E181 ABSORPTION
VAPOR COMPRESSION
LiBr/H2O ABSORPTION
H2O/ZEOLITE ADSORPTION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEIGHT - LB/KW
RADIATOR TEMPERATURE - DEG F
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100
150
200
250
300
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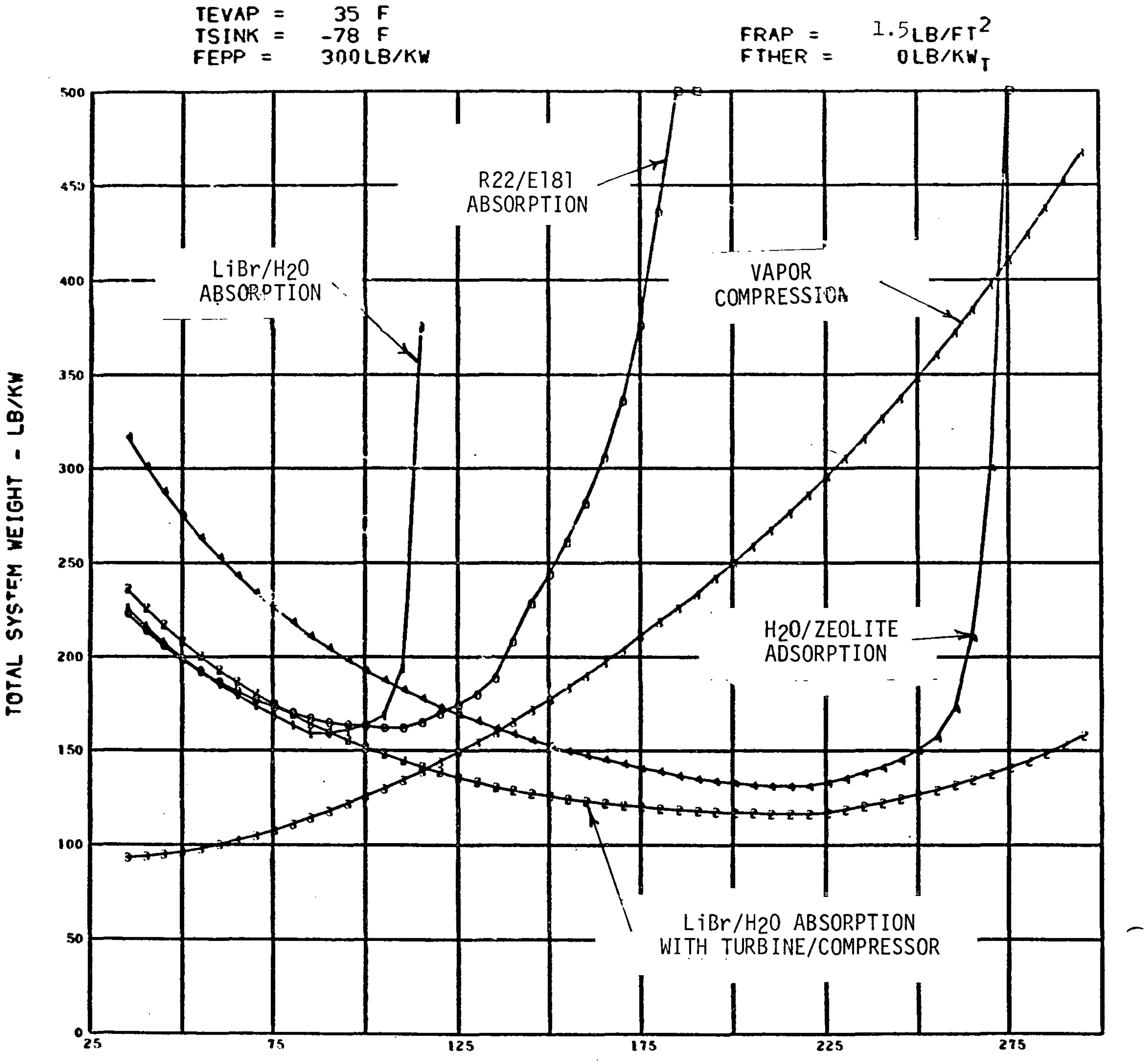
450
500
25
75
125
175
225
275



TEVAP = 35 F
TSINK = -78 F
FEPP = 300 LB/KW
FRAP = 1.5 LB/FT2
FTHER = 0 LB/KWT
R22/E181 ABSORPTION
LiBr/H2O ABSORPTION
VAPOR COMPRESSION
H2O/ZEOLITE ADSORPTION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEIGHT - LB/KW
RADIATOR TEMPERATURE - DEG F
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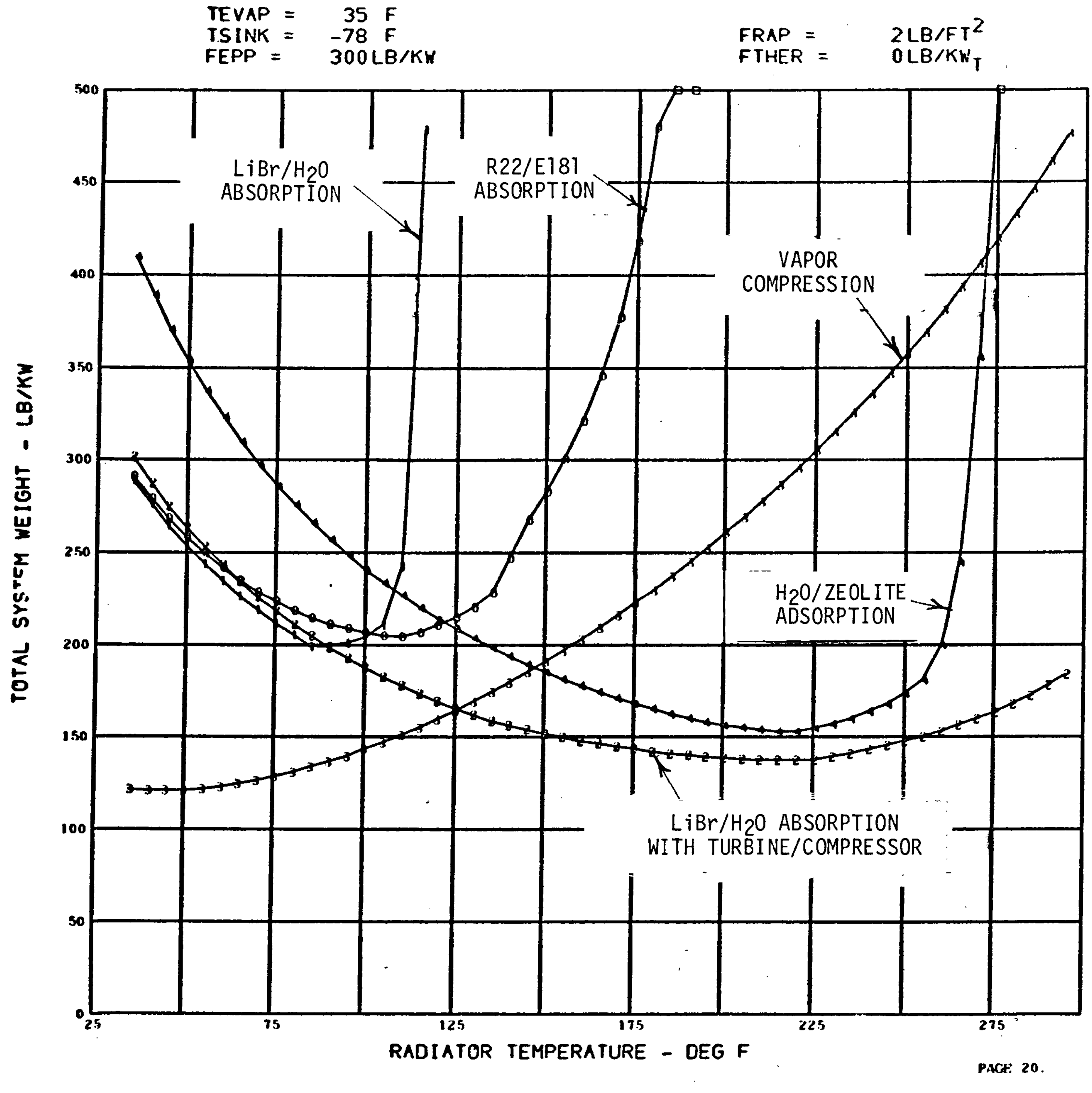
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TSINK = -78 F
FEPP = 300 LB/KW
FRAP = 2 LB/FT2
FTHER = 0 LB/KWT
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R22/E181 ABSORPTION
VAPOR COMPRESSION
H2O/ZEOLITE ADSORPTION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEIGHT - LB/KW
RADIATOR TEMPERATURE - DEG F
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450
500
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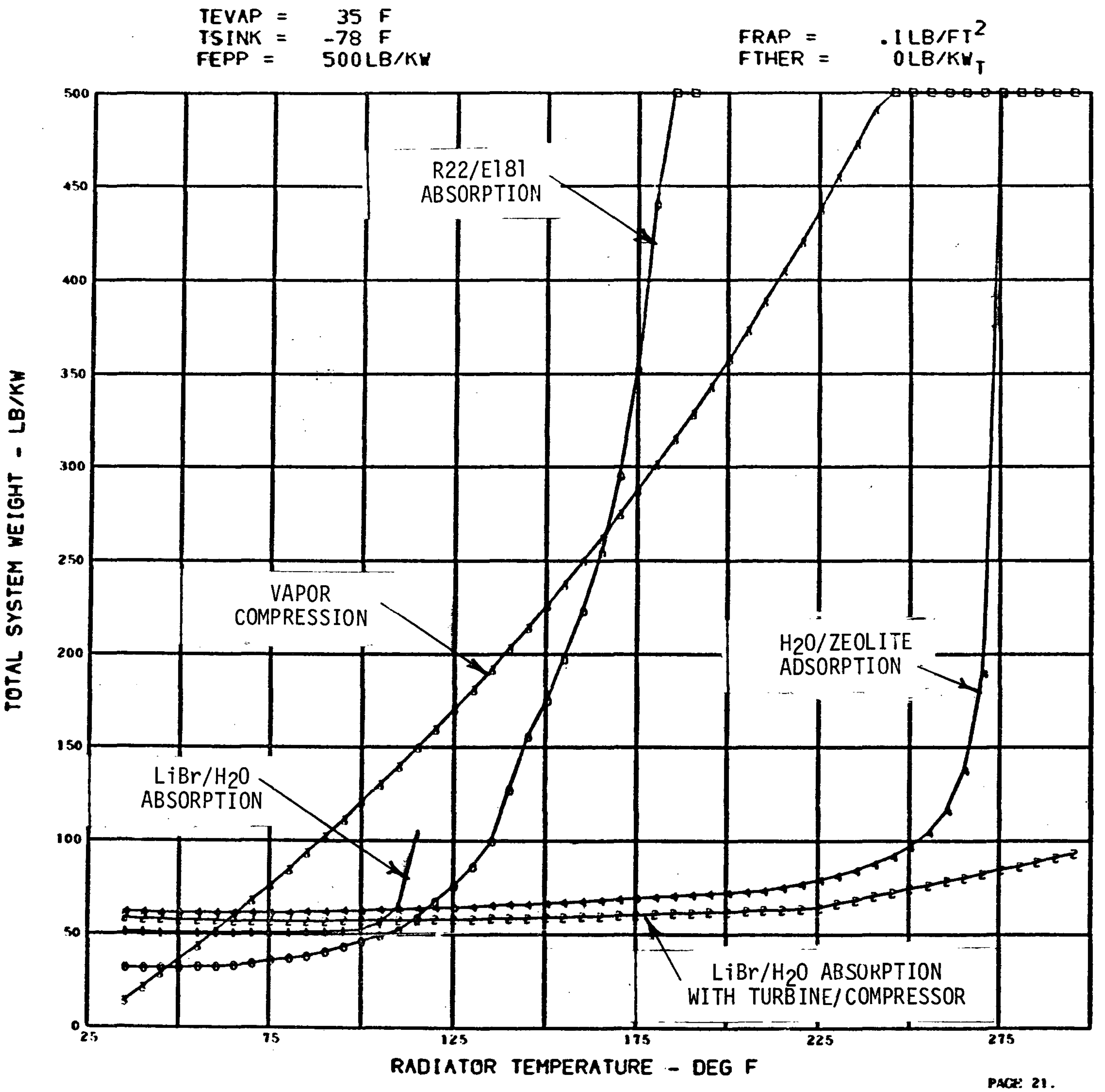
225
275
PAGE 20.



TEVAP = 35 F
TSINK = -78 F
FEPP = 500LB/KW
FRAP = .1LB/FT2
FTHER = 0LB/KWT
R22/E181 ABSORPTION
VAPOR COMPRESSION
H2O/ZEOLITE ADSORPTION
LiBr/H2O ABSORPTION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEIGHT - LB/KW
RADIATOR TEMPERATURE - DEG F
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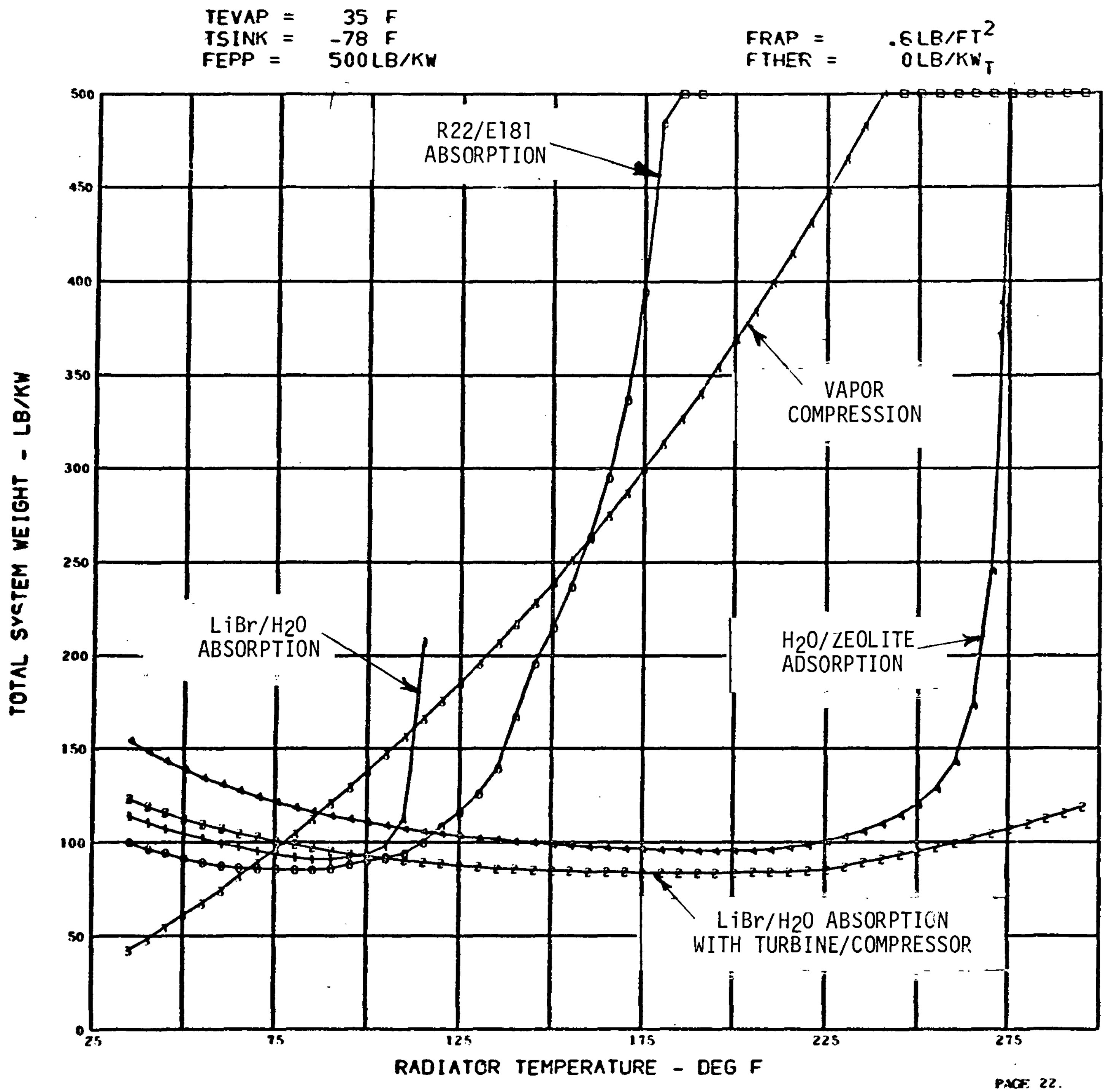
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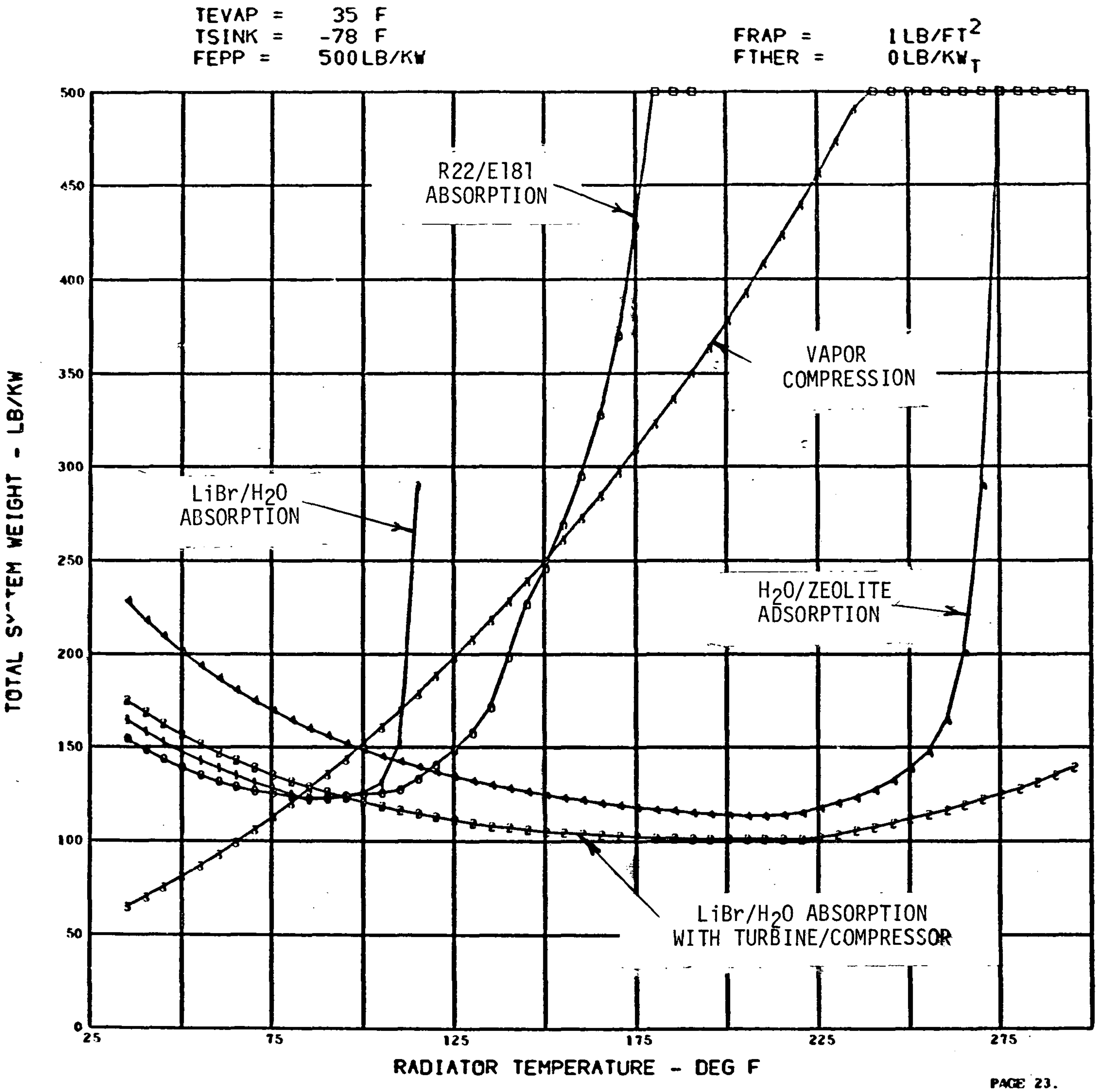
TSINK = -78 F
FEPP = 500 LB/KW
FRAP = .6 LB/FT2
FTHER = 0 LB/KWT
R22/E181 ABSORPTION
VAPOR COMPRESSION
LiBr/H2O ABSORPTION
H2O/ZEOLITE ADSORPTION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEIGHT - LB/KW
RADIATOR TEMPERATURE - DEG F
PAGE 22.



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TSINK = -78 F
FEPP = 500 LB/KW
FRAP = 1 LB/FT²
FTHER = 0 LB/KWT
R22/E181 ABSORPTION
VAPOR COMPRESSION
LiBr/H2O ABSORPTION
H2O/ZEOLITE ADSORPTION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEIGHT - LB/KW
RADIATOR TEMPERATURE - DEG F
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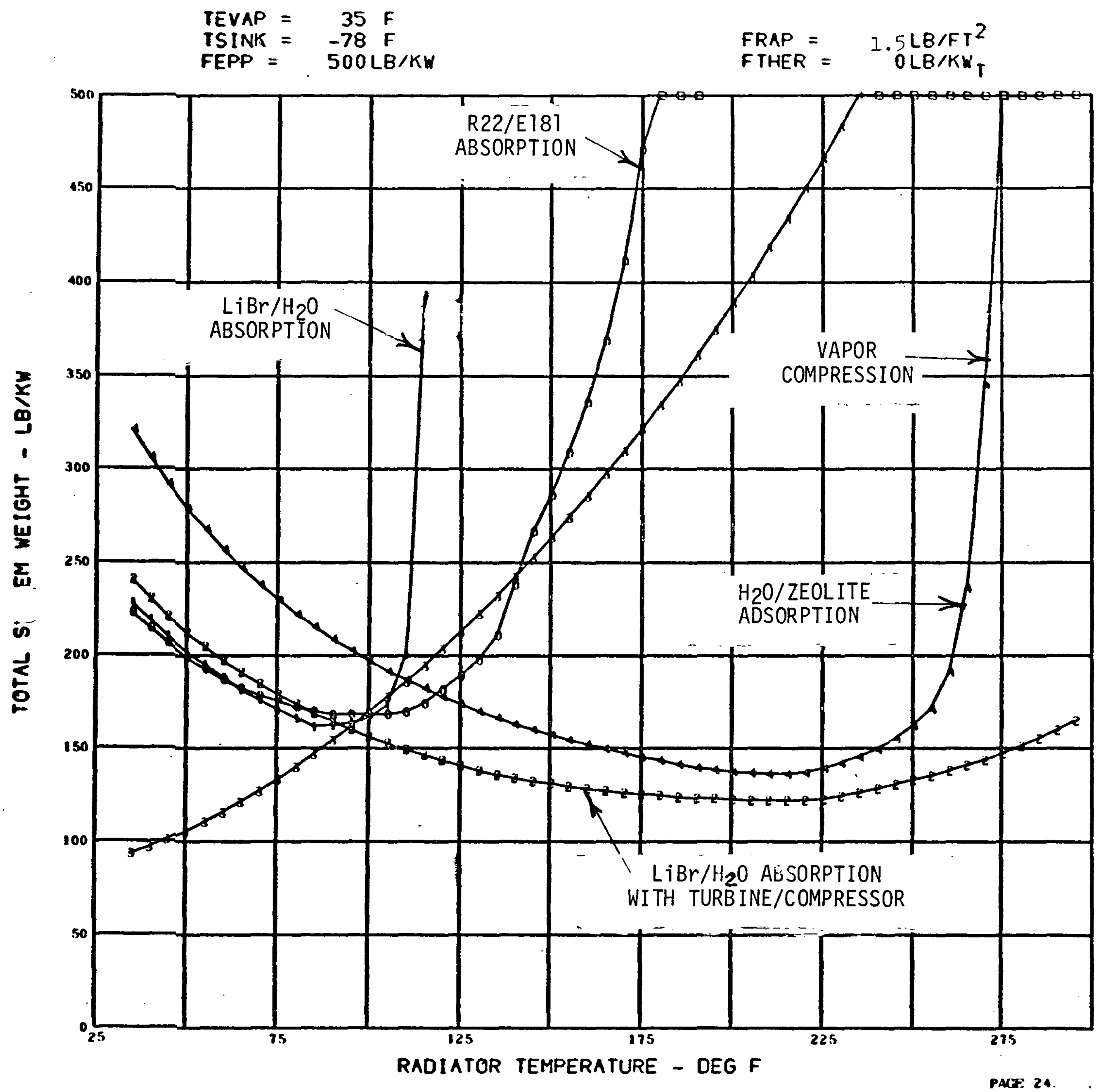
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FRAP = 1.5 LB/FT2
FTHER = 0 LB/KWT
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LiBr/H2O ABSORPTION
VAPOR COMPRESSION
H2O/ZEOLITE ADSORPTION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEIGHT - LB/KW
RADIATOR TEMPERATURE - DEG F
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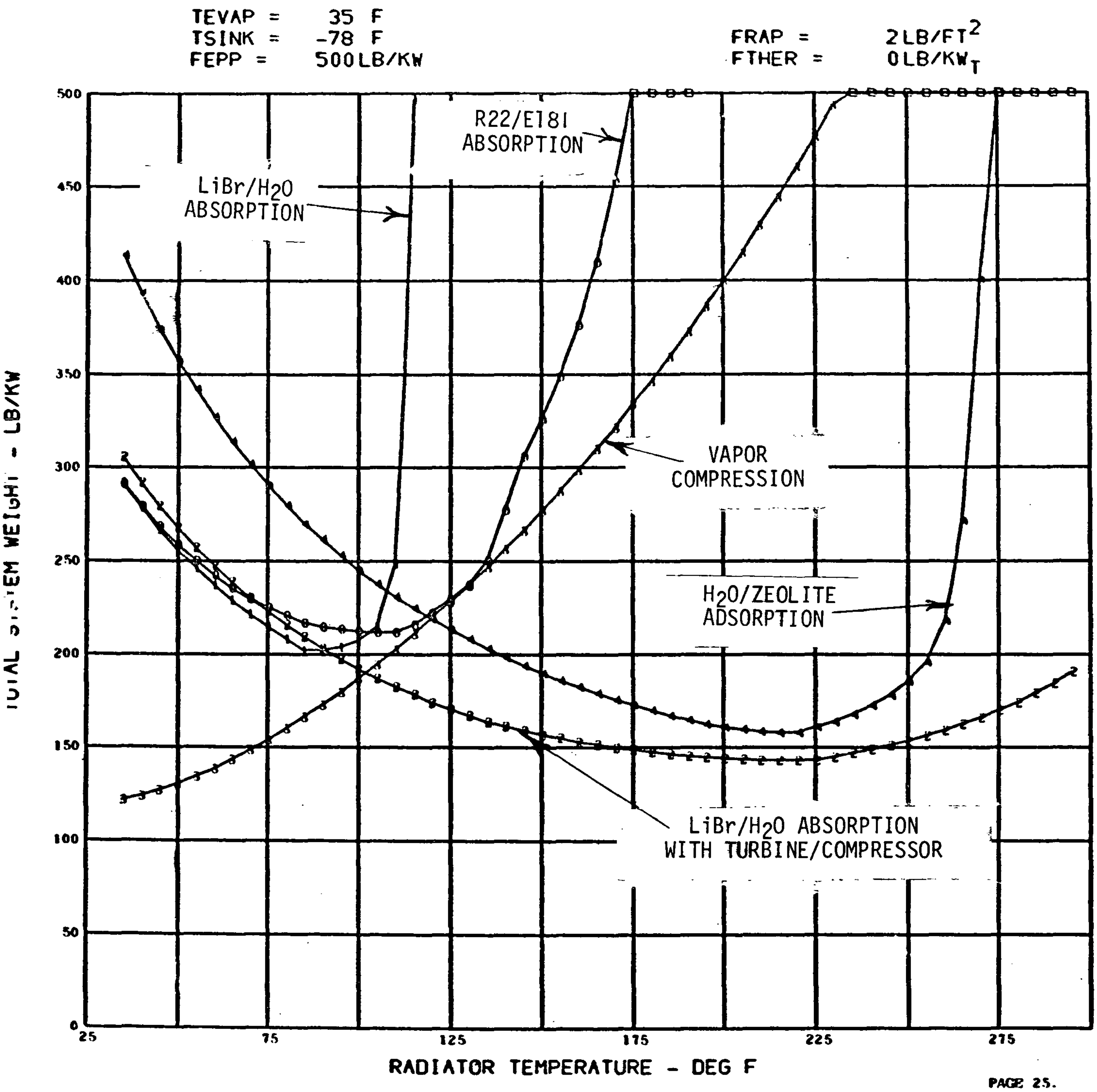
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R22/E181 ABSORPTION
LiBr/H2O ABSORPTION
VAPOR COMPRESSION
H2O/ZEOLITE ADSORPTION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEIGHT - LB/KW
RADIATOR TEMPERATURE - DEG F
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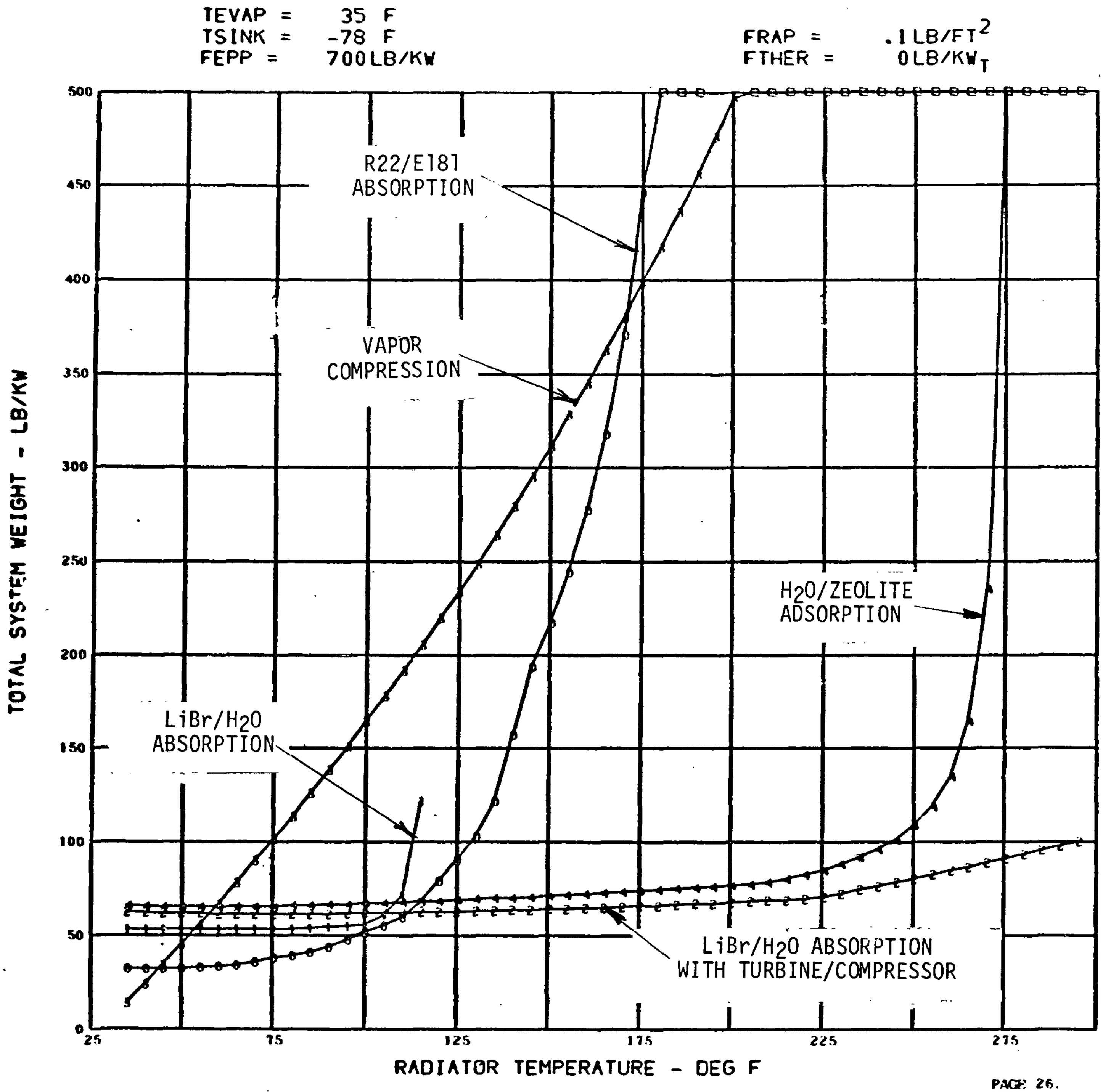
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VAPOR COMPRESSION
H2O/ZEOLITE ABSORPTION
LiBr/H2O ABSORPTION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEIGHT - LB/KW
RADIATOR TEMPERATURE - DEG F
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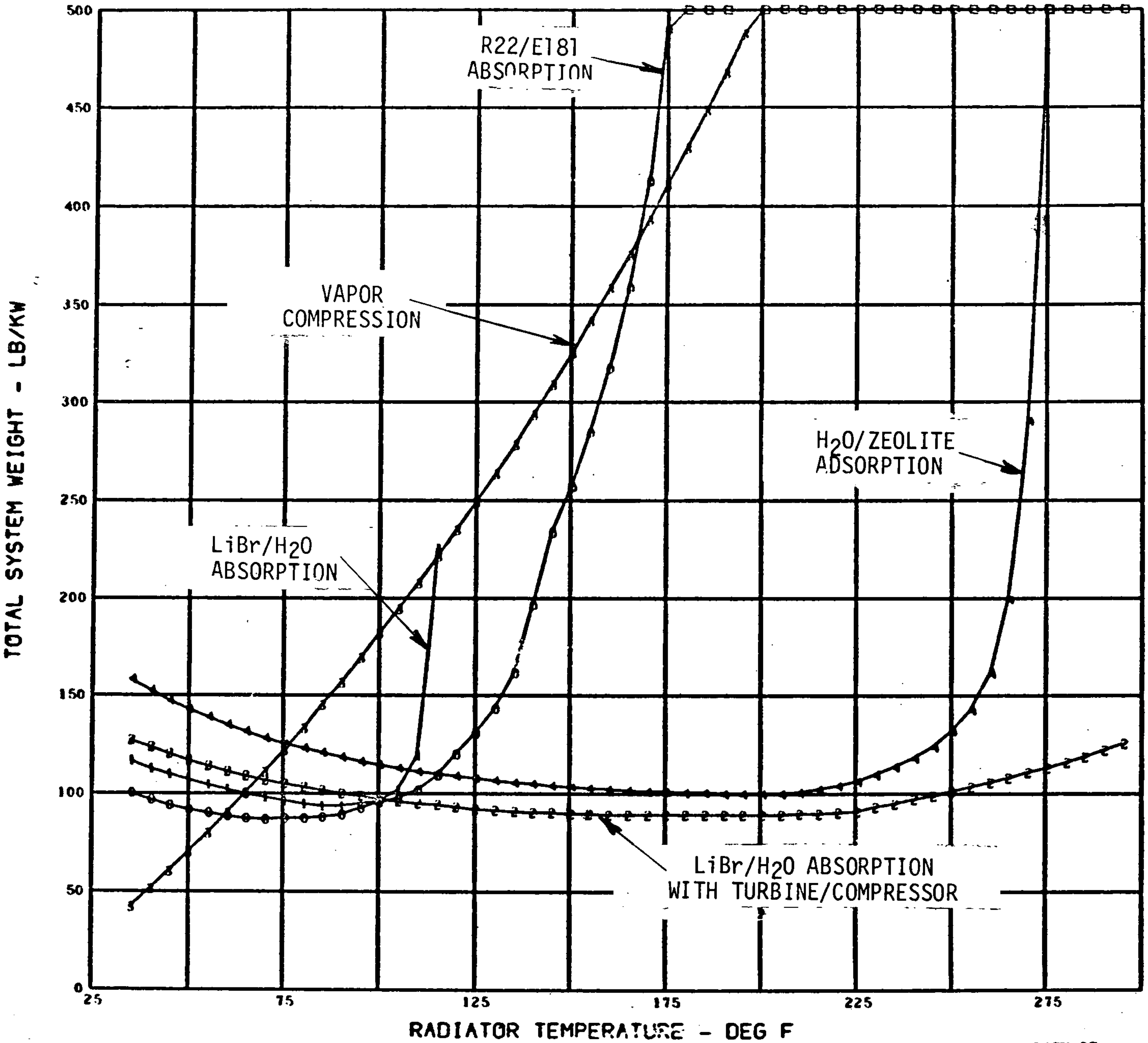
TEVAP = 35 F
TSINK = -78 F
FEPP = 700 LB/KW

FRAP = .6 LB/FT^2
FTHER = 0 LB/KW$_T$



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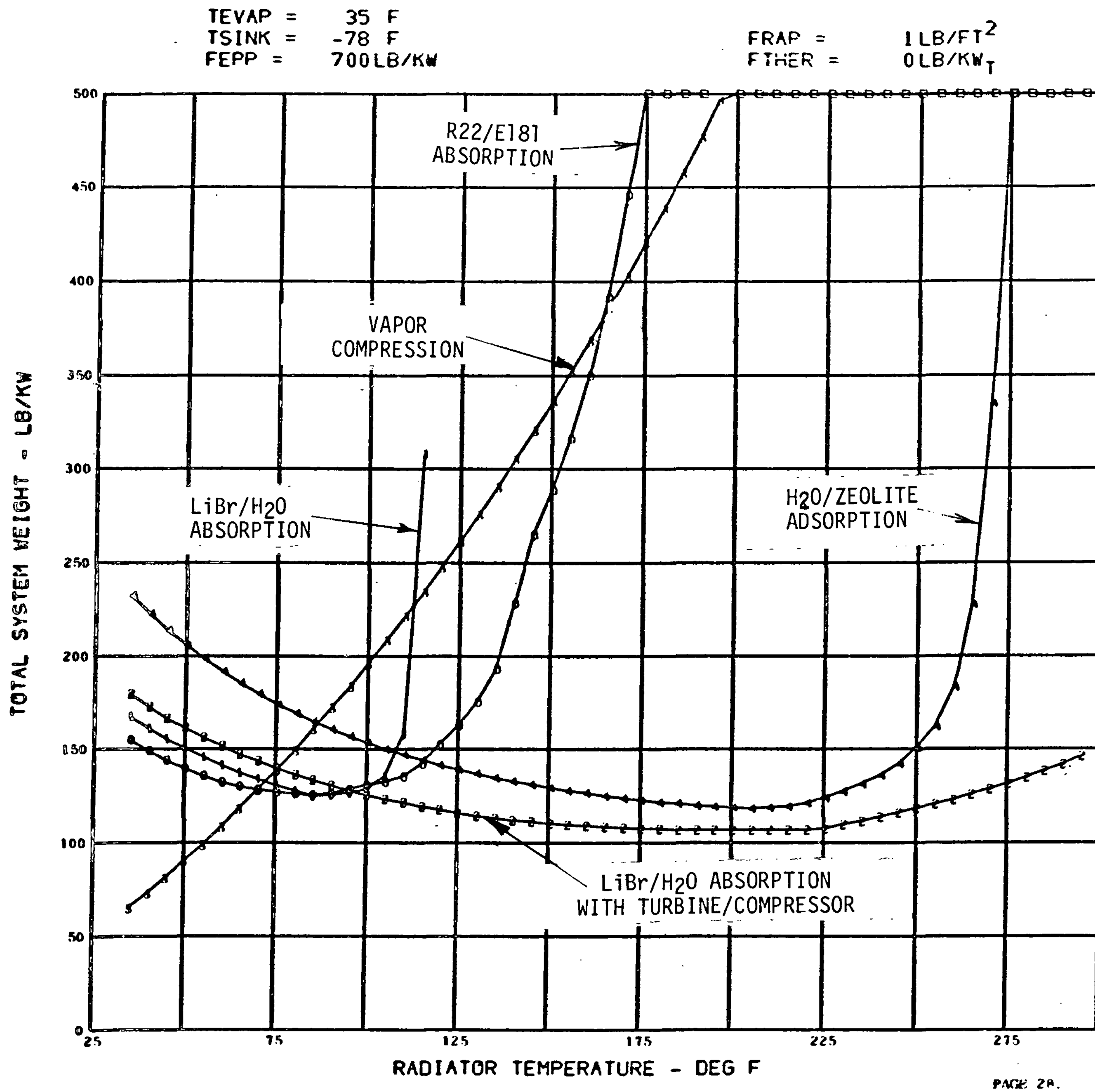

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TSINK = -78 F
FEPP = 700LB/KW
FRAP = 1LB/FT2
FTHER = 0LB/KWT
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LiBr/H2O ABSORPTION
H2O/ZEOLITE ADSORPTION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEIGHT - LB/KW
RADIATOR TEMPERATURE - DEG F
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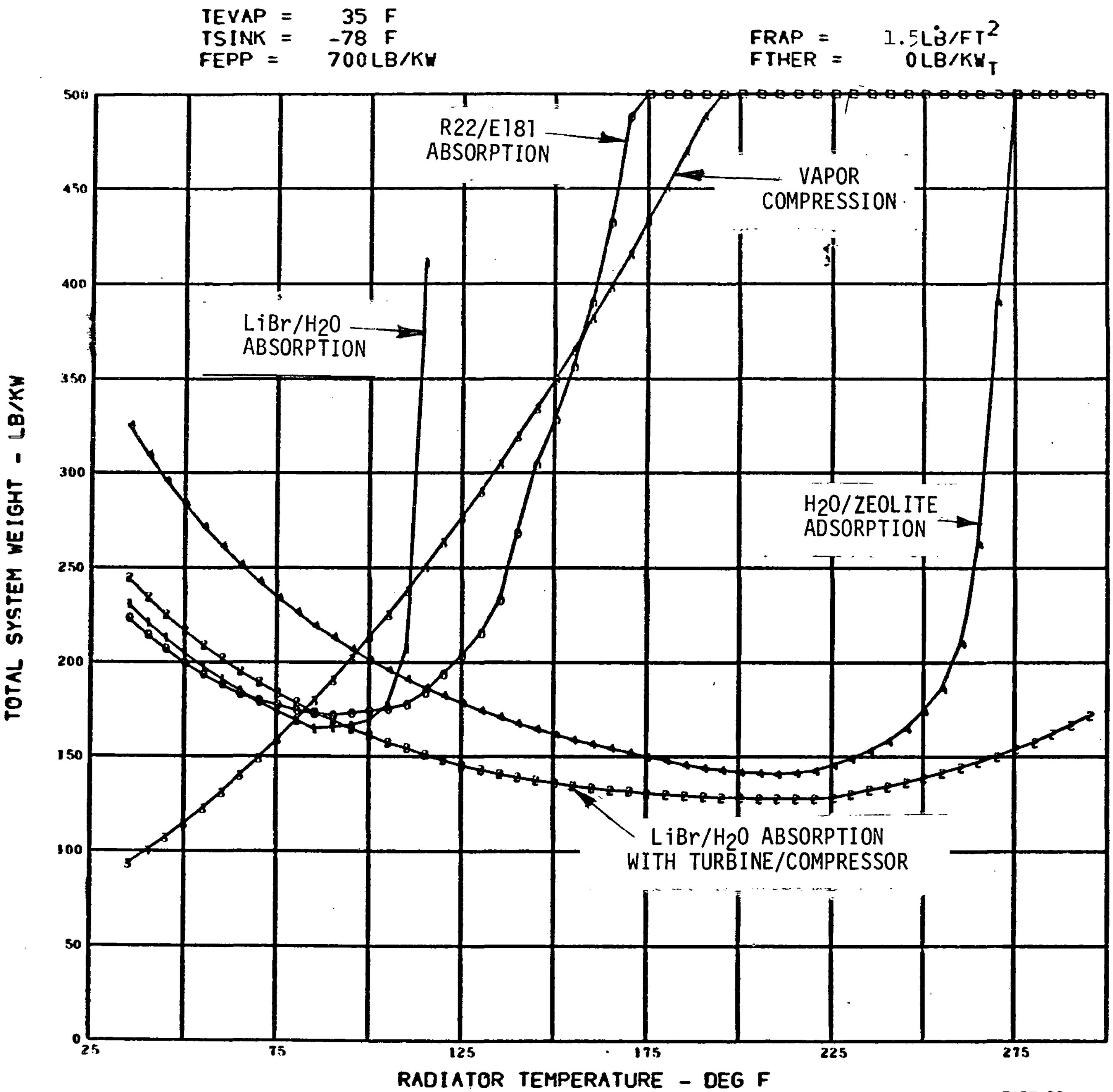
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TSINK = -78 F
FEPP = 700 LB/KW
FRAP = 1.5 LB/FT^2
FTHER = 0 LB/KW$_T$
R22/E181 ABSORPTION
VAPOR COMPRESSION
LiBr/H$_2$O ABSORPTION
H$_2$O/ZEOLITE ADSORPTION
LiBr/H$_2$O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEIGHT - LB/KW
RADIATOR TEMPERATURE - DEG F
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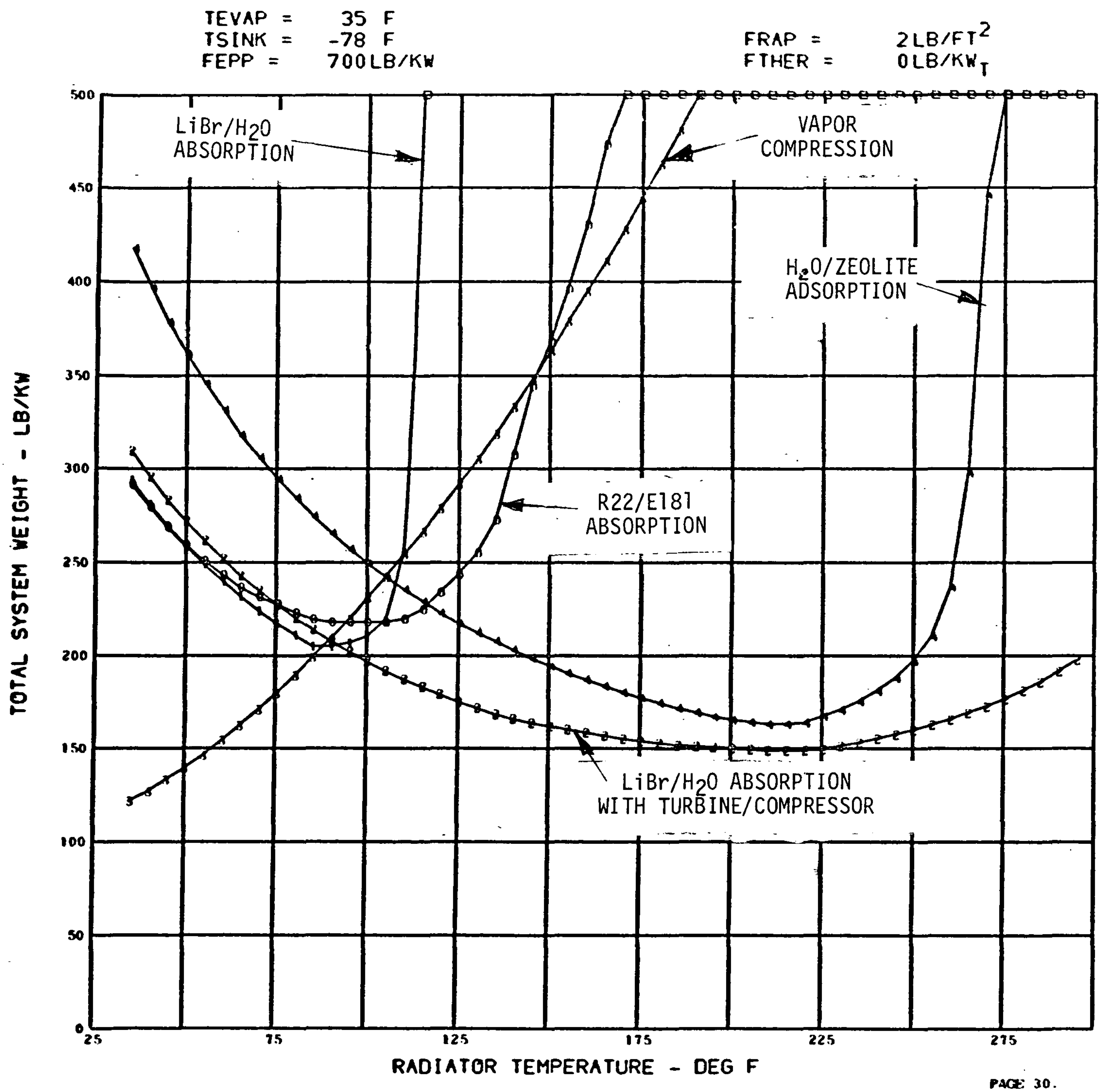
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FRAP = 2LB/FT2
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VAPOR COMPRESSION
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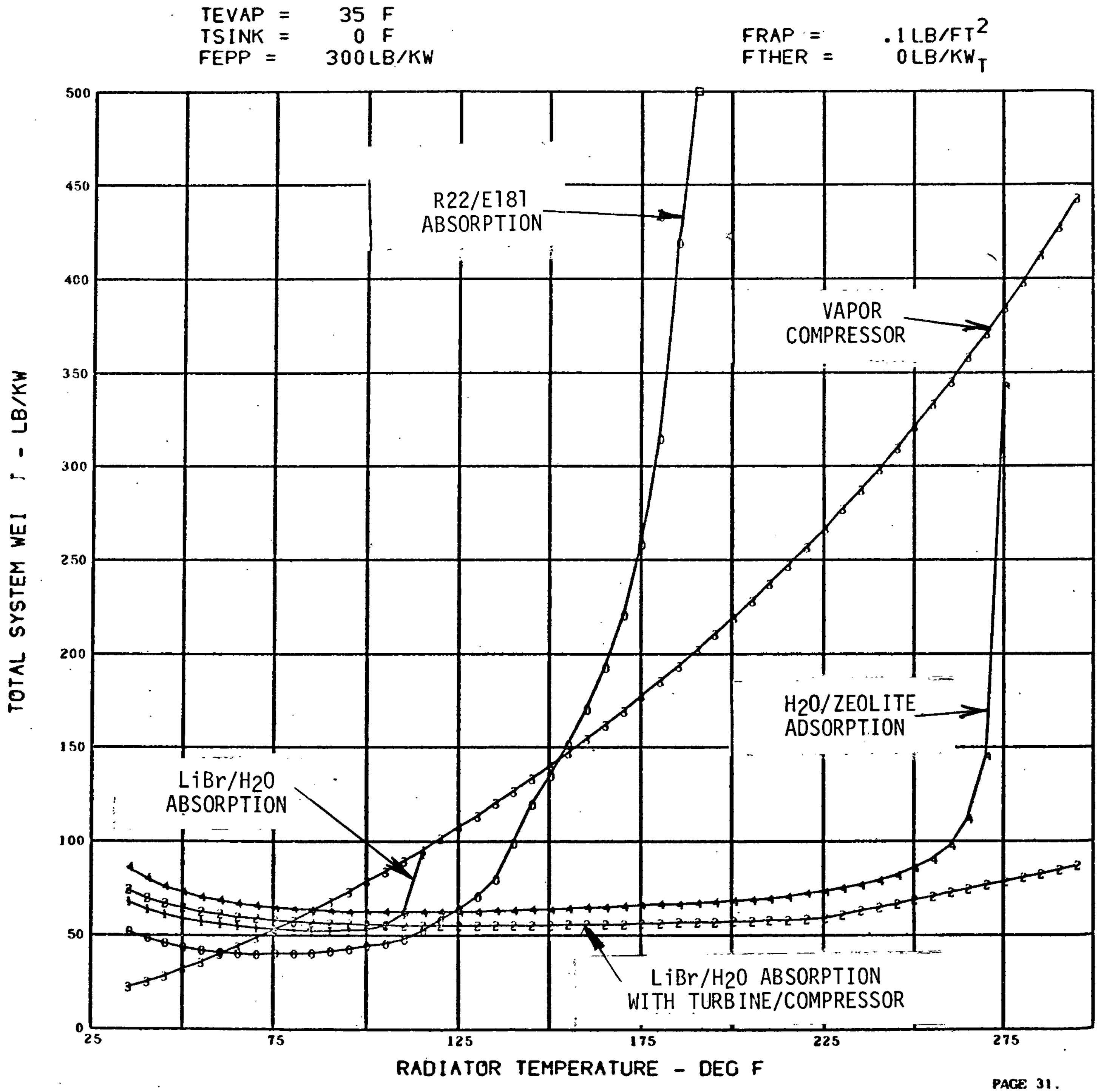
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FRAP = .1 LB/FT2
FTHER = 0 LB/KWT
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VAPOR COMPRESSOR
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LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEI - LB/KW
RADIATOR TEMPERATURE - DEG F
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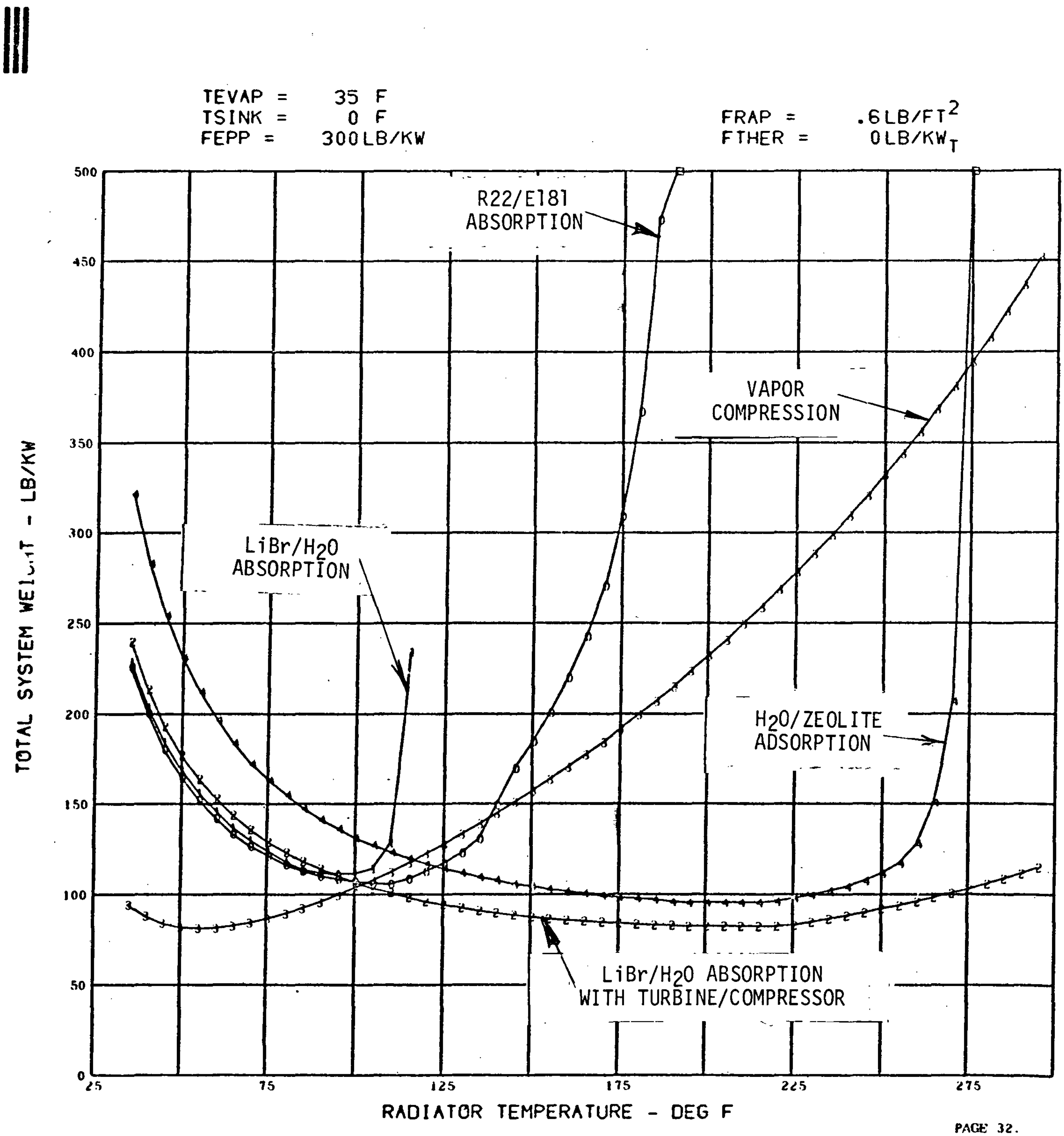
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FRAP = .6LB/FT2
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VAPOR COMPRESSION
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H2O/ZEOLITE ADSORPTION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEIGHT - LB/KW
RADIATOR TEMPERATURE - DEG F
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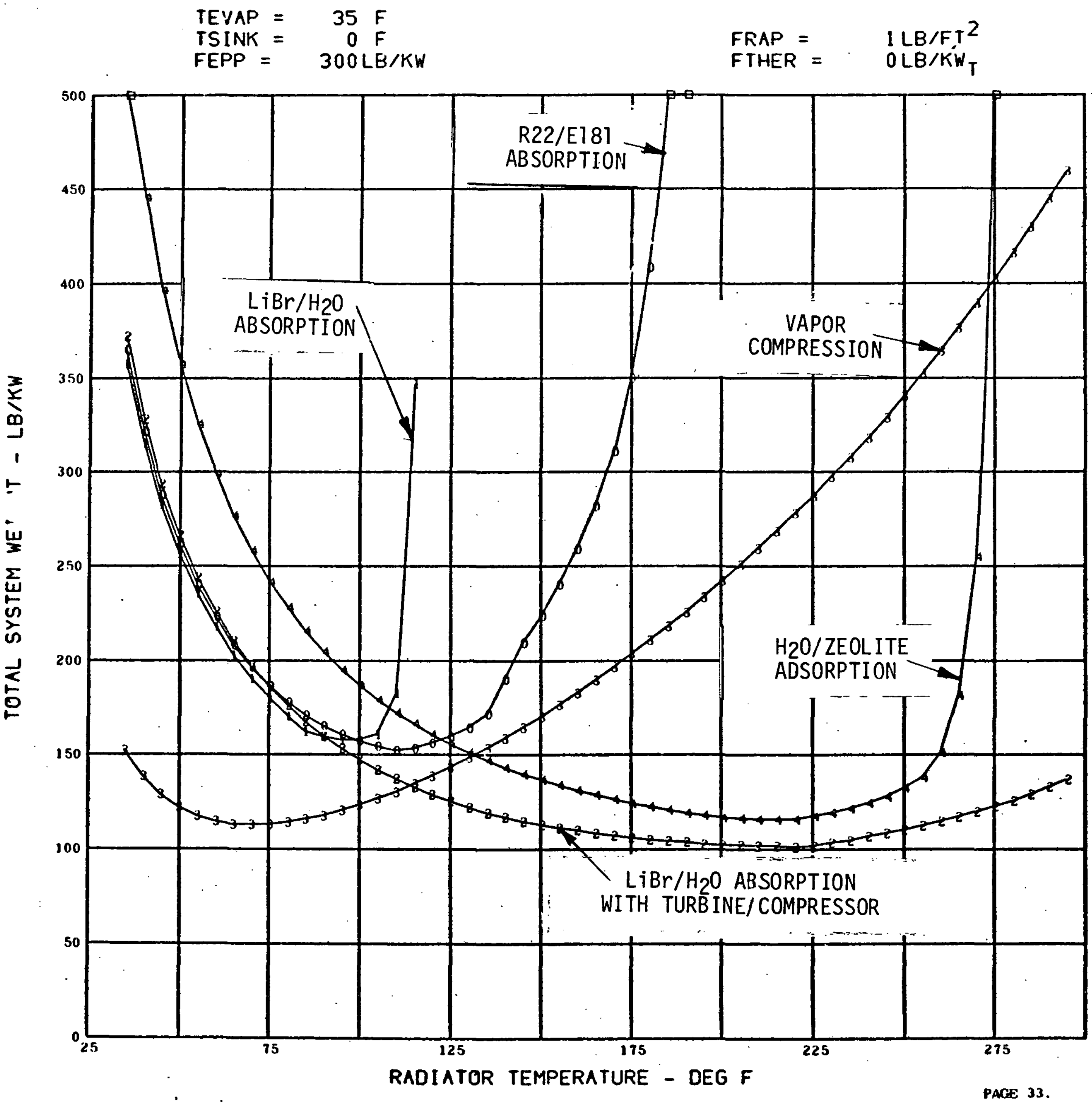

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TSINK = 0 F
FEPP = 300LB/KW
FRAP = 1LB/FT2
FTHER = 0LB/KWT
R22/E181 ABSORPTION
LiBr/H2O ABSORPTION
VAPOR COMPRESSION
H2O/ZEOLITE ADSORPTION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WE' 'T - LB/KW
RADIATOR TEMPERATURE - DEG F
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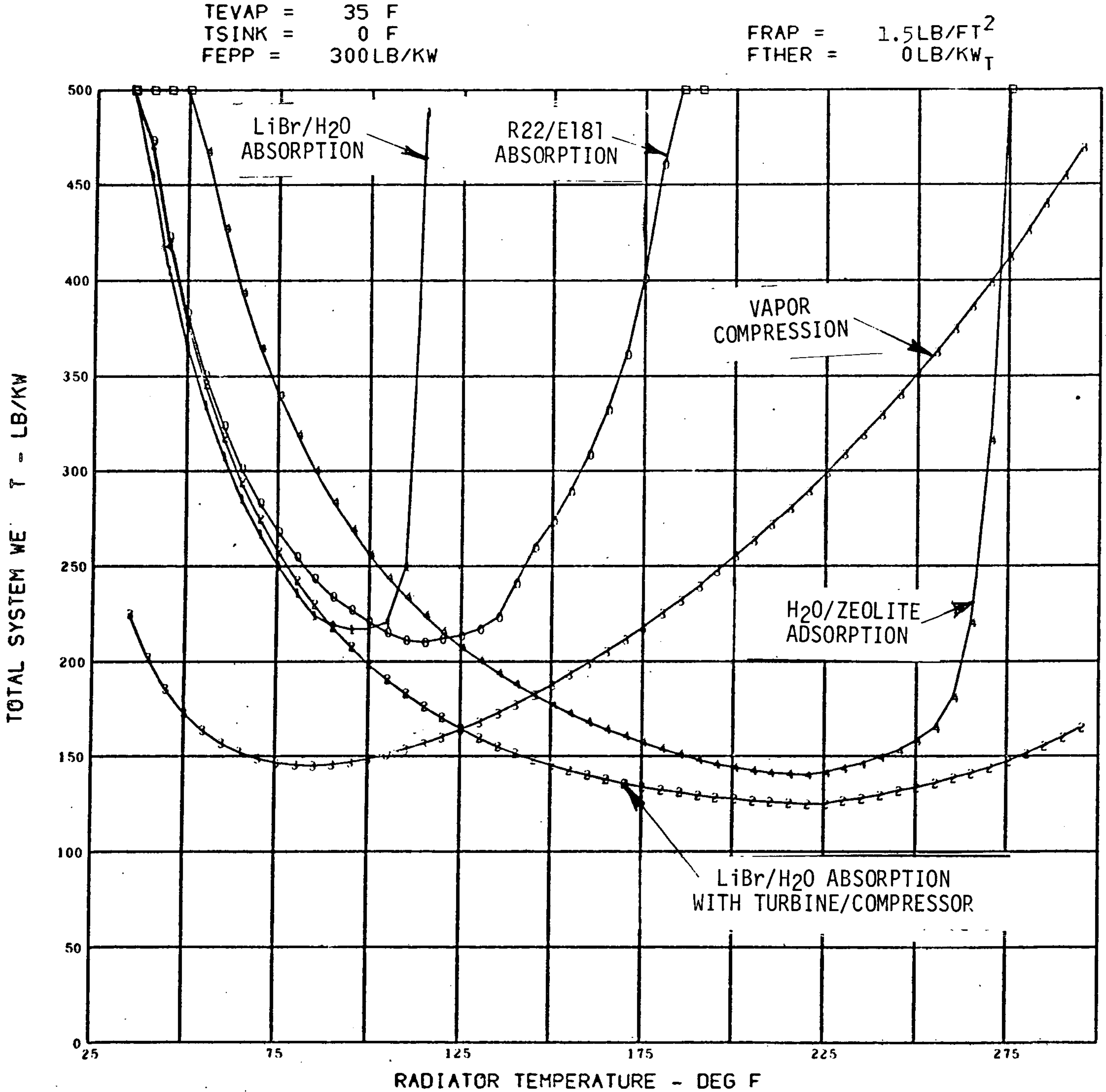

TEVAP = 35 F
TSINK = 0 F
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FRAP = 1.5LB/FT2
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LiBr/H2O ABSORPTION
R22/E181 ABSORPTION
VAPOR COMPRESSION
H2O/ZEOLITE ADSORPTION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEIGHT - LB/KW
RADIATOR TEMPERATURE - DEG F
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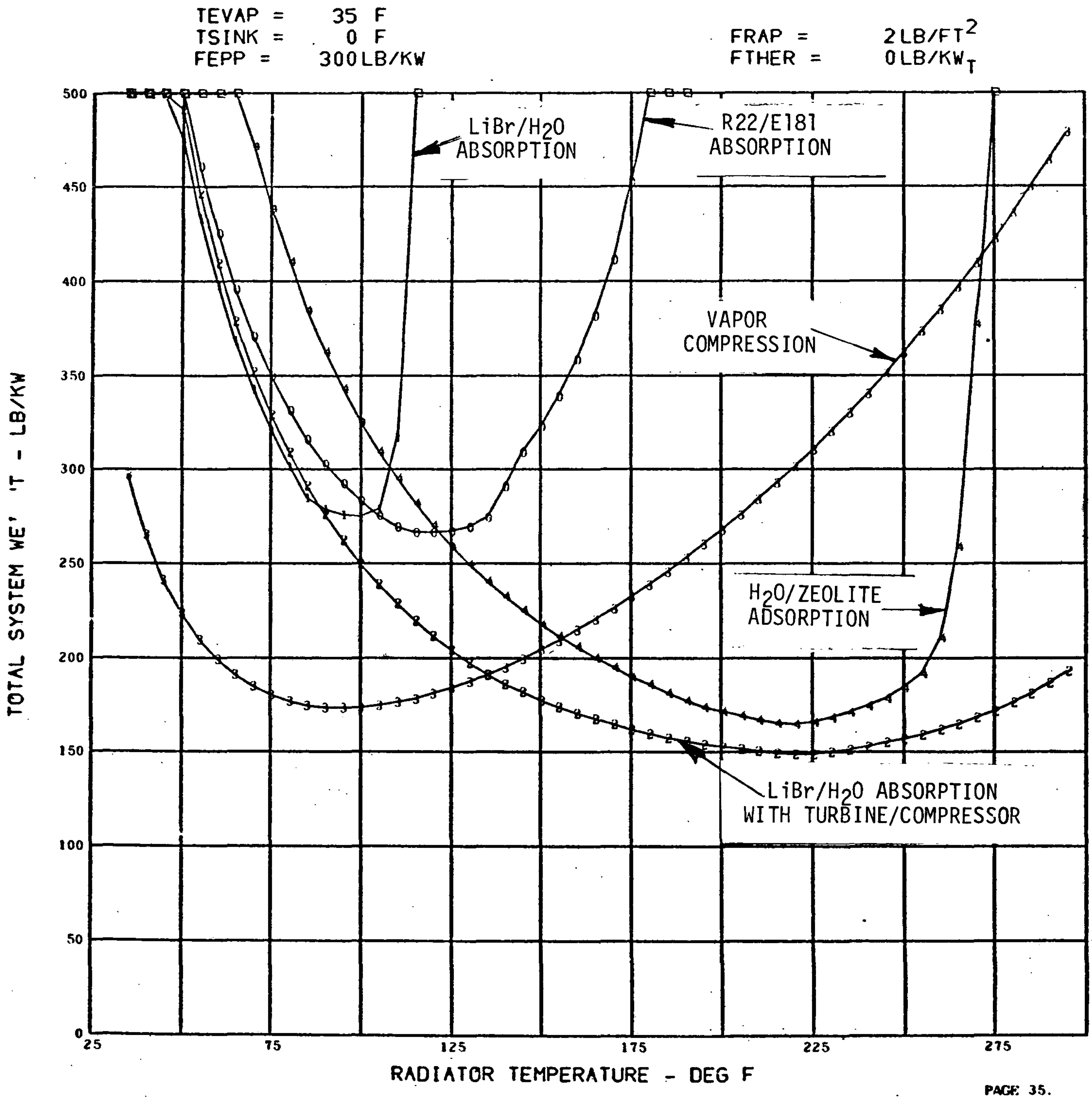
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LiBr/H2O ABSORPTION
R22/E181 ABSORPTION
VAPOR COMPRESSION
H2O/ZEOLITE ADSORPTION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WE' 'T - LB/KW
RADIATOR TEMPERATURE - DEG F
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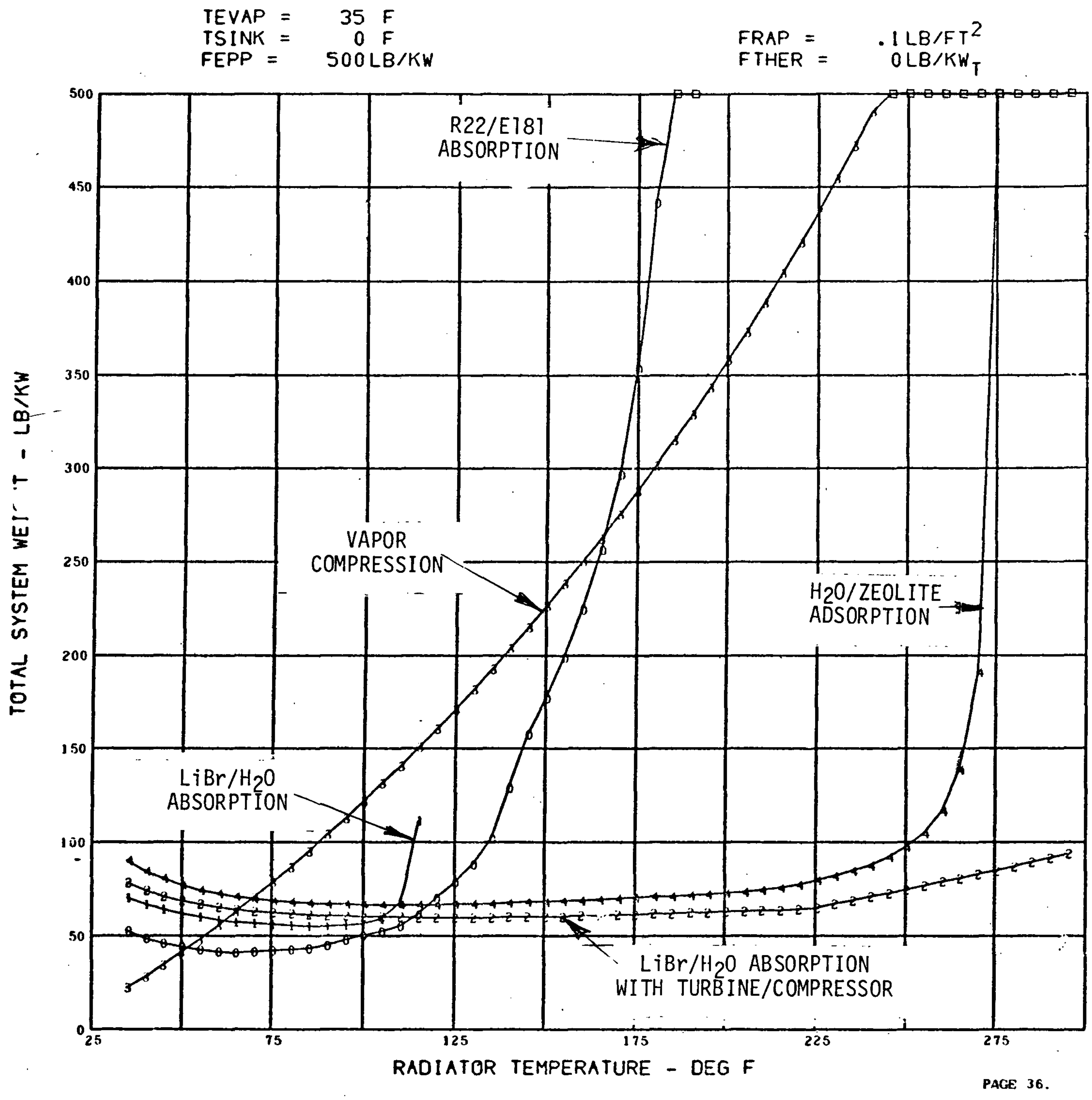
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TSINK = 0 F
FEPP = 500LB/KW
FRAP = .1LB/FT²
FTHER = 0LB/KW_T
R22/E181 ABSORPTION
VAPOR COMPRESSION
H2O/ZEOLITE ADSORPTION
LiBr/H2O ABSORPTION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEI^ T - LB/KW
RADIATOR TEMPERATURE - DEG F
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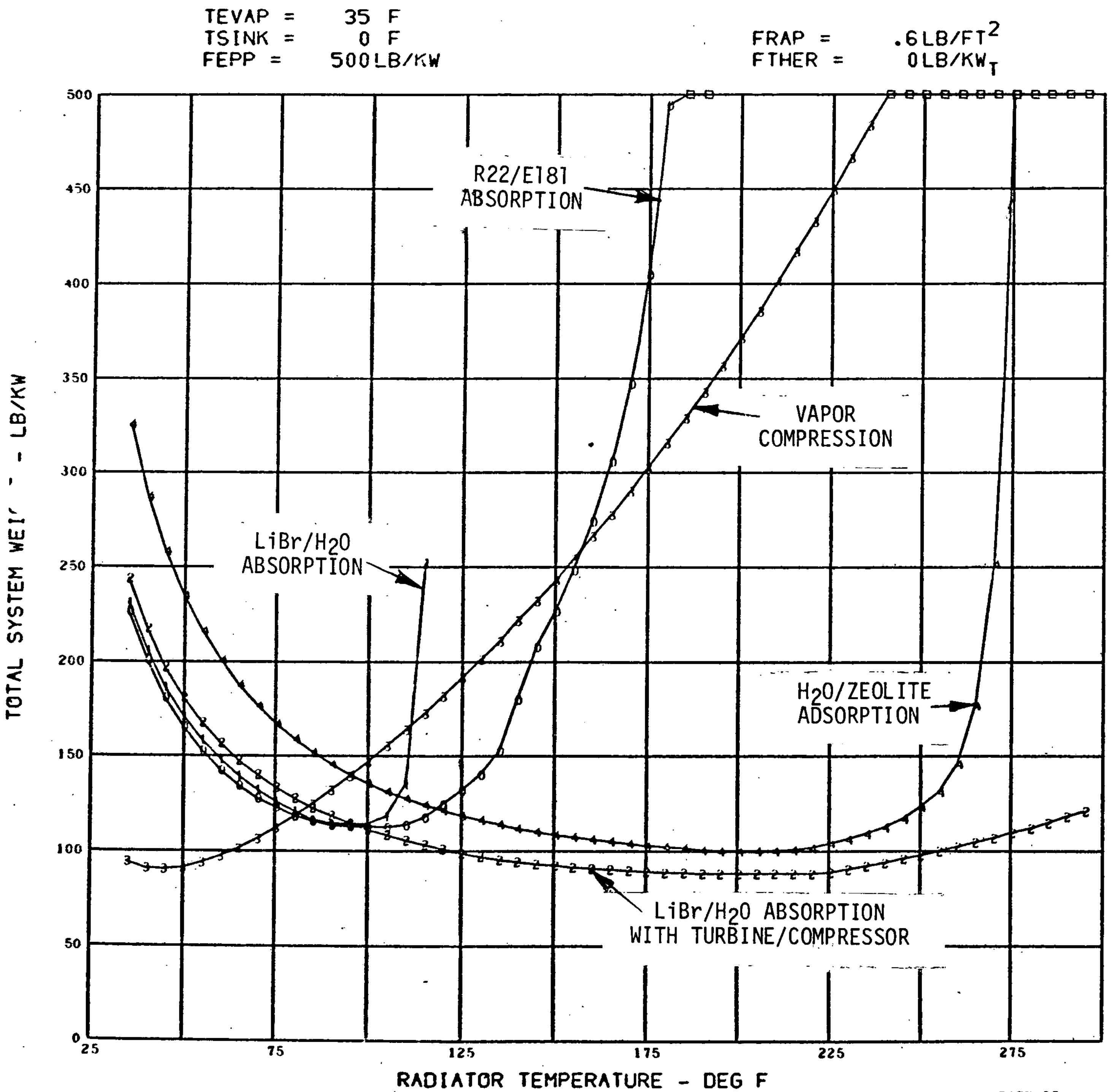
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TSINK = 0 F
FEPP = 500 LB/KW
FRAP = .6 LB/FT2
FTHER = 0 LB/KWT
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VAPOR COMPRESSION
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LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEI' - LB/KW
RADIATOR TEMPERATURE - DEG F
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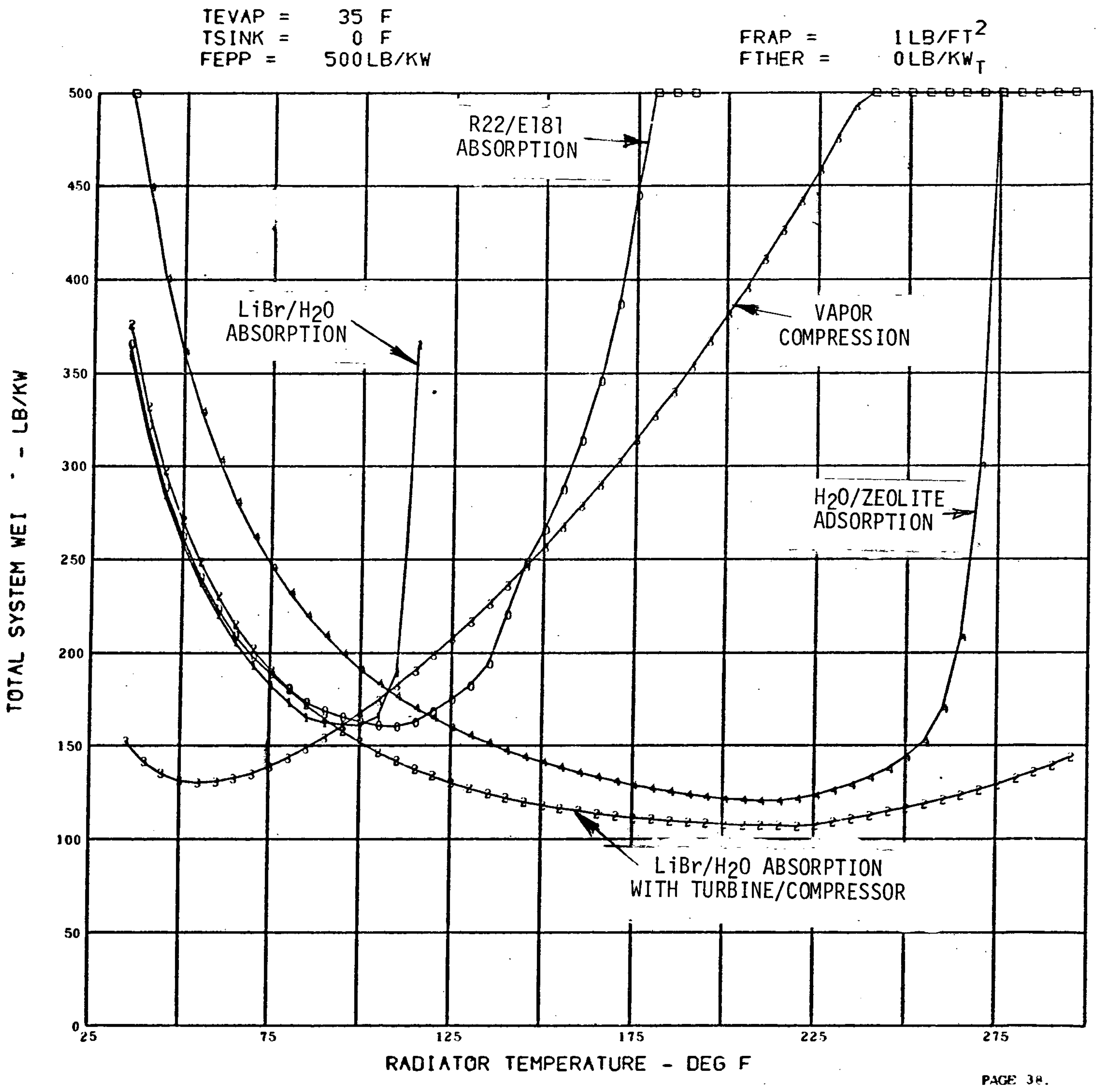
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TSINK = 0 F
FEPP = 500LB/KW
FRAP = 1LB/FT2
FTHER = 0LB/KWT
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LiBr/H2O ABSORPTION
VAPOR COMPRESSION
H2O/ZEOLITE ADSORPTION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEI - LB/KW
RADIATOR TEMPERATURE - DEG F
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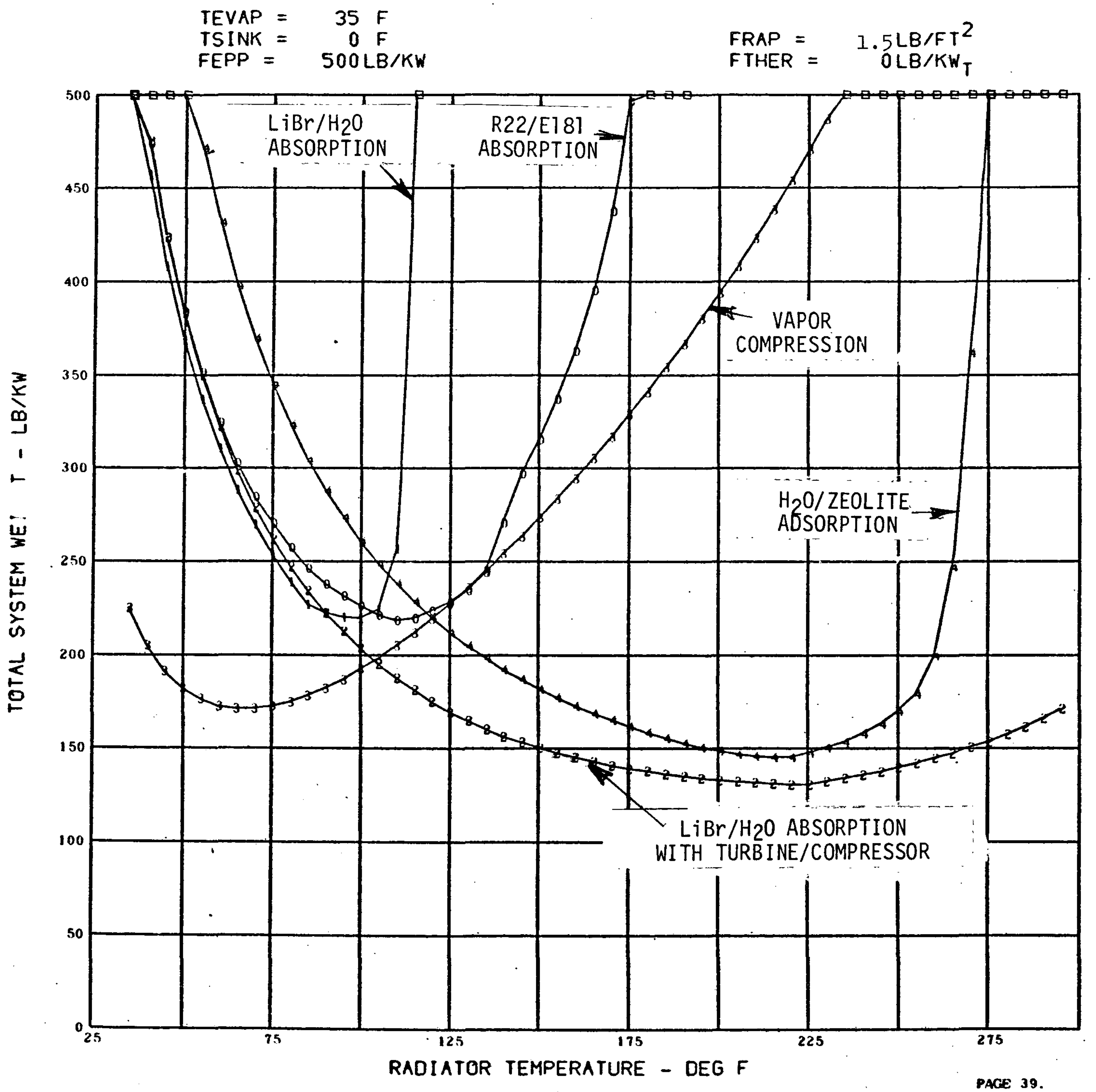
PAGE 38.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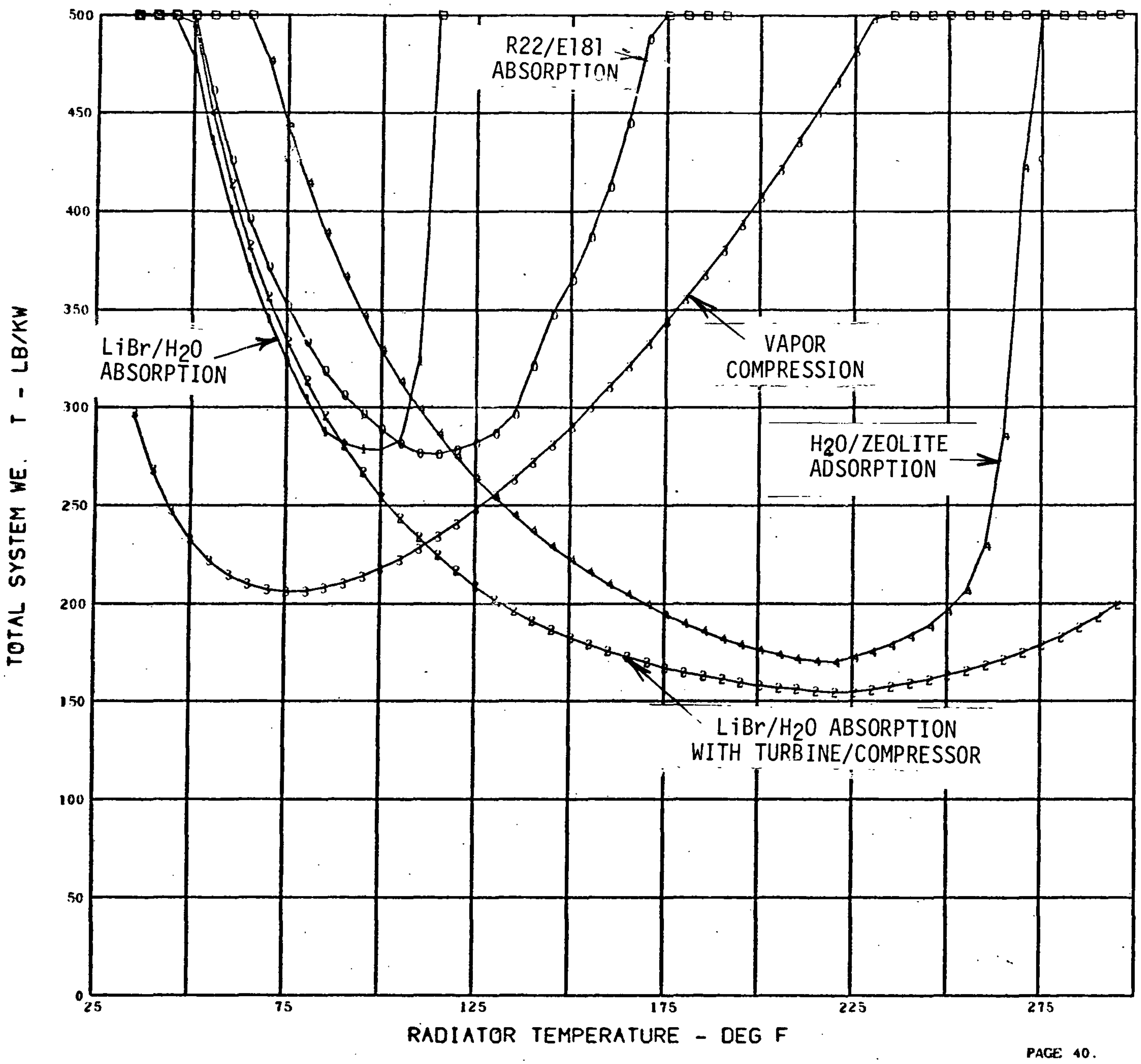
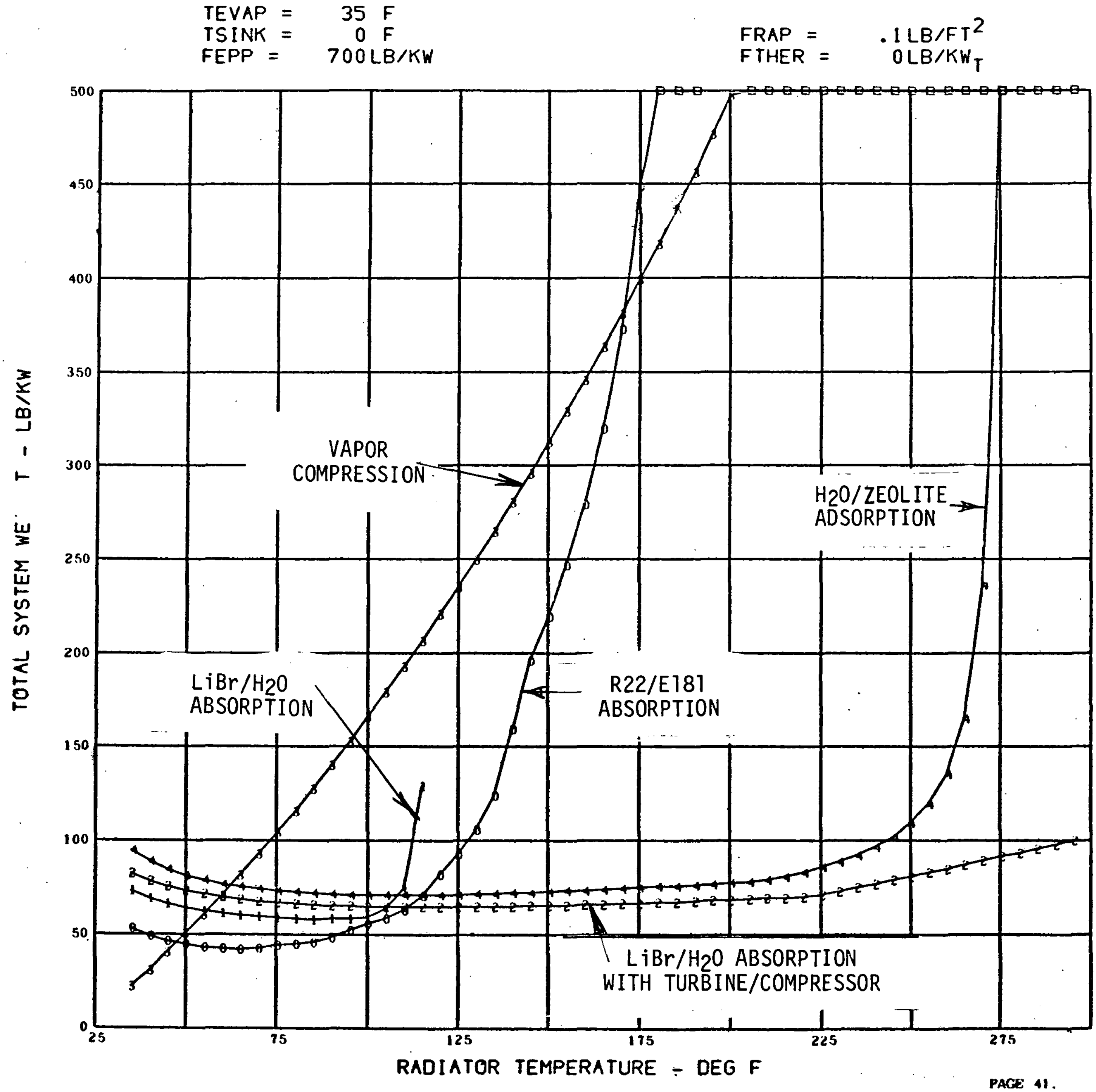
FRAP = 1.5LB/FT²
FTHER = 0LB/KW_T
LiBr/H2O ABSORPTION
R22/E181 ABSORPTION
VAPOR COMPRESSION
H2O/ZEOLITE ADSORPTION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WE! T - LB/KW
RADIATOR TEMPERATURE - DEG F
PAGE 39.

TEVAP = 35 F
TSINK = 0 F
FEPP = 500 LB/KW

FRAP = 2 LB/FT^2
FTHER = 0 LB/KW$_T$





TEVAP = 35 F
TSINK = 0 F
FEPP = 700 LB/KW
FRAP = .1 LB/FT2
FTHER = 0 LB/KWT
TOTAL SYSTEM WE T - LB/KW
RADIATOR TEMPERATURE - DEG F
VAPOR COMPRESSION
H2O/ZEOLITE ADSORPTION
LiBr/H2O ABSORPTION
R22/E181 ABSORPTION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
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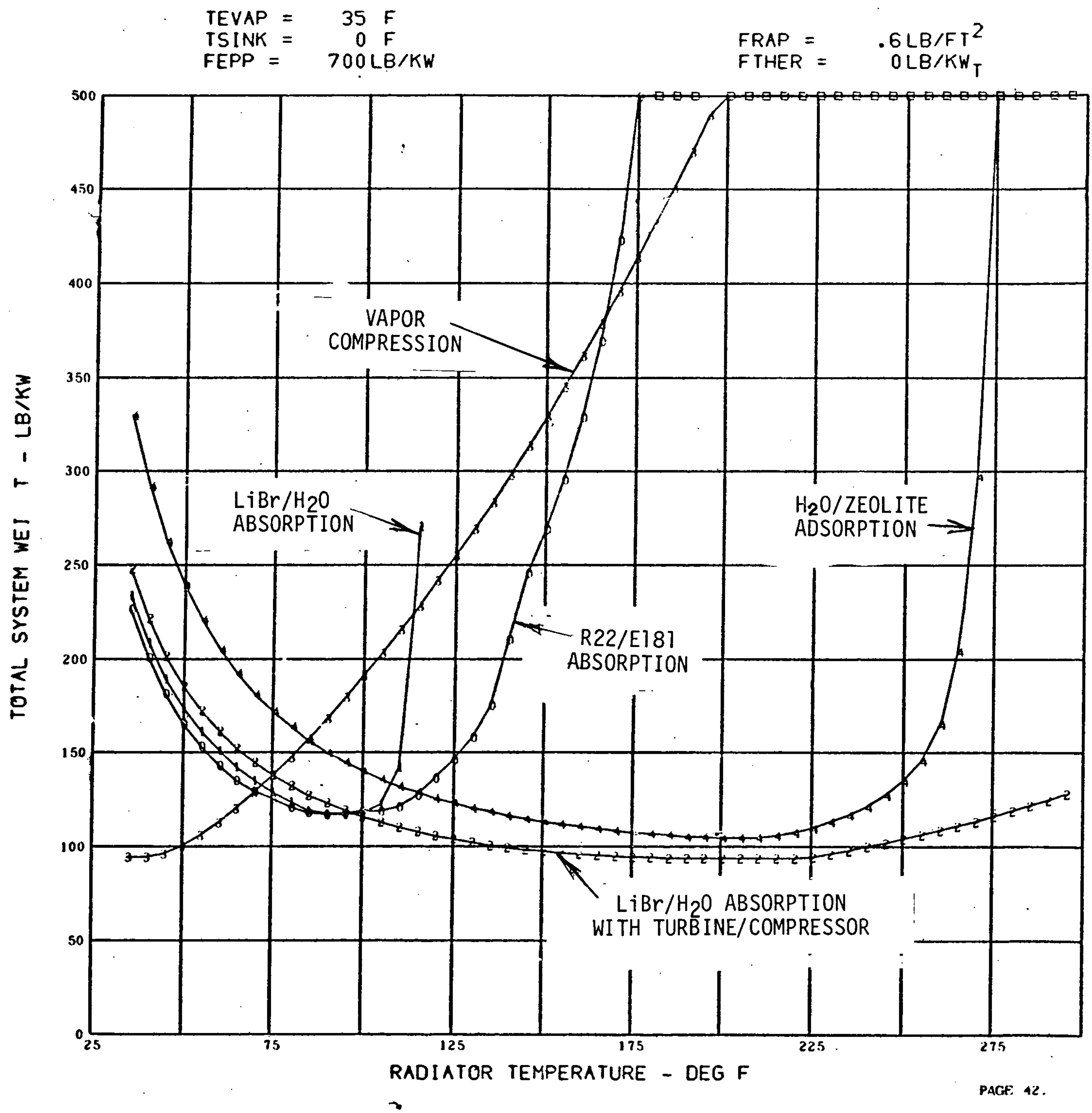
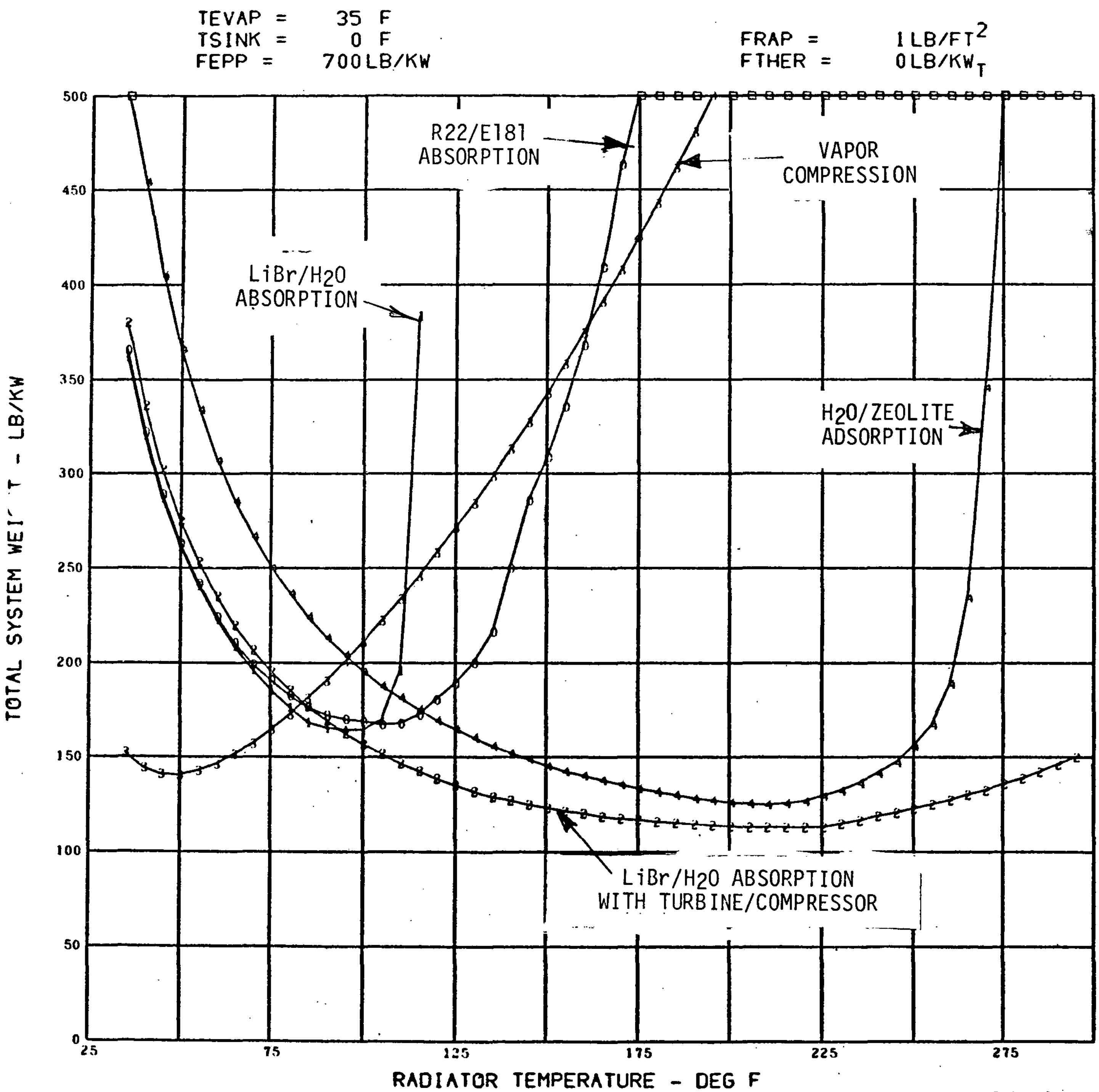
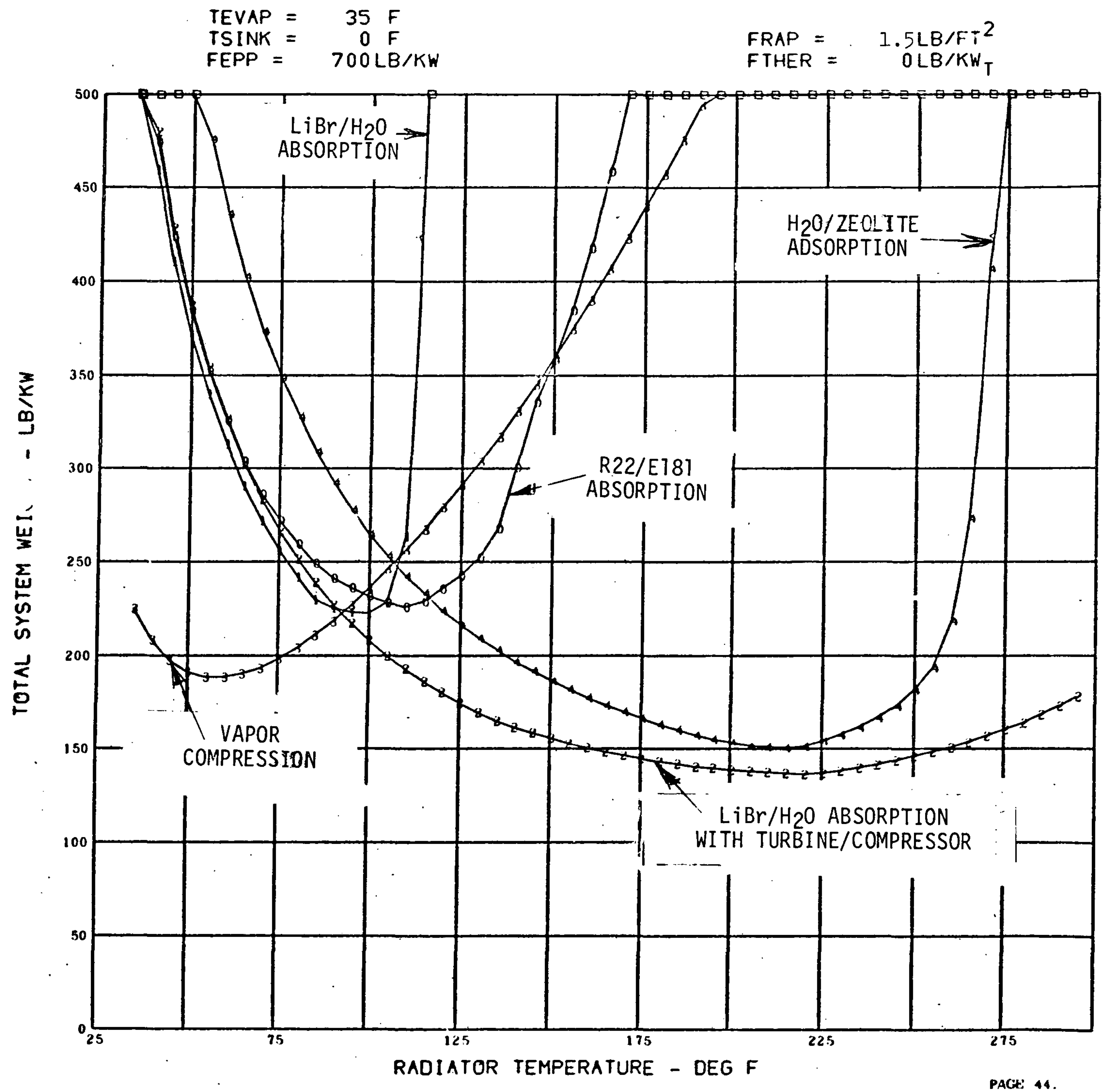
125
175
225
275
PAGE 41.


TEVAP = 35 F
TSINK = 0 F
FEPP = 700 LB/KW
FRAP = .6 LB/FT2
FTHER = 0 LB/KWT
VAPOR COMPRESSION
LiBr/H2O ABSORPTION
H2O/ZEOLITE ADSORPTION
R22/E181 ABSORPTION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEI T - LB/KW
RADIATOR TEMPERATURE - DEG F
500
450
400
350
300
250
200
150
100
50
0
25
75
125
175
225
275
PAGE 42.



TEVAP = 35 F
TSINK = 0 F
FEPP = 700 LB/KW
FRAP = 1 LB/FT^2
FTHER = 0 LB/KW$_T$
R22/E181 ABSORPTION
VAPOR COMPRESSION
LiBr/H_2O ABSORPTION
H_2O/ZEOLITE ADSORPTION
LiBr/H_2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEIGHT - LB/KW
RADIATOR TEMPERATURE - DEG F
0
50
100
150
200
250
300
350
400
450
500
25
75
125
175
225
275

PAGE 43.


TEVAP = 35 F
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FRAP = 1.5LB/FT2
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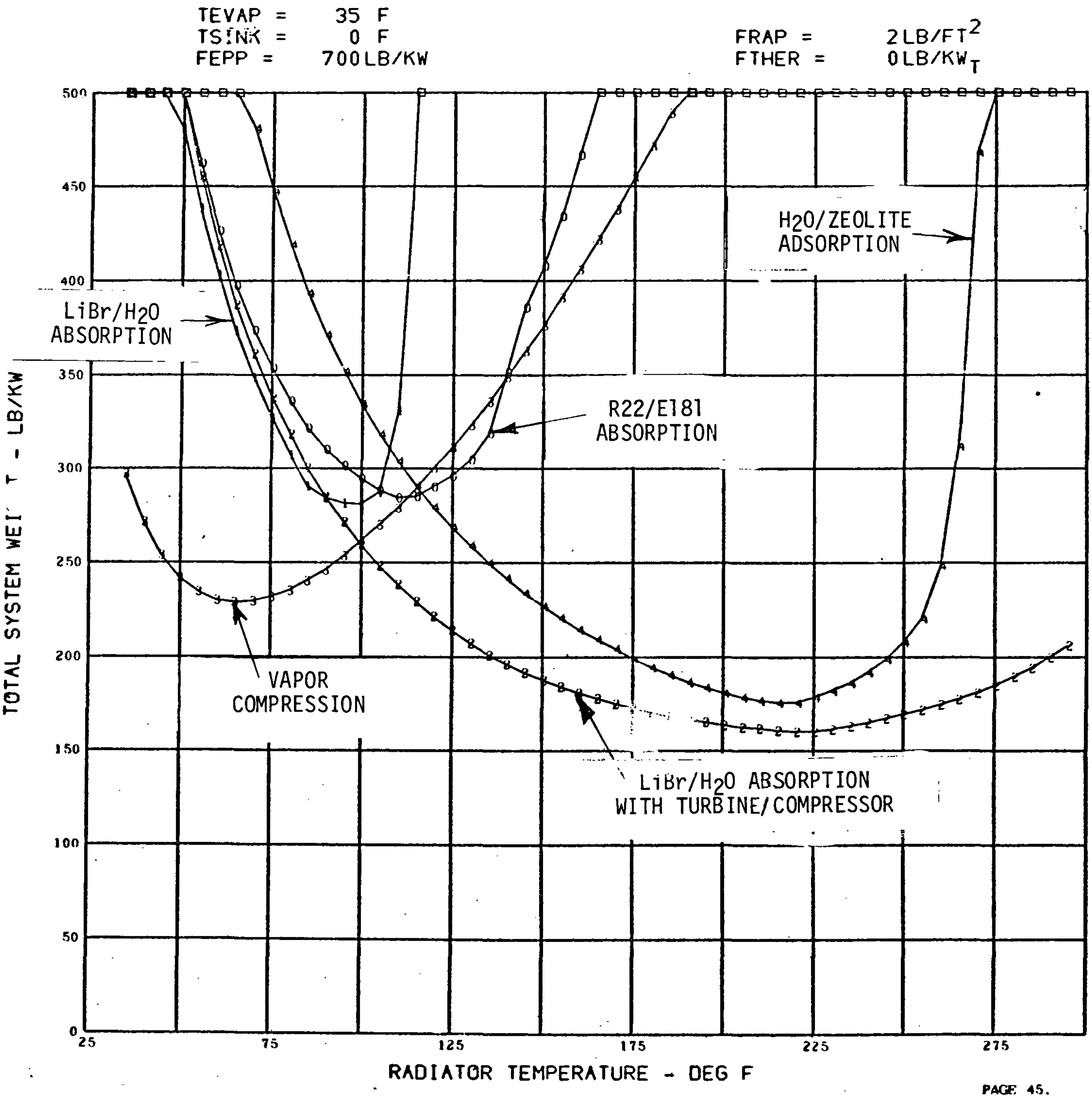
LiBr/H2O ABSORPTION
H2O/ZEOLITE ABSORPTION
R22/E181 ABSORPTION
VAPOR COMPRESSION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEI. - LB/KW
RADIATOR TEMPERATURE - DEG F
PAGE 44.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FRAP = 2 LB/FT2
FTHER = 0 LB/KWT
H2O/ZEOLITE ADSORPTION
LiBr/H2O ABSORPTION
R22/E181 ABSORPTION
VAPOR COMPRESSION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEIGHT - LB/KW
RADIATOR TEMPERATURE - DEG F
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150
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250
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350
400
450
500
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125
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225
275
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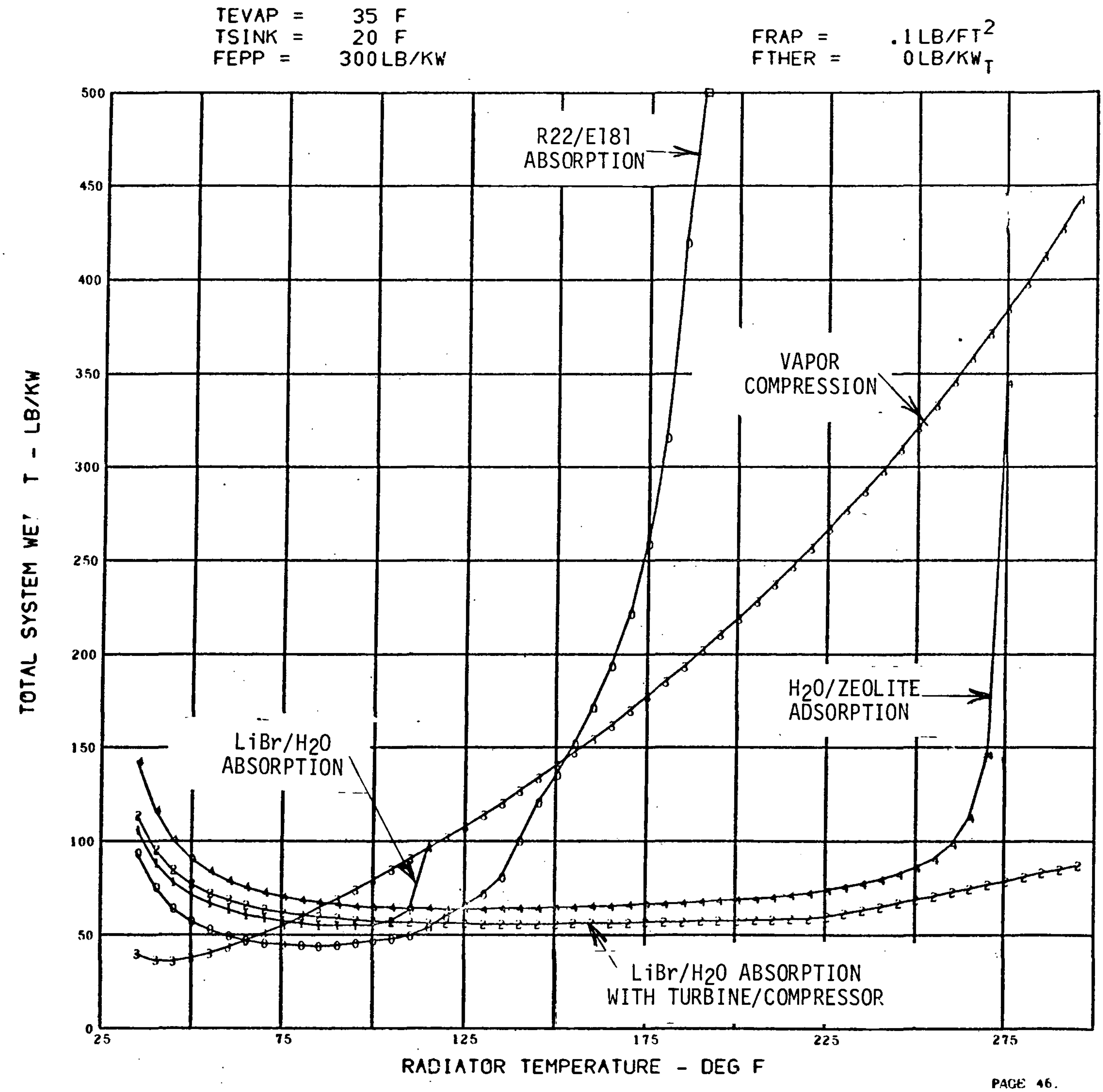

TEVAP = 35 F
TSINK = 20 F
FEPP = 300 LB/KW
FRAP = .1 LB/FT2
FTHER = 0 LB/KWT
R22/E181 ABSORPTION
VAPOR COMPRESSION
H2O/ZEOLITE ADSORPTION
LiBr/H2O ABSORPTION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WE' T - LB/KW
RADIATOR TEMPERATURE - DEG F
0
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100
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200
250
300
350
400
450
500
25
75
125
175
225
275
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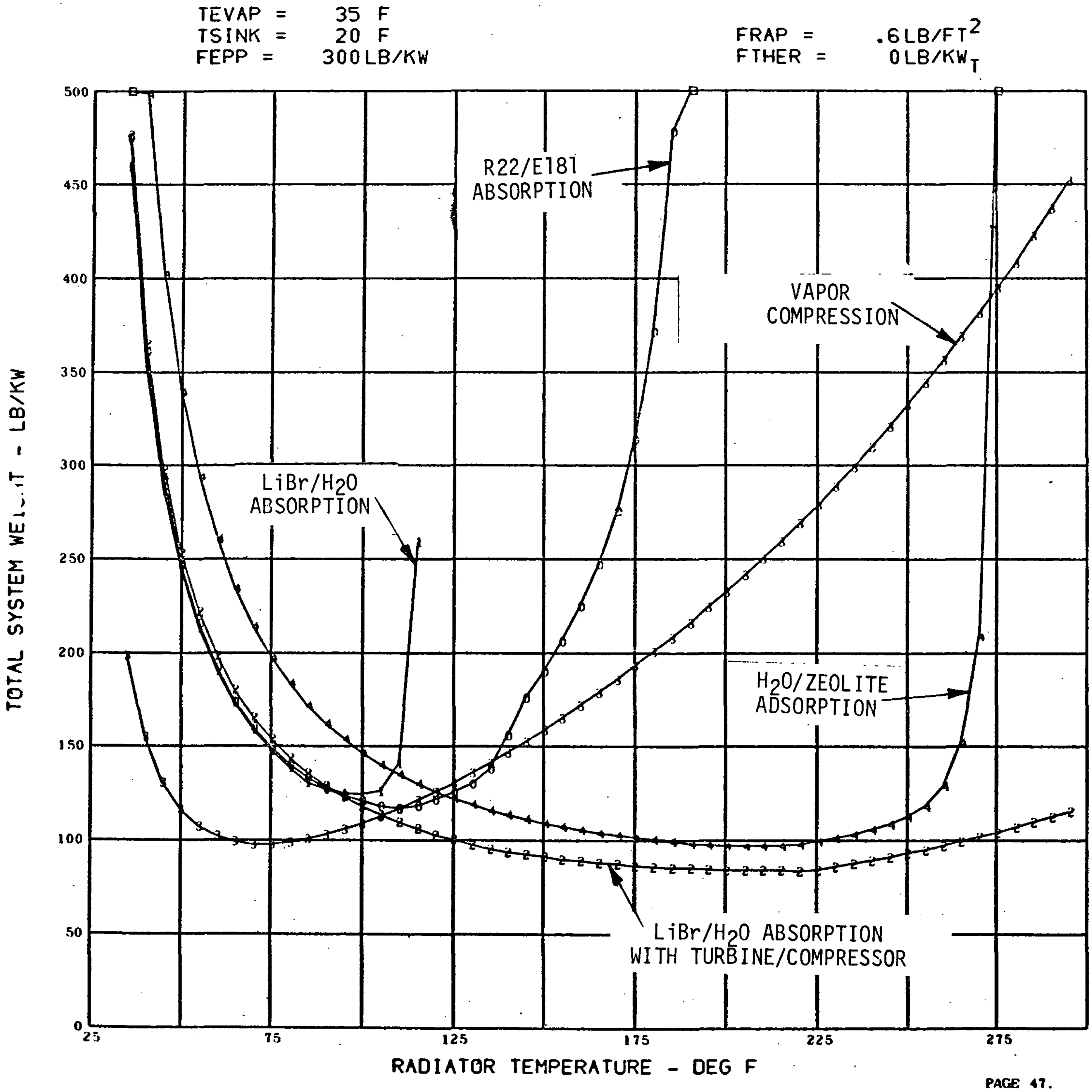

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FRAP = .6 LB/FT2
FTHER = 0 LB/KWT
R22/E181 ABSORPTION
VAPOR COMPRESSION
LiBr/H2O ABSORPTION
H2O/ZEOLITE ADSORPTION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEIGHT - LB/KW
RADIATOR TEMPERATURE - DEG F
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400
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300
250
200
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50
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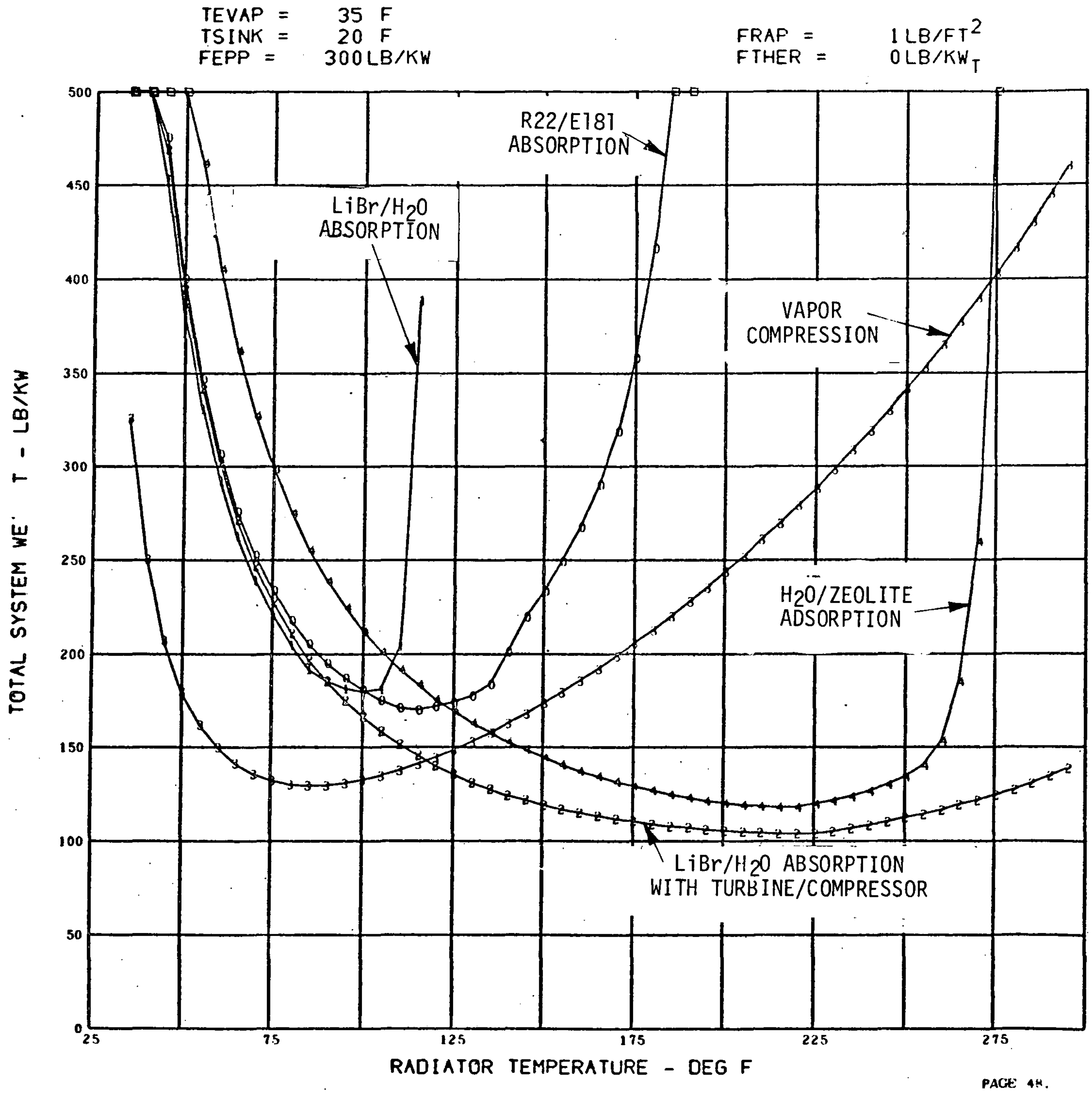
25
75
125
175
225
275
PAGE 47.


TEVAP = 35 F
TSINK = 20 F
FEPP = 300 LB/KW
FRAP = 1 LB/FT2
FTHER = 0 LB/KWT
R22/E181 ABSORPTION
LiBr/H2O ABSORPTION
VAPOR COMPRESSION
H2O/ZEOLITE ADSORPTION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WE' T - LB/KW
RADIATOR TEMPERATURE - DEG F
0
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450
500
25
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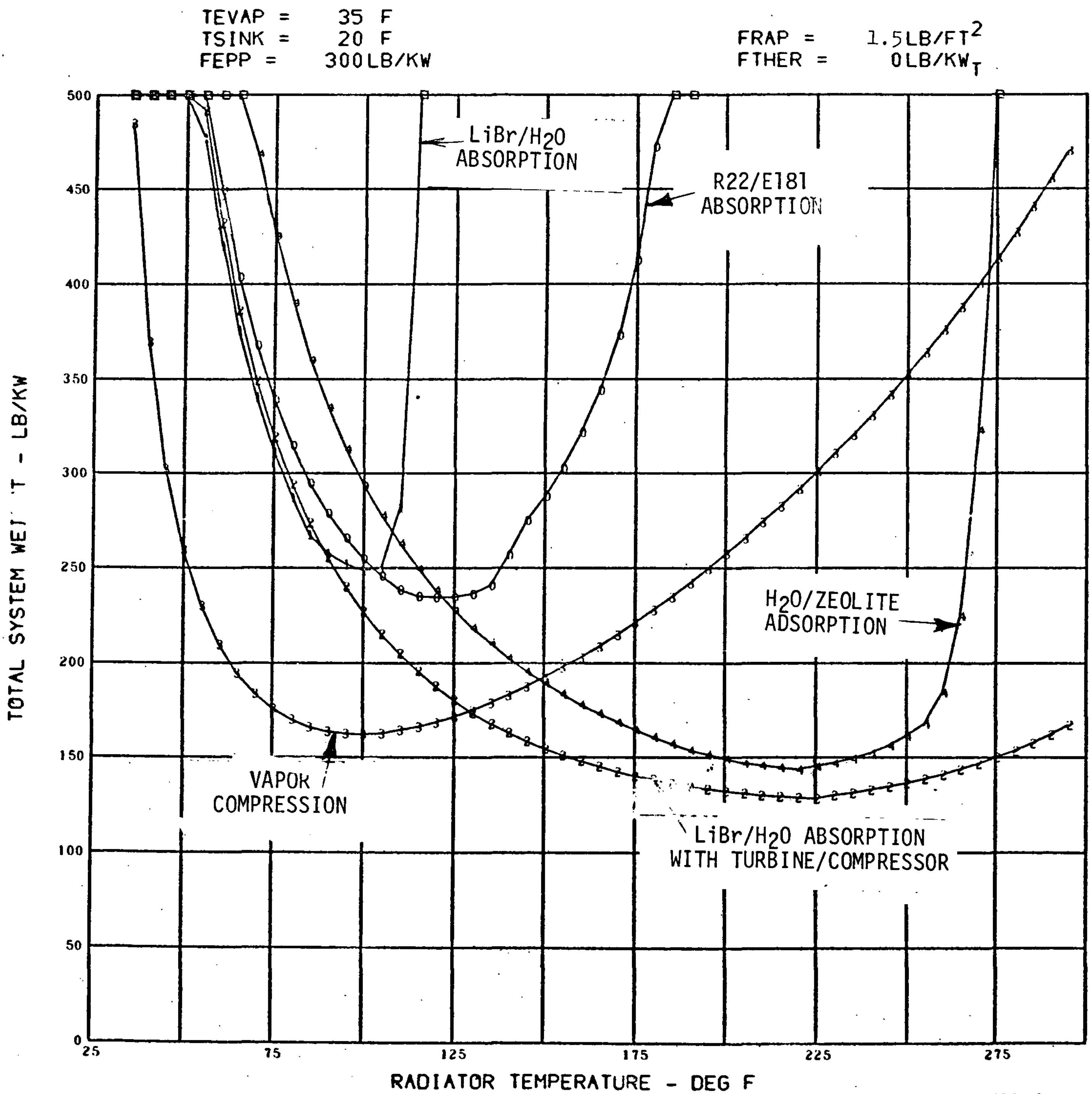
125
175
225
275
PAGE 48.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FEPP = 300LB/KW
FRAP = 1.5LB/FT2
FTHER = 0LB/KWT
LiBr/H2O ABSORPTION
R22/E181 ABSORPTION
H2O/ZEOLITE ADSORPTION
VAPOR COMPRESSION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEI 'T - LB/KW
RADIATOR TEMPERATURE - DEG F
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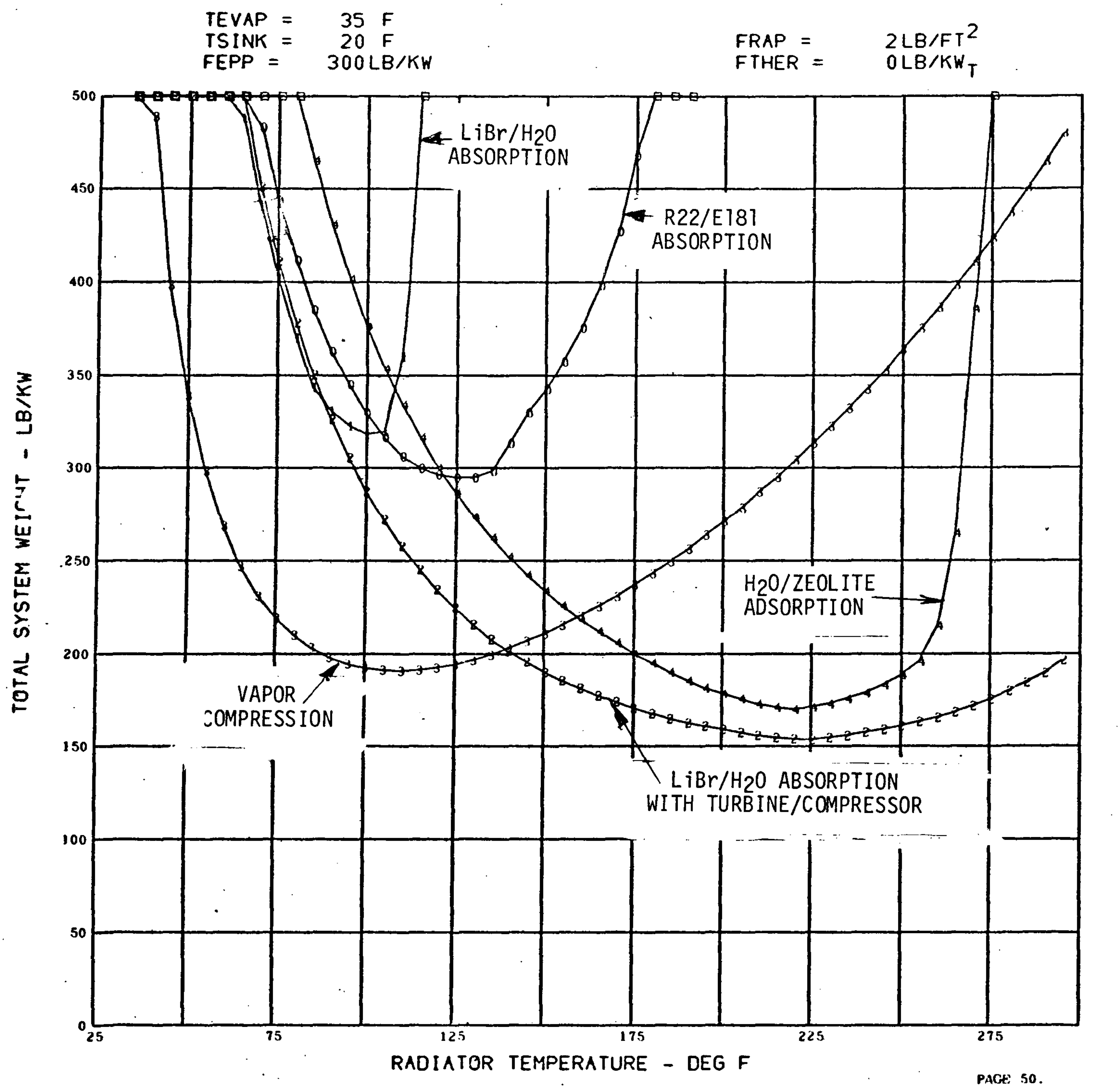
150
200
250
300
350
400
450
500
25
75
125
175
225
275

PAGE 49.



TEVAP = 35 F
TSINK = 20 F
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FRAP = 2 LB/FT2
FTHER = 0 LB/KWT
LiBr/H2O ABSORPTION
R22/E181 ABSORPTION
H2O/ZEOLITE ADSORPTION
VAPOR COMPRESSION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEIGHT - LB/KW
RADIATOR TEMPERATURE - DEG F
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25
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125
175
225
275
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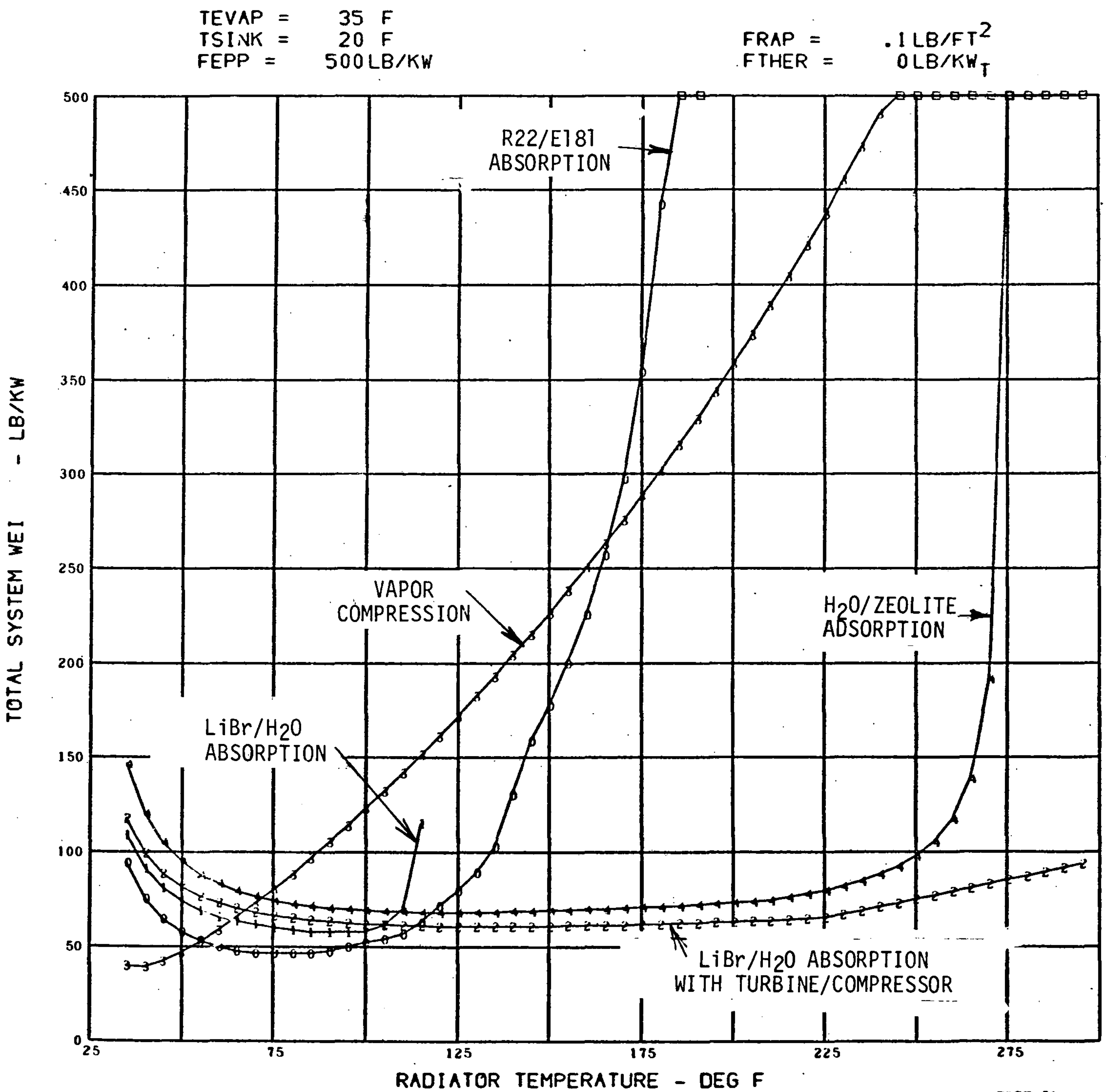
TEVAP = 35 F
TSINK = 20 F
FEPP = 500 LB/KW
FRAP = .1 LB/FT2
FTHER = 0 LB/KWT
R22/E181 ABSORPTION
VAPOR COMPRESSION
H2O/ZEOLITE ADSORPTION
LiBr/H2O ABSORPTION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEI - LB/KW
RADIATOR TEMPERATURE - DEG F
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150
200
250
300
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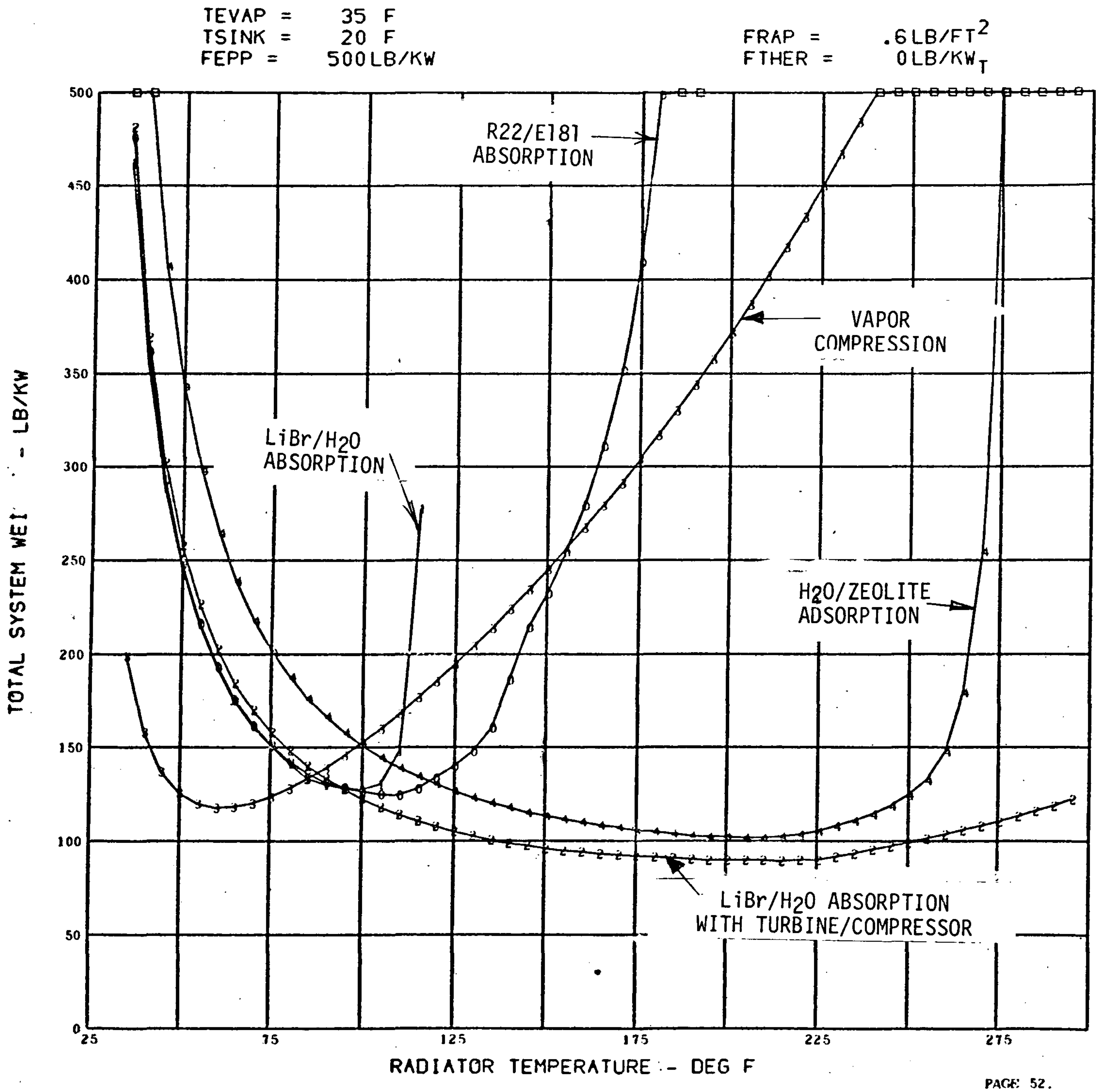
400
450
500
25
75
125
175
225
275

PAGE 51.


TEVAP = 35 F
TSINK = 20 F
FEPP = 500LB/KW
FRAP = .6LB/FT^2
FTHER = 0LB/KW$_T$
R22/E181 ABSORPTION
VAPOR COMPRESSION
LiBr/H_2O ABSORPTION
H_2O/ZEOLITE ADSORPTION
LiBr/H_2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEI - LB/KW
RADIATOR TEMPERATURE - DEG F
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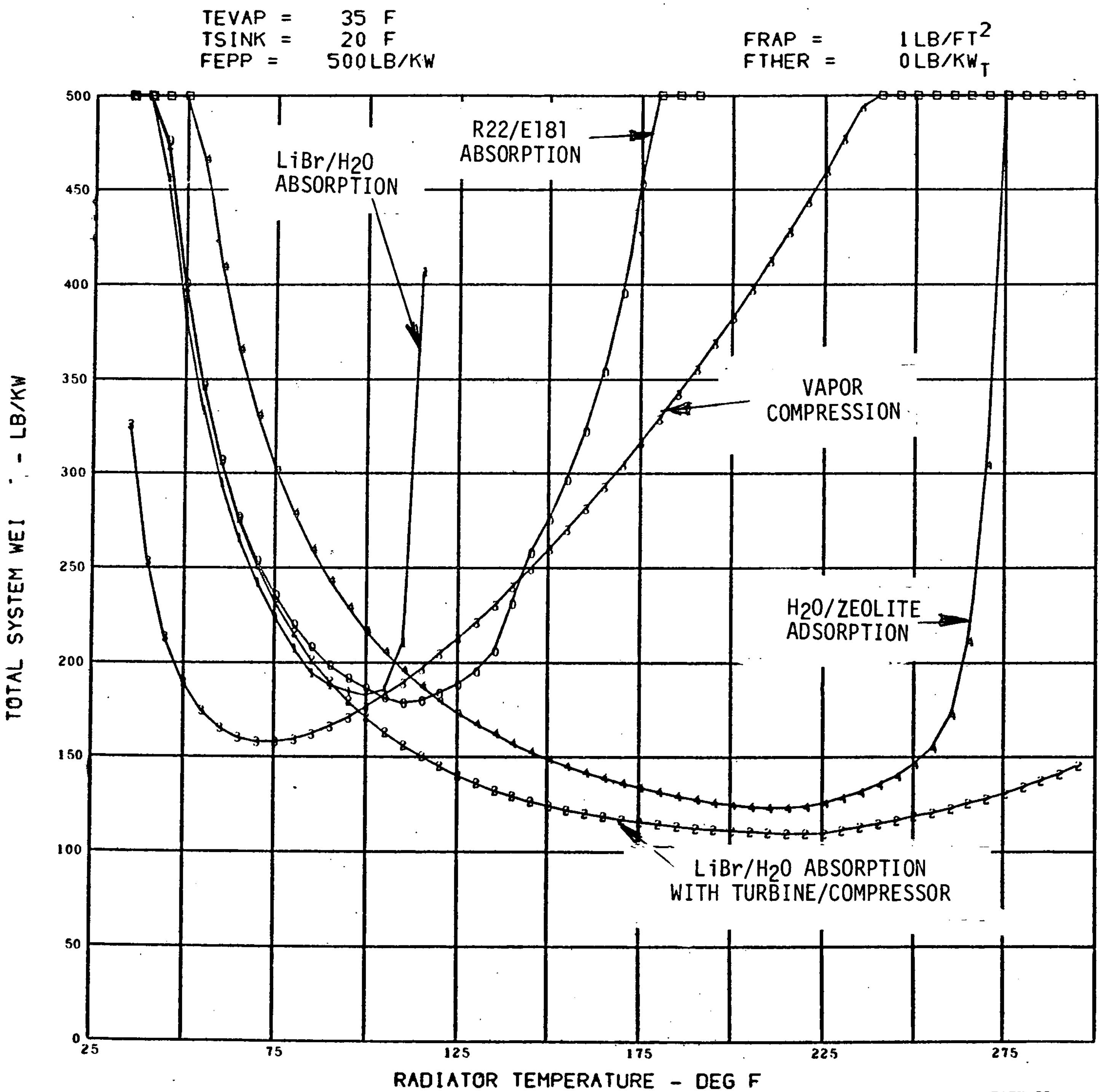
450
500
25
75
125
175
225
275
PAGE 52.



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TSINK = 20 F
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FRAP = 1LB/FT2
FTHER = 0LB/KWT
R22/E181 ABSORPTION
LiBr/H2O ABSORPTION
VAPOR COMPRESSION
H2O/ZEOLITE ADSORPTION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEI - LB/KW
RADIATOR TEMPERATURE - DEG F
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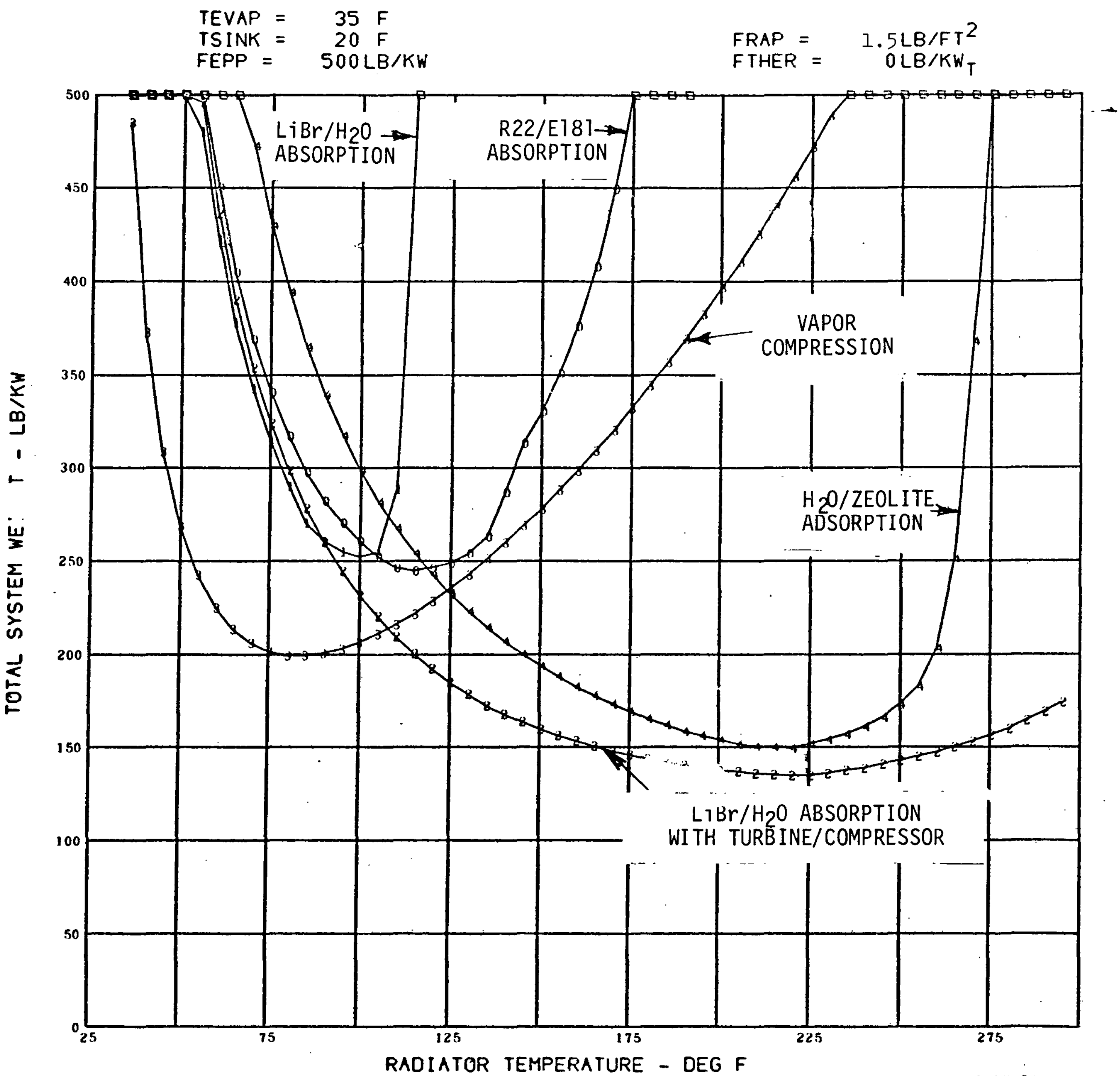
200
250
300
350
400
450
500
25
75
125
175
225
275

PAGE 53.



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TSINK = 20 F
FEPP = 500LB/KW
FRAP = 1.5LB/FT^2
FTHER = 0LB/KW$_T$
LiBr/H$_2$O ABSORPTION
R22/E181 ABSORPTION
VAPOR COMPRESSION
H$_2$O/ZEOLITE ADSORPTION
LiBr/H$_2$O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEIGHT - LB/KW
RADIATOR TEMPERATURE - DEG F
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75
125
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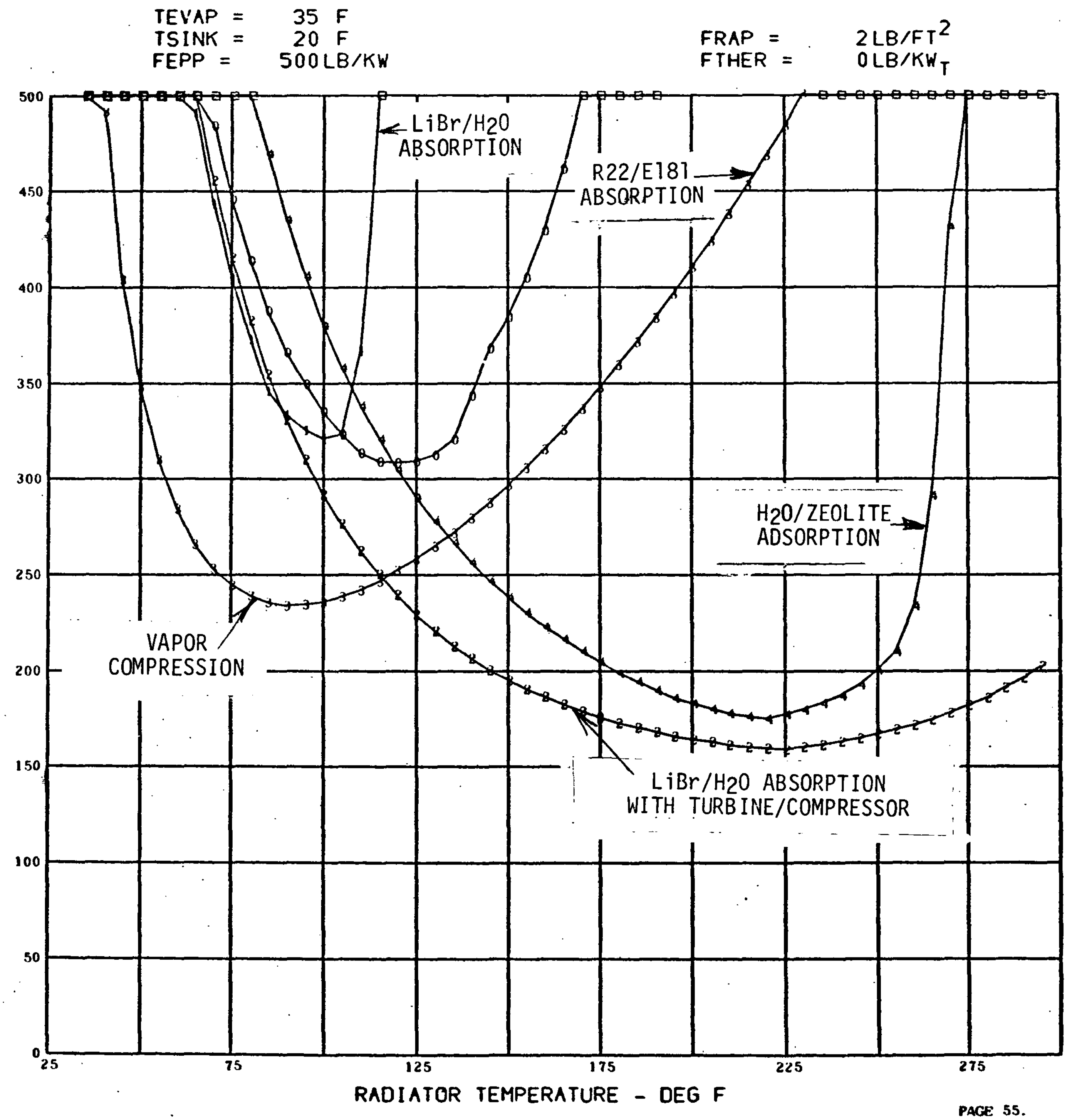
275

PAGE 54.



TEVAP = 35 F
TSINK = 20 F
FEPP = 500 LB/KW
FRAP = 2 LB/FT2
FTHER = 0 LB/KWT
LiBr/H2O ABSORPTION
R22/E181 ABSORPTION
H2O/ZEOLITE ADSORPTION
VAPOR COMPRESSION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
500
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300
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200
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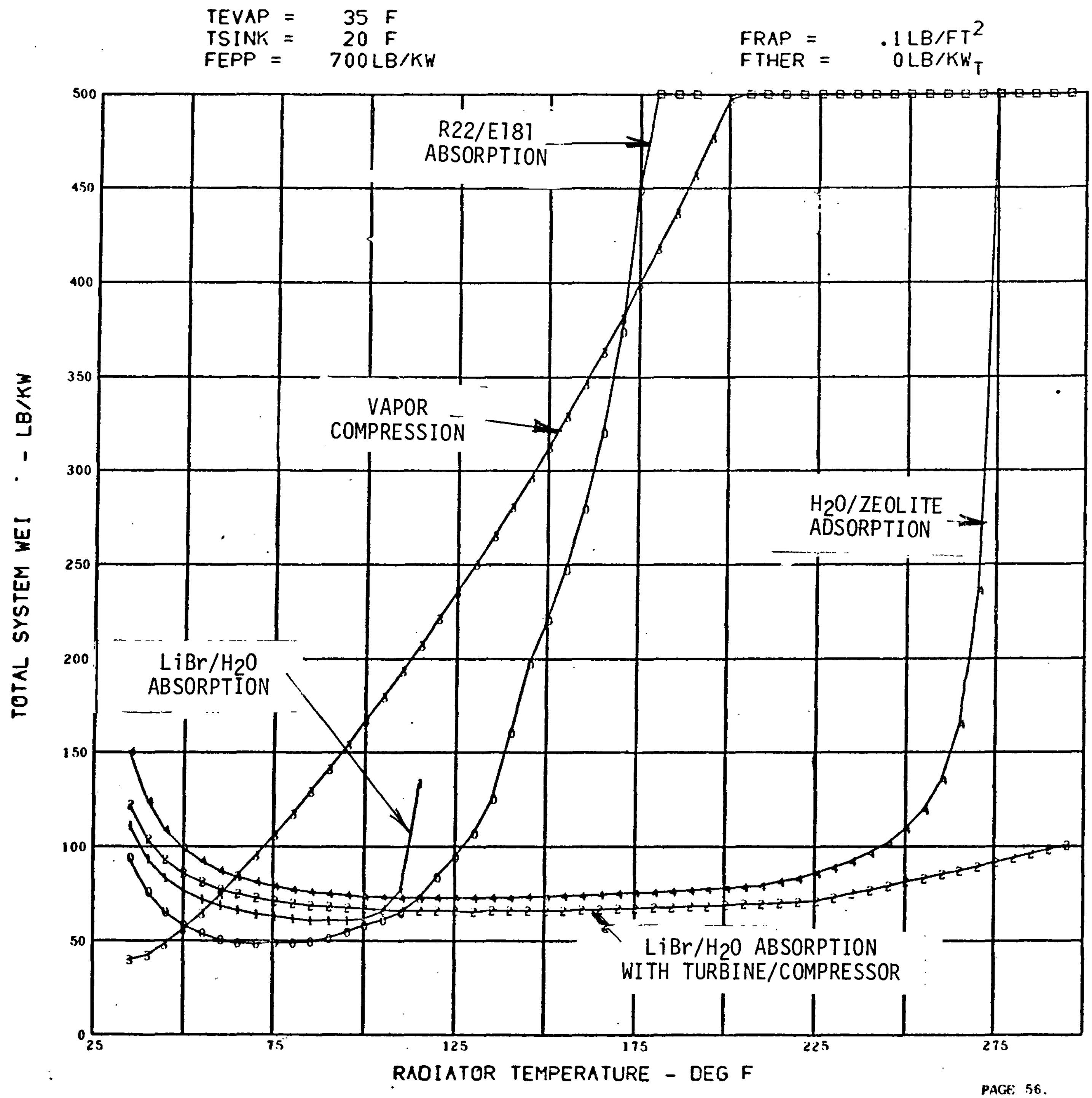
75
125
175
225
275
TOTAL SYSTEM WEI T - LB/KW
RADIATOR TEMPERATURE - DEG F
PAGE 55.



TEVAP = 35 F
TSINK = 20 F
FEPP = 700LB/KW
FRAP = .1LB/FT2
FTHER = 0LB/KWT
R22/E181 ABSORPTION
VAPOR COMPRESSION
H2O/ZEOLITE ADSORPTION
LiBr/H2O ABSORPTION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEI · - LB/KW
RADIATOR TEMPERATURE - DEG F
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100
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200
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300
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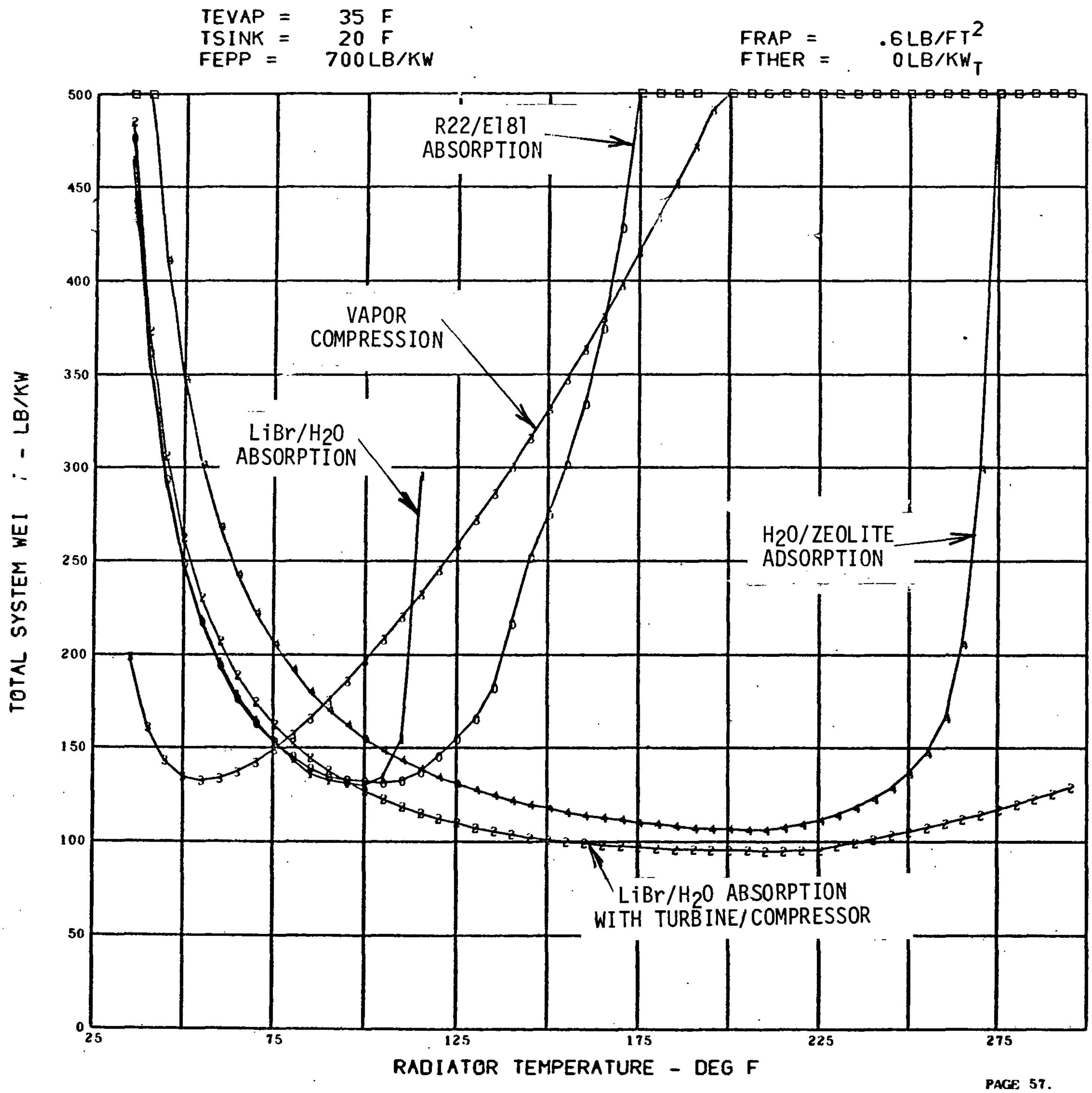
400
450
500
25
75
125
175
225
275
PAGE 56.


TEVAP = 35 F
TSINK = 20 F
FEPP = 700LB/KW
FRAP = .6LB/FT2
FTHER = 0LB/KWT
R22/E181 ABSORPTION
VAPOR COMPRESSION
LiBr/H2O ABSORPTION
H2O/ZEOLITE ADSORPTION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEI ; - LB/KW
RADIATOR TEMPERATURE - DEG F
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200
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300
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400
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500
25
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125
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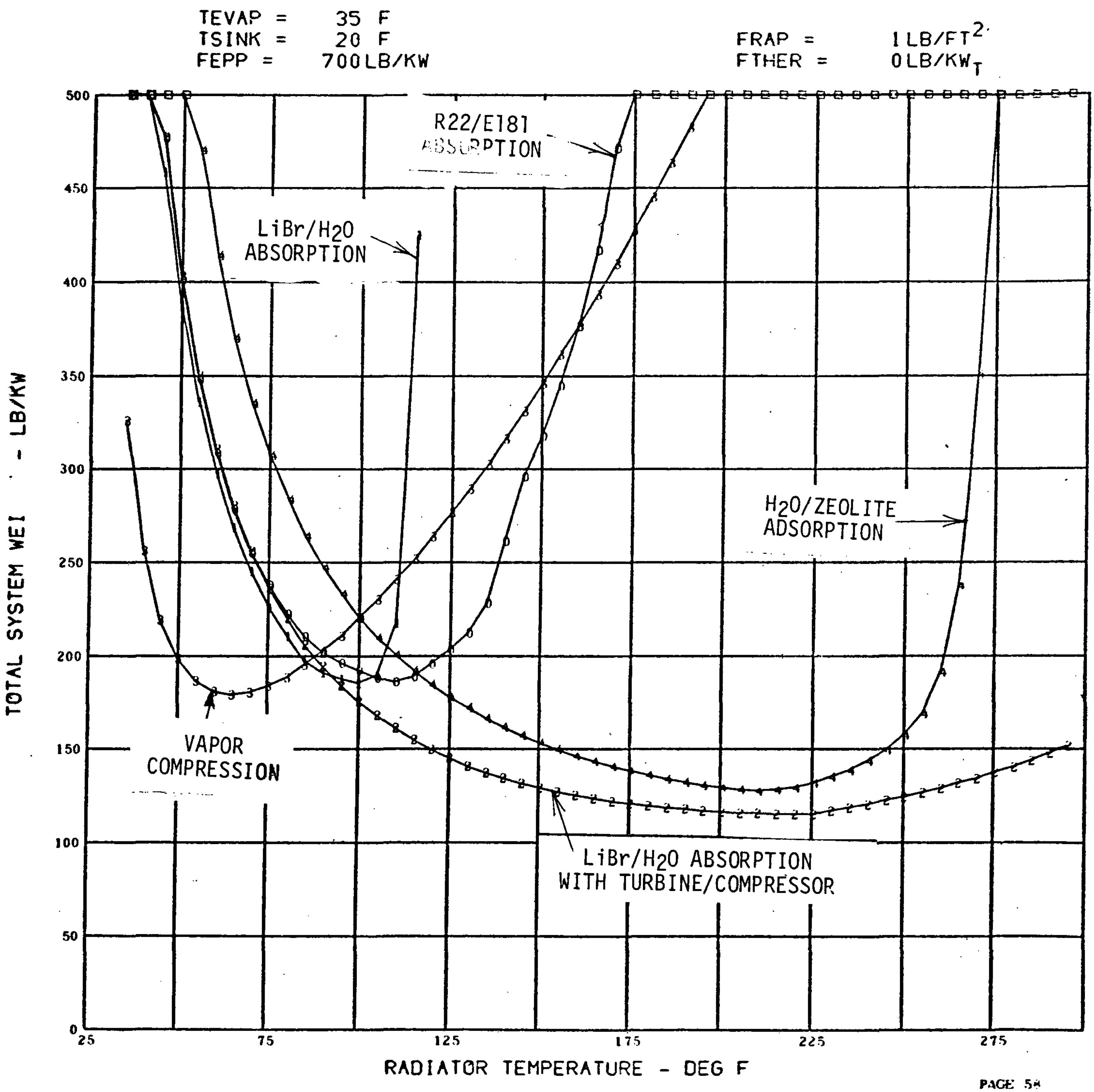
PAGE 57.



TEVAP = 35 F
TSINK = 20 F
FEPP = 700 LB/KW
FRAP = 1 LB/FT2
FTHER = 0 LB/KWT
R22/E181 ABSORPTION
LiBr/H2O ABSORPTION
H2O/ZEOLITE ADSORPTION
VAPOR COMPRESSION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEI - LB/KW
RADIATOR TEMPERATURE - DEG F
500
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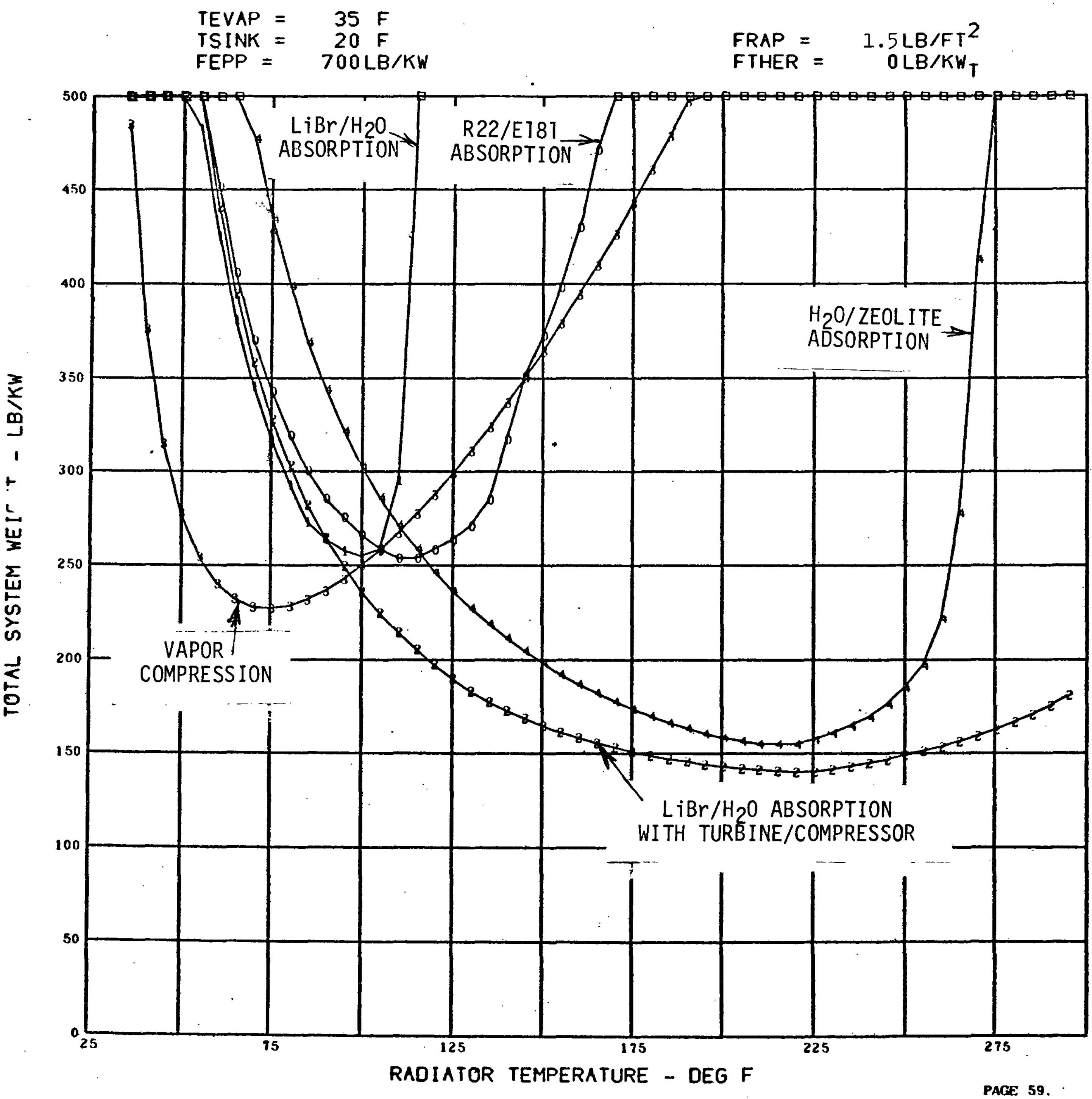
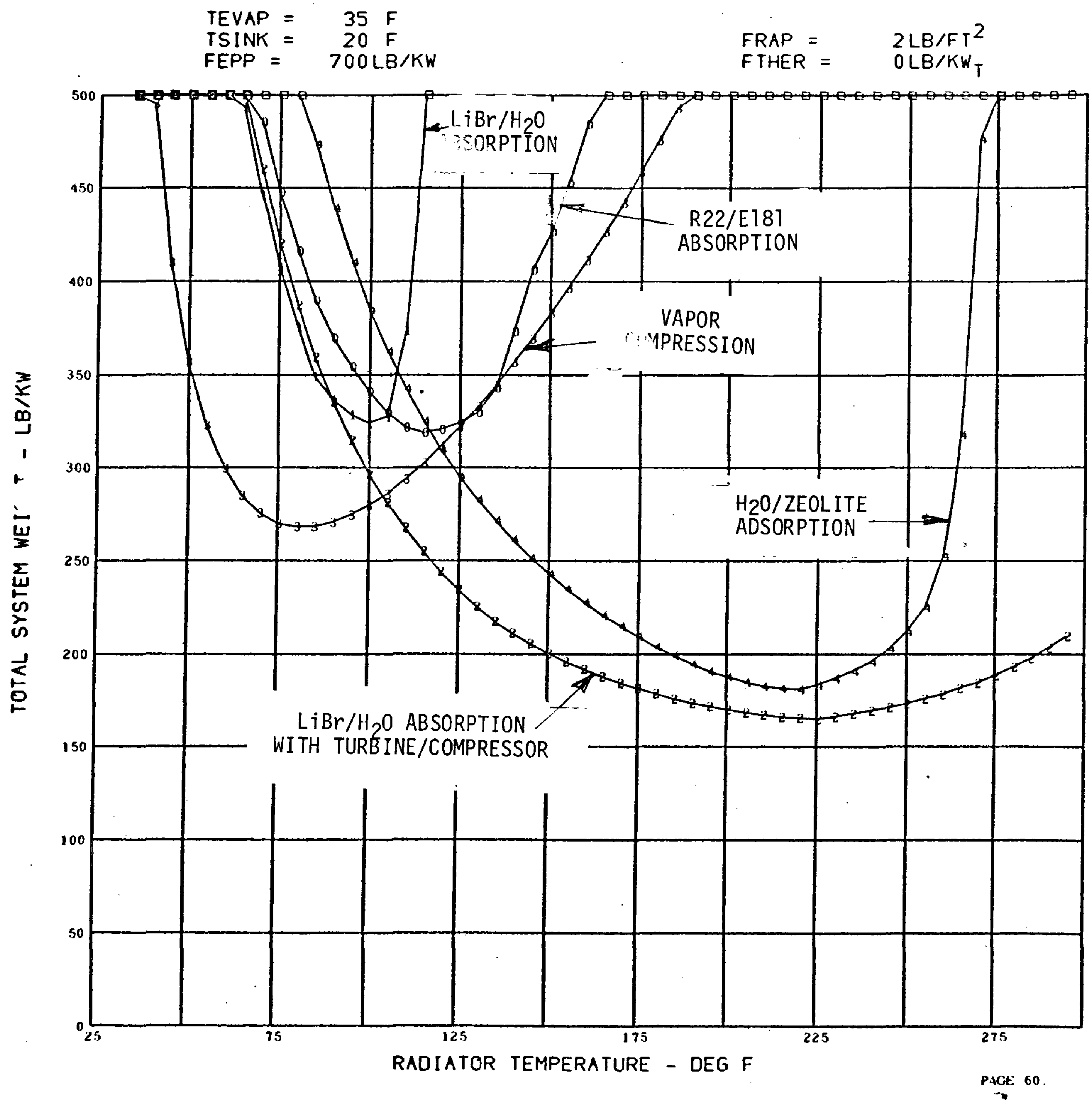
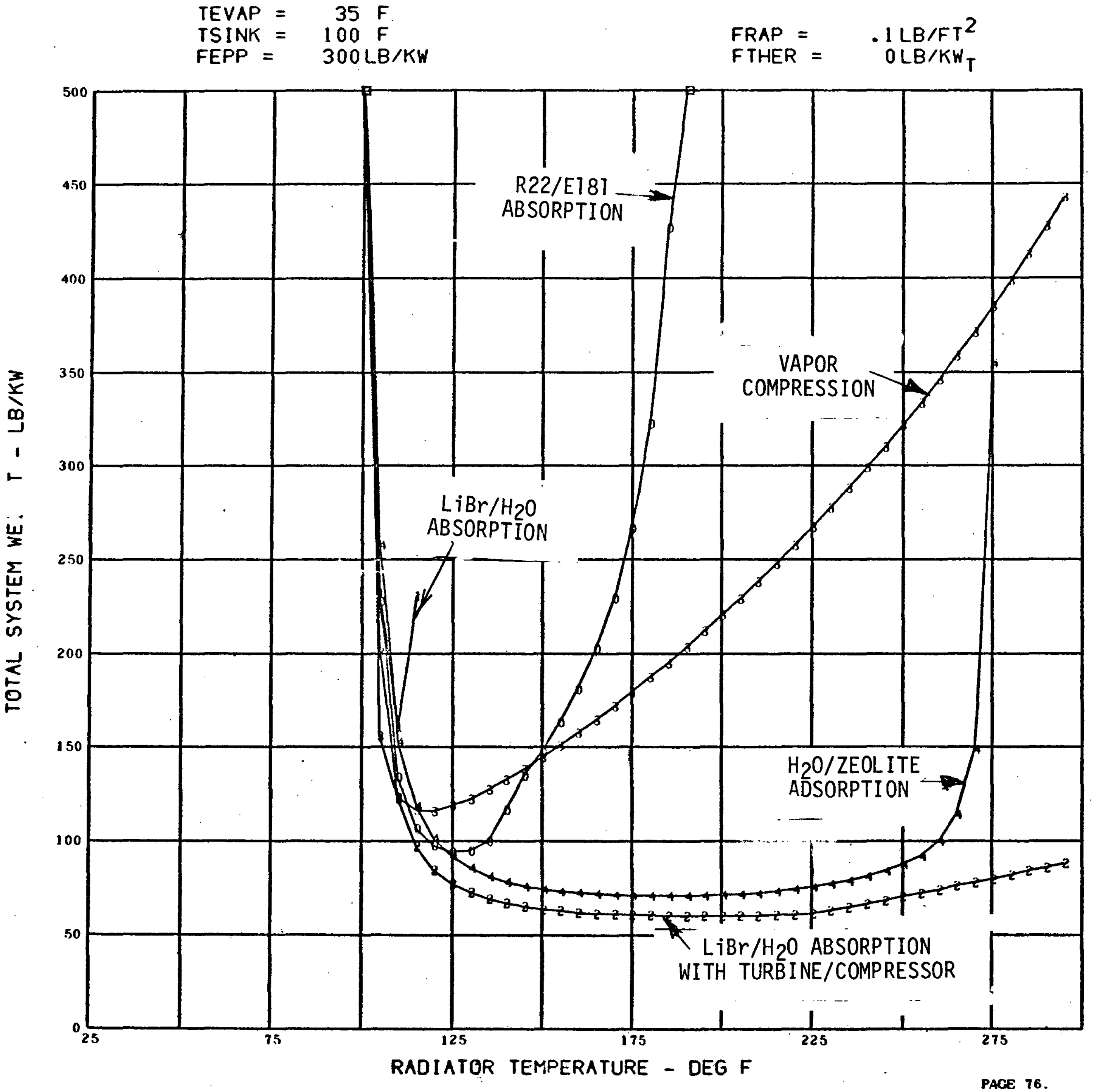
400
350
300
250
200
150
100
50
0
25
75
125
175
225
275
PAGE 58



TEVAP = 35 F
TSINK = 20 F
FEPP = 700LB/KW
FRAP = 1.5LB/FT2
FTHER = 0LB/KWT
LiBr/H2O ABSORPTION
R22/E181 ABSORPTION
H2O/ZEOLITE ADSORPTION
VAPOR COMPRESSION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEI T - LB/KW
RADIATOR TEMPERATURE - DEG F
PAGE 59.



TEVAP = 35 F
TSINK = 20 F
FEPP = 700LB/KW
FRAP = 2LB/FT2
FTHER = 0LB/KWT
LiBr/H2O ABSORPTION
R22/E181 ABSORPTION
VAPOR COMPRESSION
H2O/ZEOLITE ADSORPTION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEIGHT - LB/KW
RADIATOR TEMPERATURE - DEG F
PAGE 60.



TEVAP = 35 F
TSINK = 100 F
FEPP = 300 LB/KW
FRAP = .1 LB/FT2
FTHER = 0 LB/KWT
R22/E181 ABSORPTION
VAPOR COMPRESSION
LiBr/H2O ABSORPTION
H2O/ZEOLITE ADSORPTION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WE. T - LB/KW
RADIATOR TEMPERATURE - DEG F
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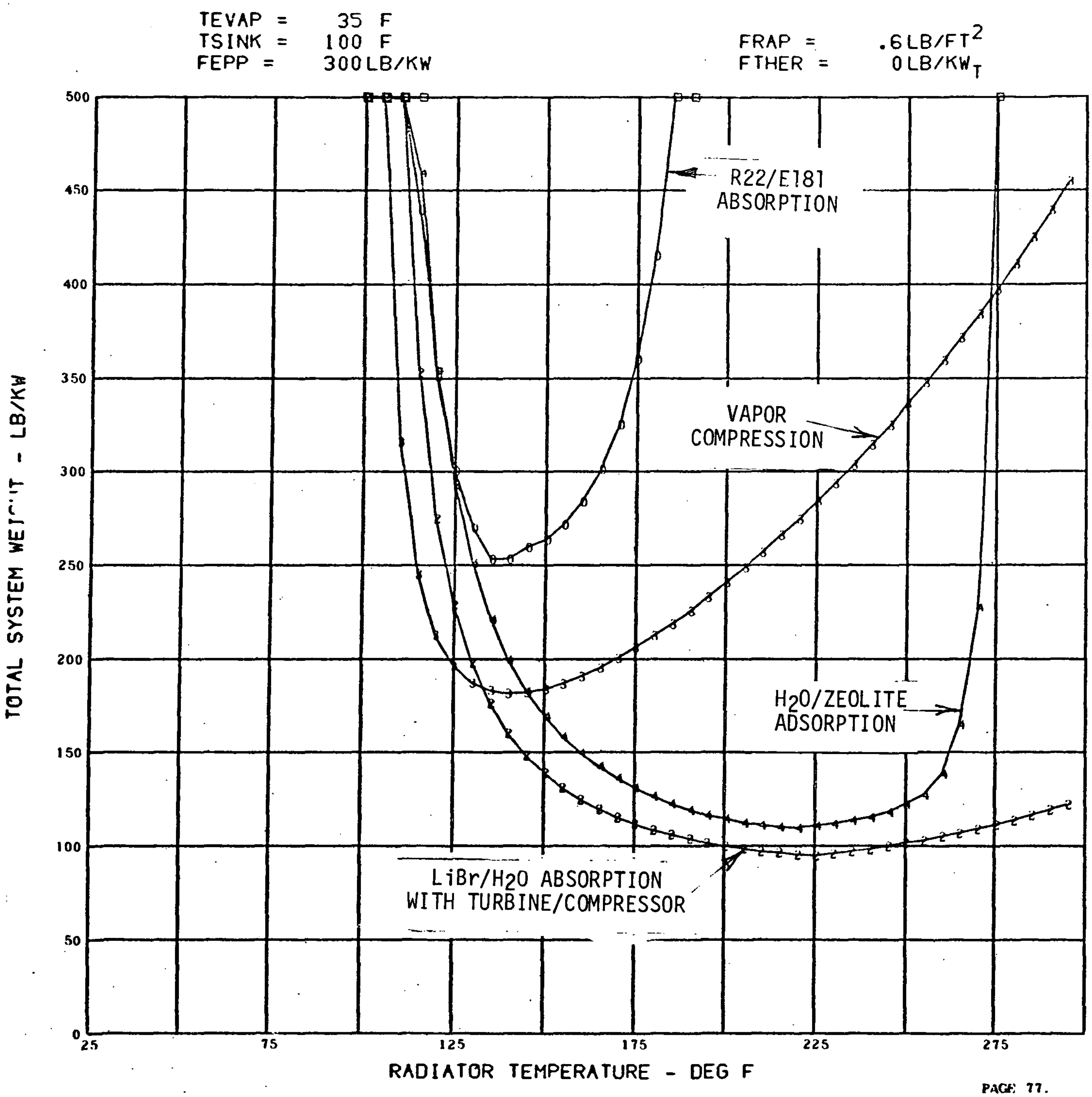
200
250
300
350
400
450
500
25
75
125
175
225
275
PAGE 76.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TSINK = 100 F
FEPP = 300 LB/KW
FRAP = .6 LB/FT2
FTHER = 0 LB/KWT
R22/E181 ABSORPTION
VAPOR COMPRESSION
H2O/ZEOLITE ADSORPTION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEIGHT - LB/KW
RADIATOR TEMPERATURE - DEG F
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200
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300
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400
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500
25
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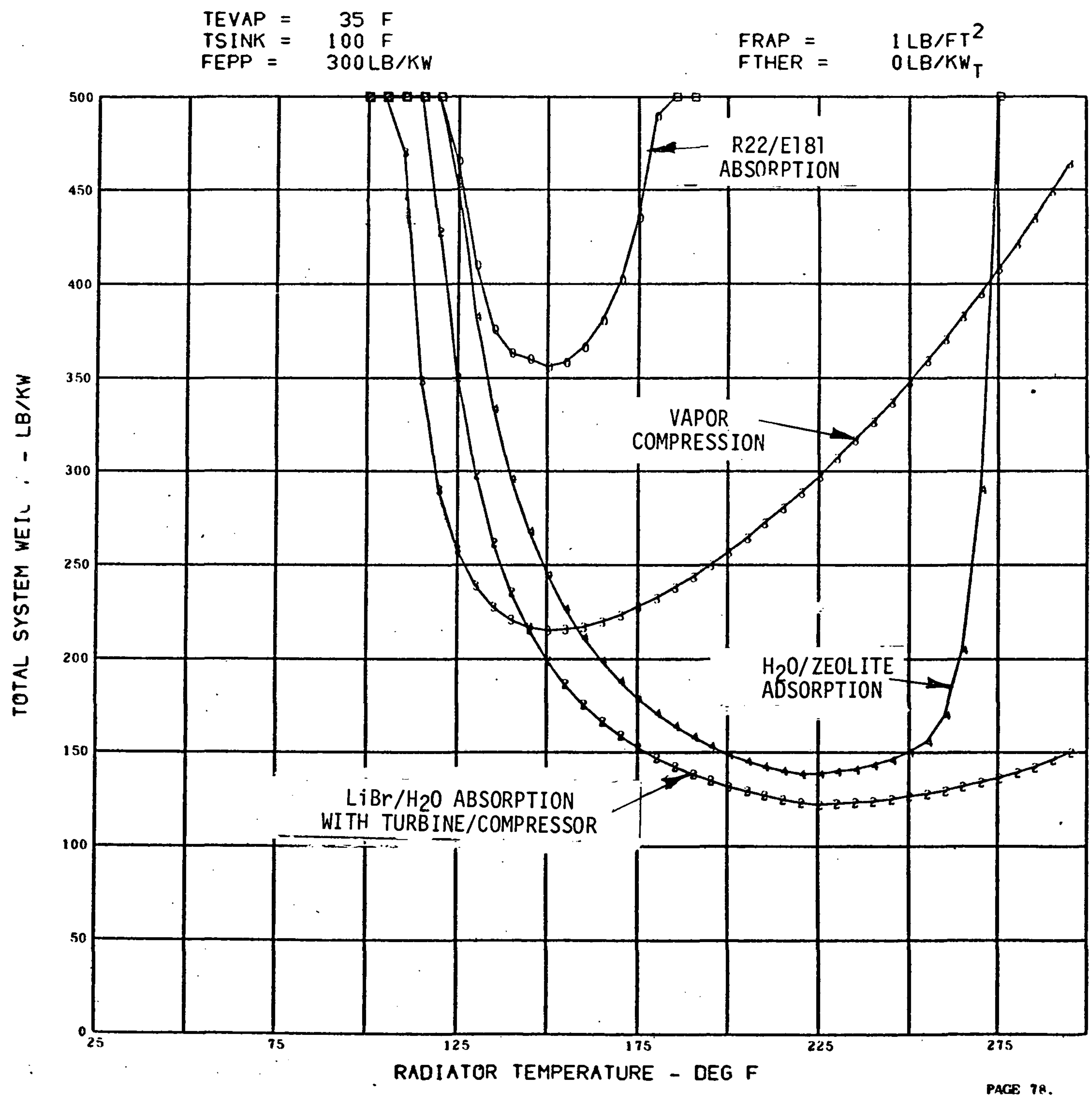
125
175
225
275
PAGE 77.



TEVAP = 35 F
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FRAP = 1LB/FT2
FTHER = 0LB/KWT
R22/E181 ABSORPTION
VAPOR COMPRESSION
H2O/ZEOLITE ADSORPTION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEI. . - LB/KW
RADIATOR TEMPERATURE - DEG F
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200
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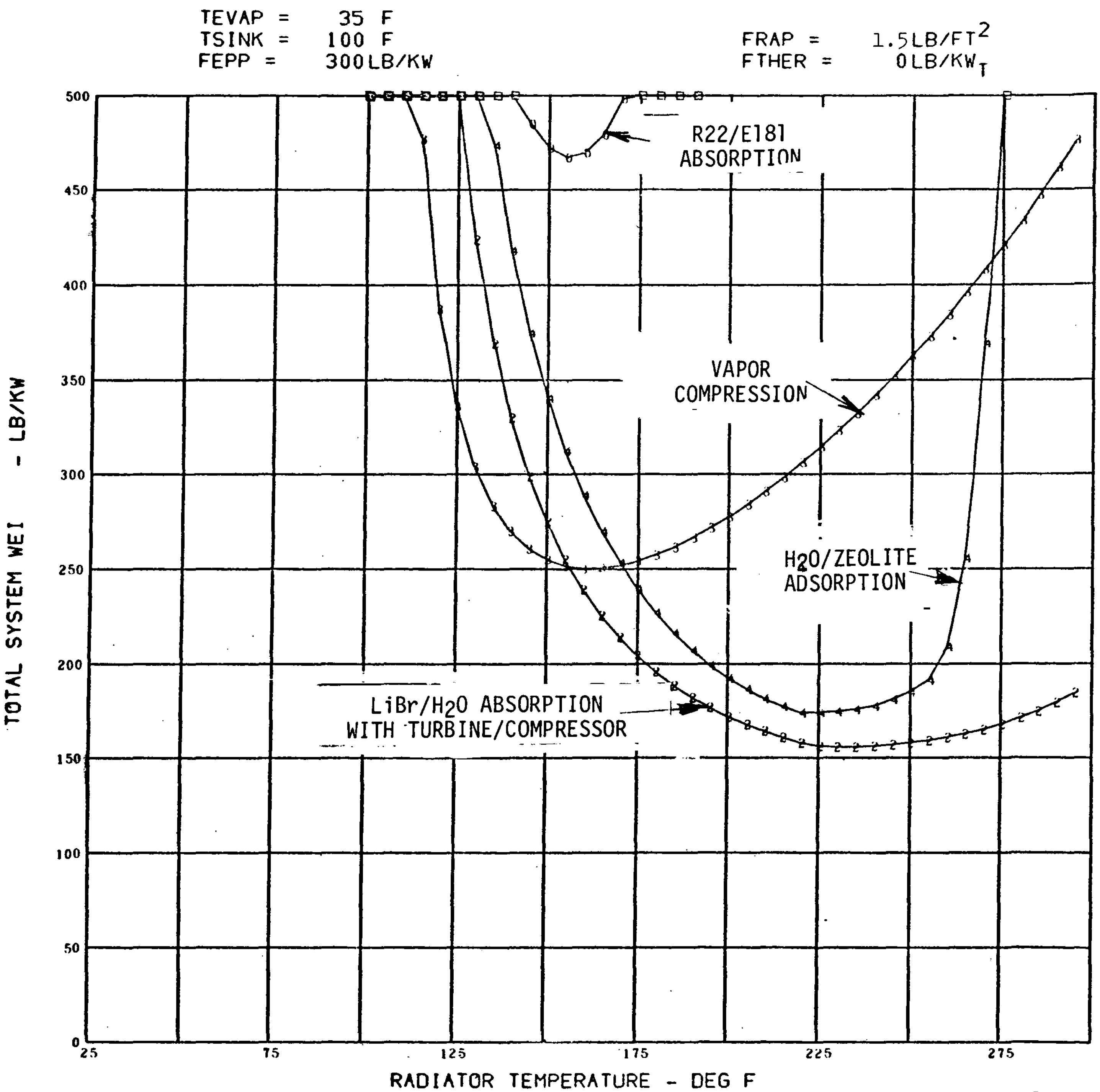
PAGE 78.



TEVAP = 35 F
TSINK = 100 F
FEPP = 300 LB/KW
FRAP = 1.5 LB/FT2
FTHER = 0 LB/KWT
R22/E181 ABSORPTION
VAPOR COMPRESSION
H2O/ZEOLITE ADSORPTION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEI - LB/KW
RADIATOR TEMPERATURE - DEG F
500
450
400
350
300
250
200
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100
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0
25
75
125
175
225
275

PAGE 79.

TEVAP = 35 F
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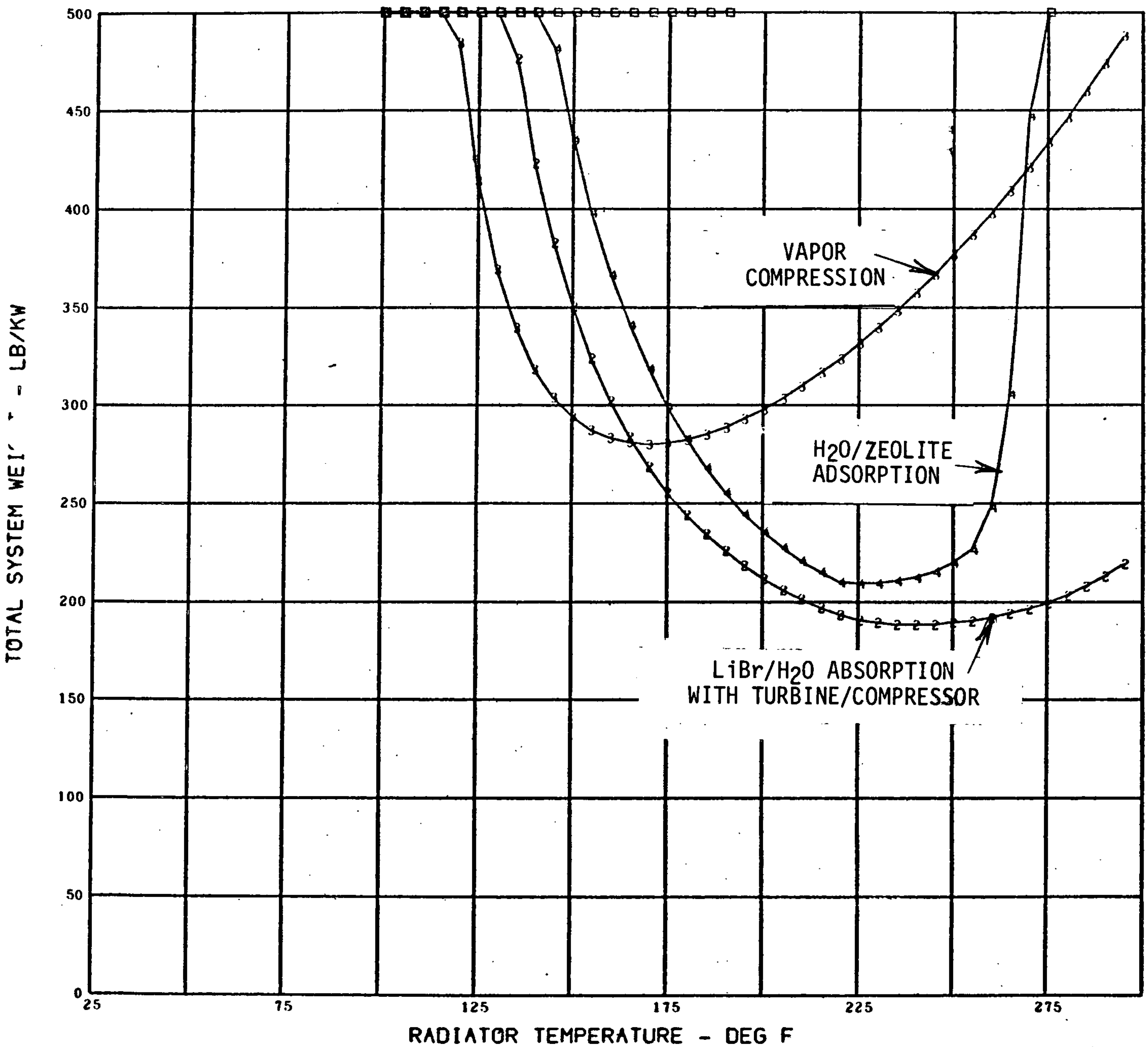
FEPP = 300 LB/KW

FRAP = 2 LB/FT^2
FTHER = 0 LB/KW$_T$



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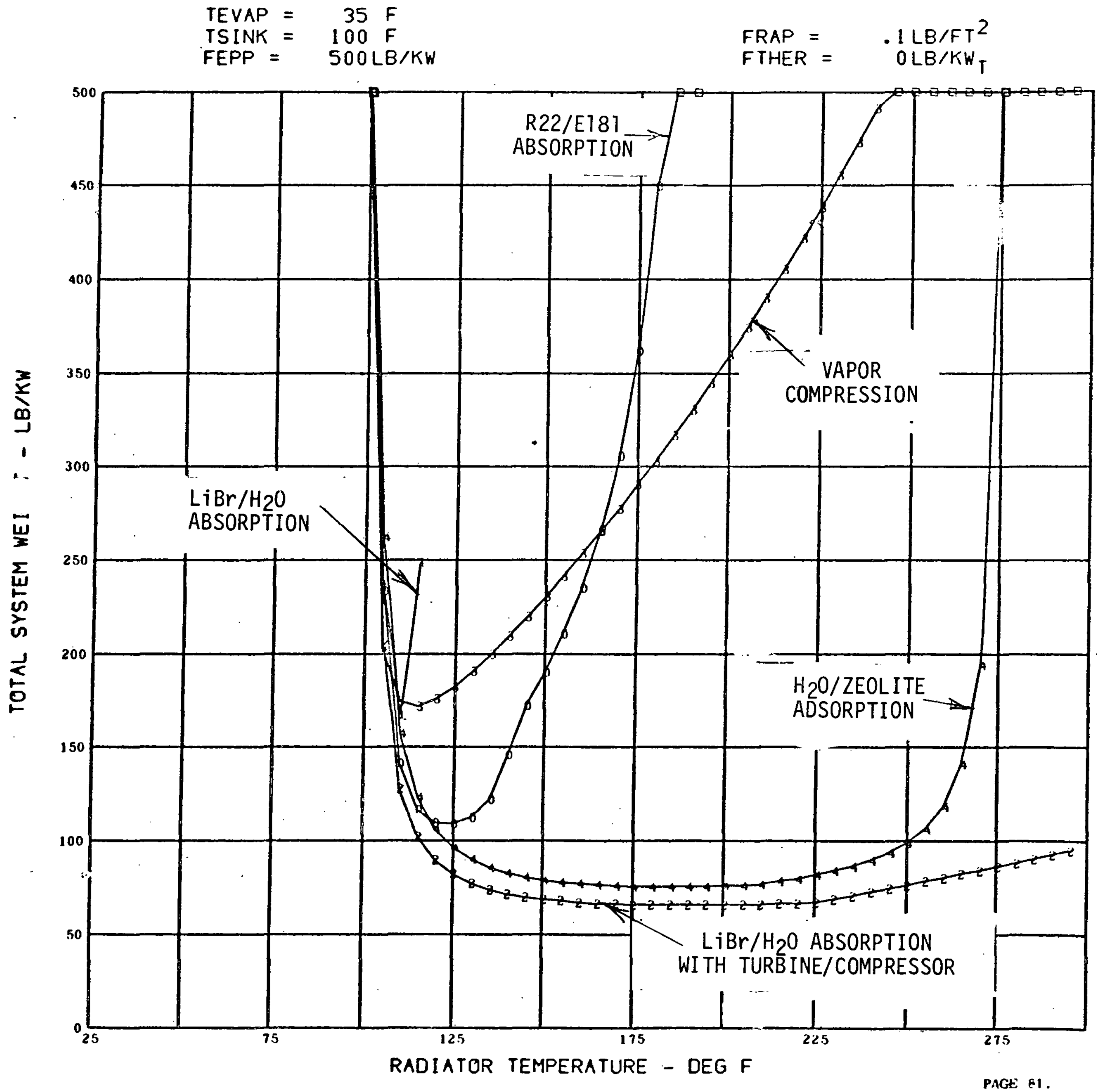

TEVAP = 35 F
TSINK = 100 F
FEPP = 500LB/KW
FRAP = .1LB/FT2
FTHER = 0LB/KWT
R22/E181 ABSORPTION
VAPOR COMPRESSION
LiBr/H2O ABSORPTION
H2O/ZEOLITE ADSORPTION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEI - LB/KW
RADIATOR TEMPERATURE - DEG F
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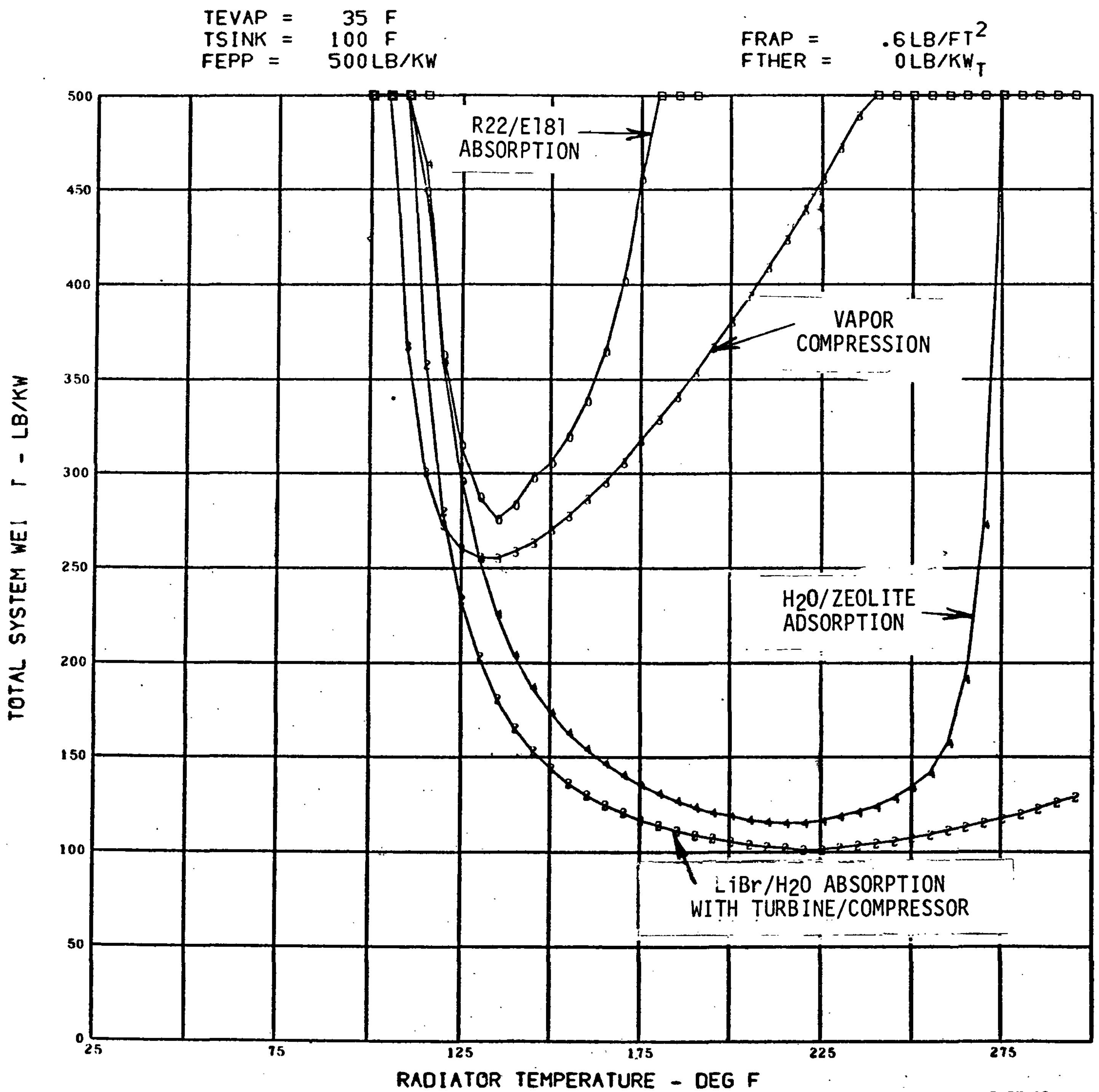
PAGE 61.



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TSINK = 100 F
FEPP = 500LB/KW
FRAP = .6LB/FT2
FTHER = 0LB/KWT
R22/E181 ABSORPTION
VAPOR COMPRESSION
H2O/ZEOLITE ADSORPTION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEI   T - LB/KW
RADIATOR TEMPERATURE - DEG F
0
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125
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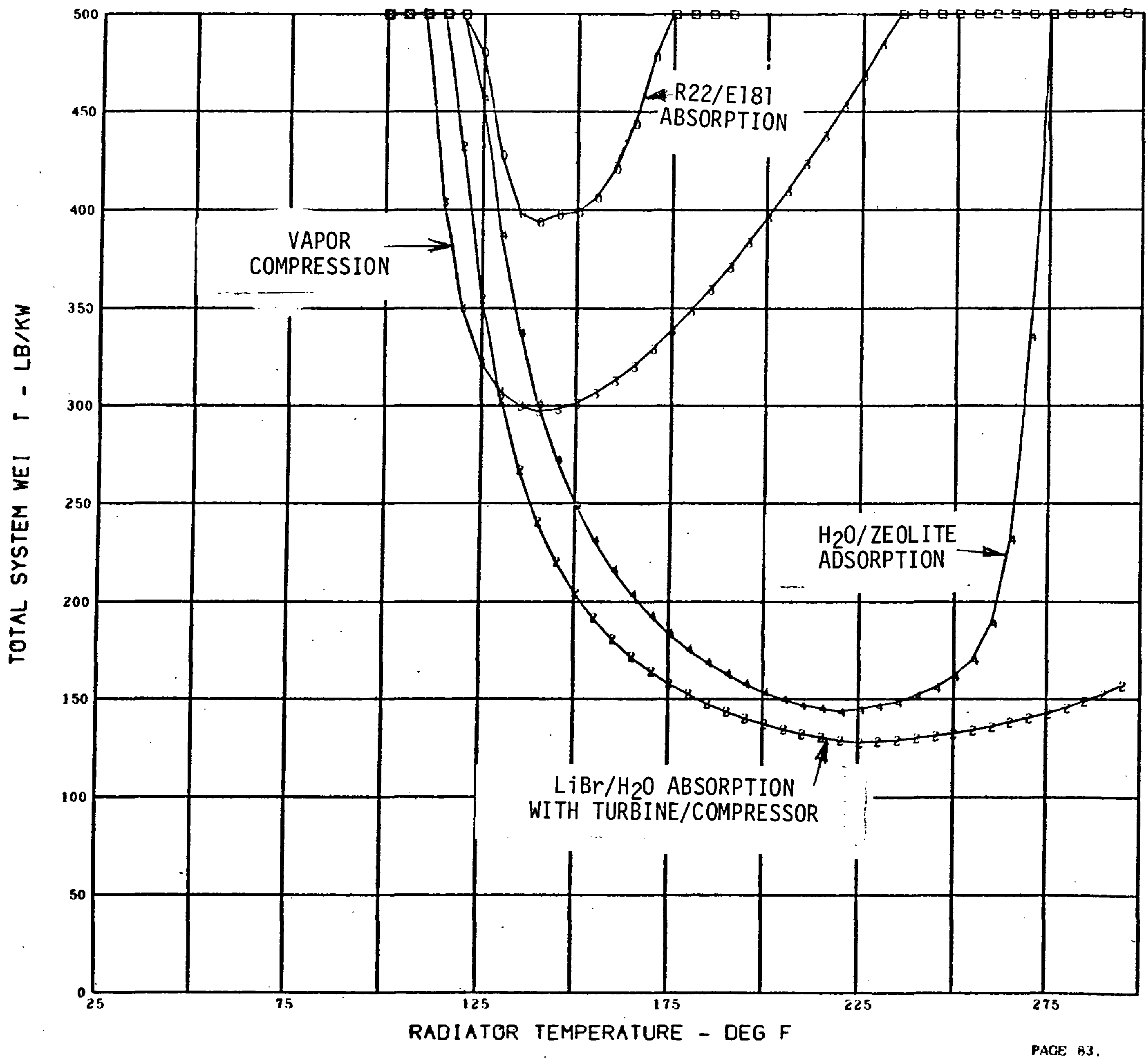
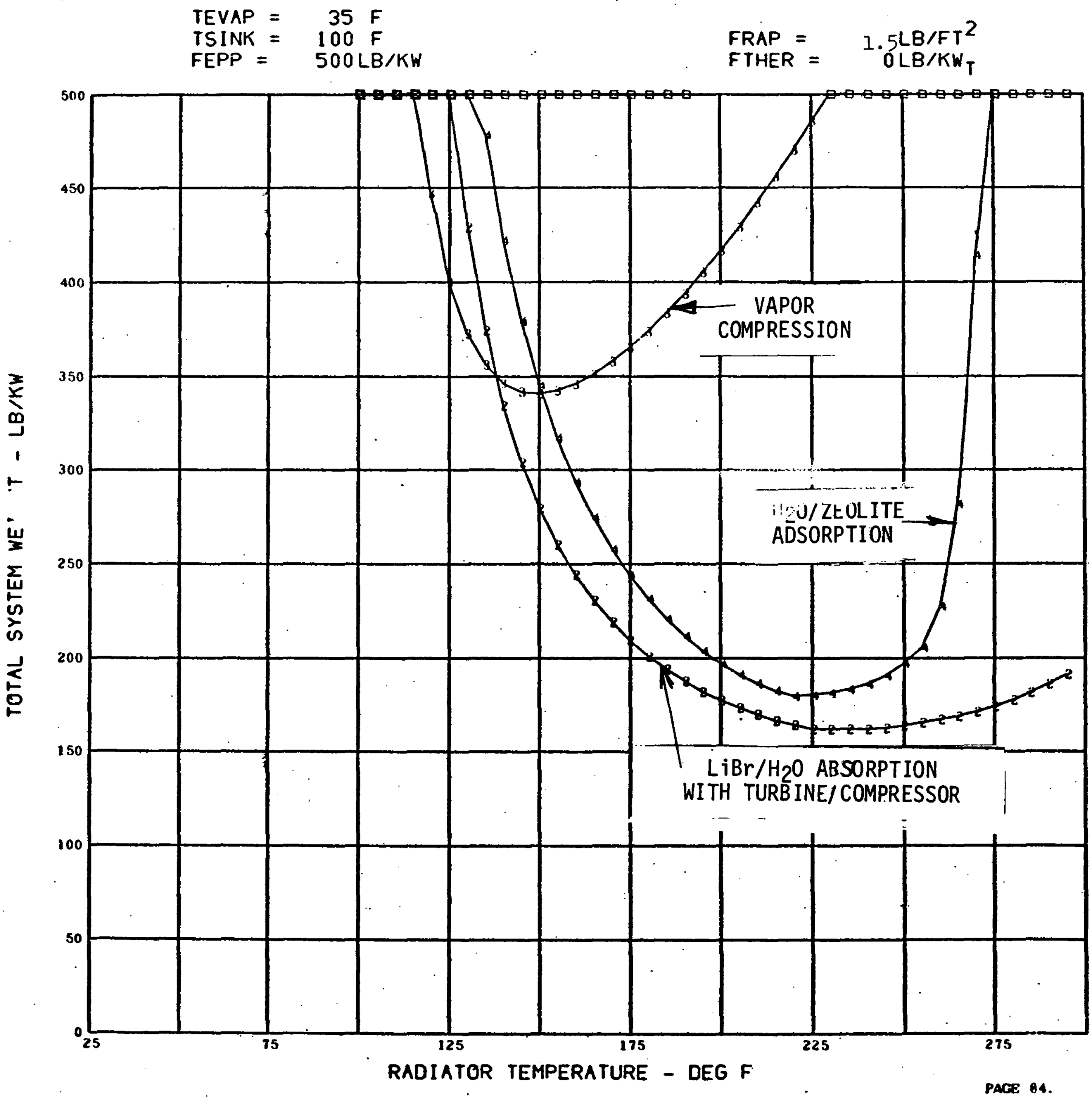
275

PAGE 82.

TEVAP = 35 F
TSINK = 100 F
FEPP = 500 LB/KW

FRAP = 1 LB/FT^2
FTHER = 0 LB/KW$_T$





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TSINK = 100 F
FEPP = 500 LB/KW
FRAP = 1.5 LB/FT2
FTHER = 0 LB/KWT
VAPOR COMPRESSION
H2O/ZEOLITE ADSORPTION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WE' 'T - LB/KW
RADIATOR TEMPERATURE - DEG F
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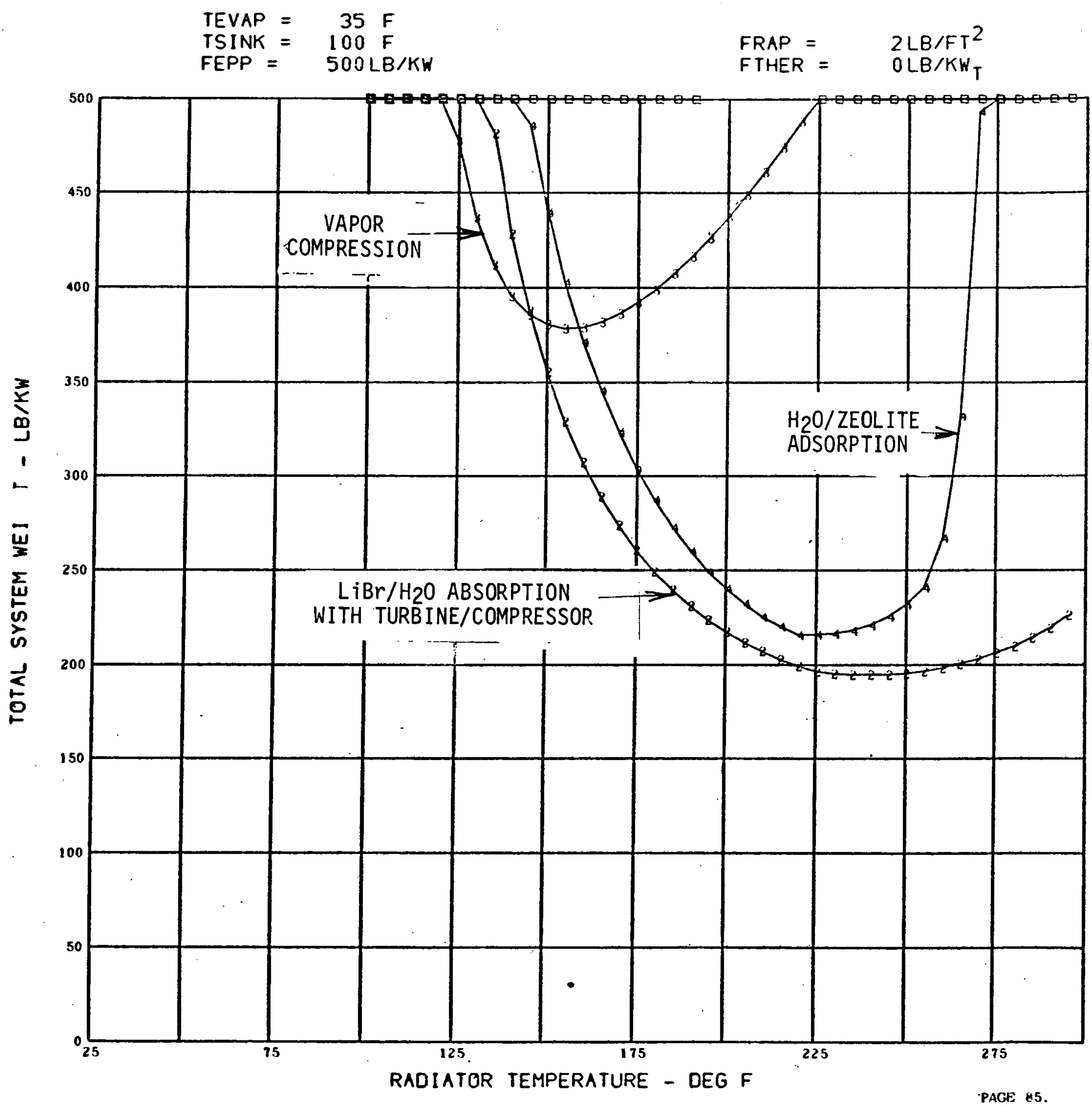
PAGE 84.


TEVAP = 35 F
TSINK = 100 F
FEPP = 500LB/KW
FRAP = 2LB/FT2
FTHER = 0LB/KWT
VAPOR COMPRESSION
H2O/ZEOLITE ADSORPTION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEI T - LB/KW
RADIATOR TEMPERATURE - DEG F
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225
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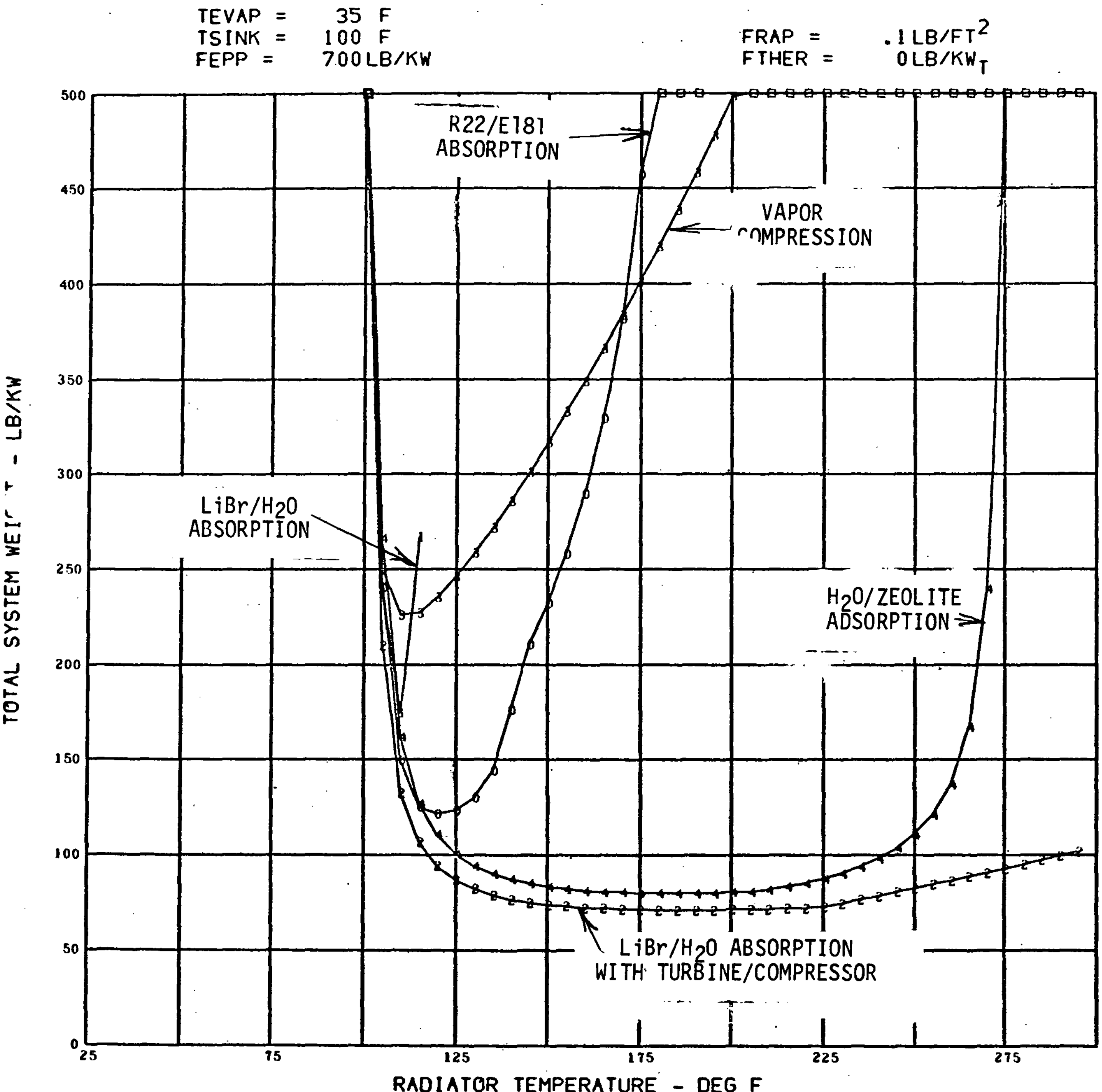
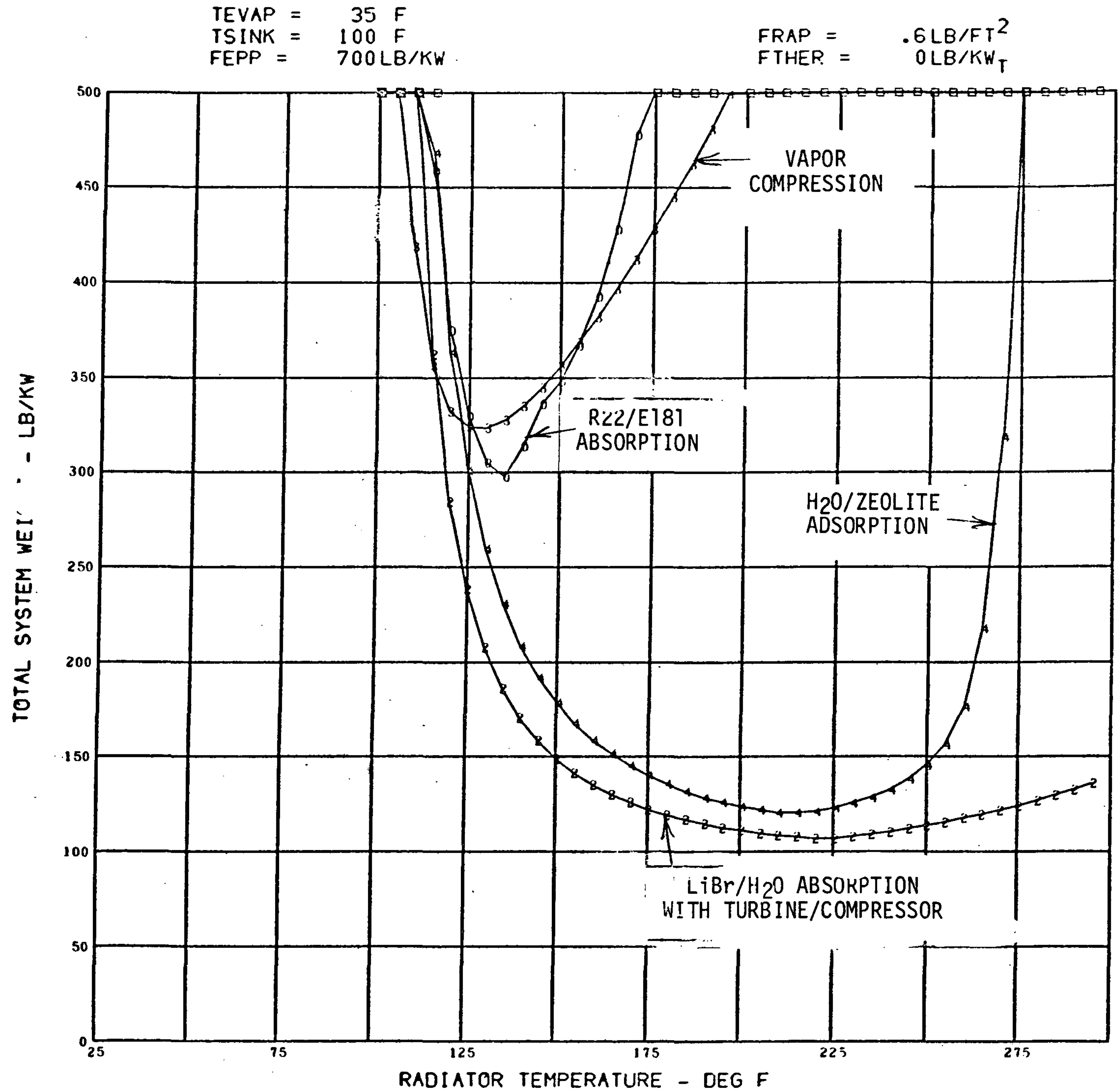
PAGE 85.

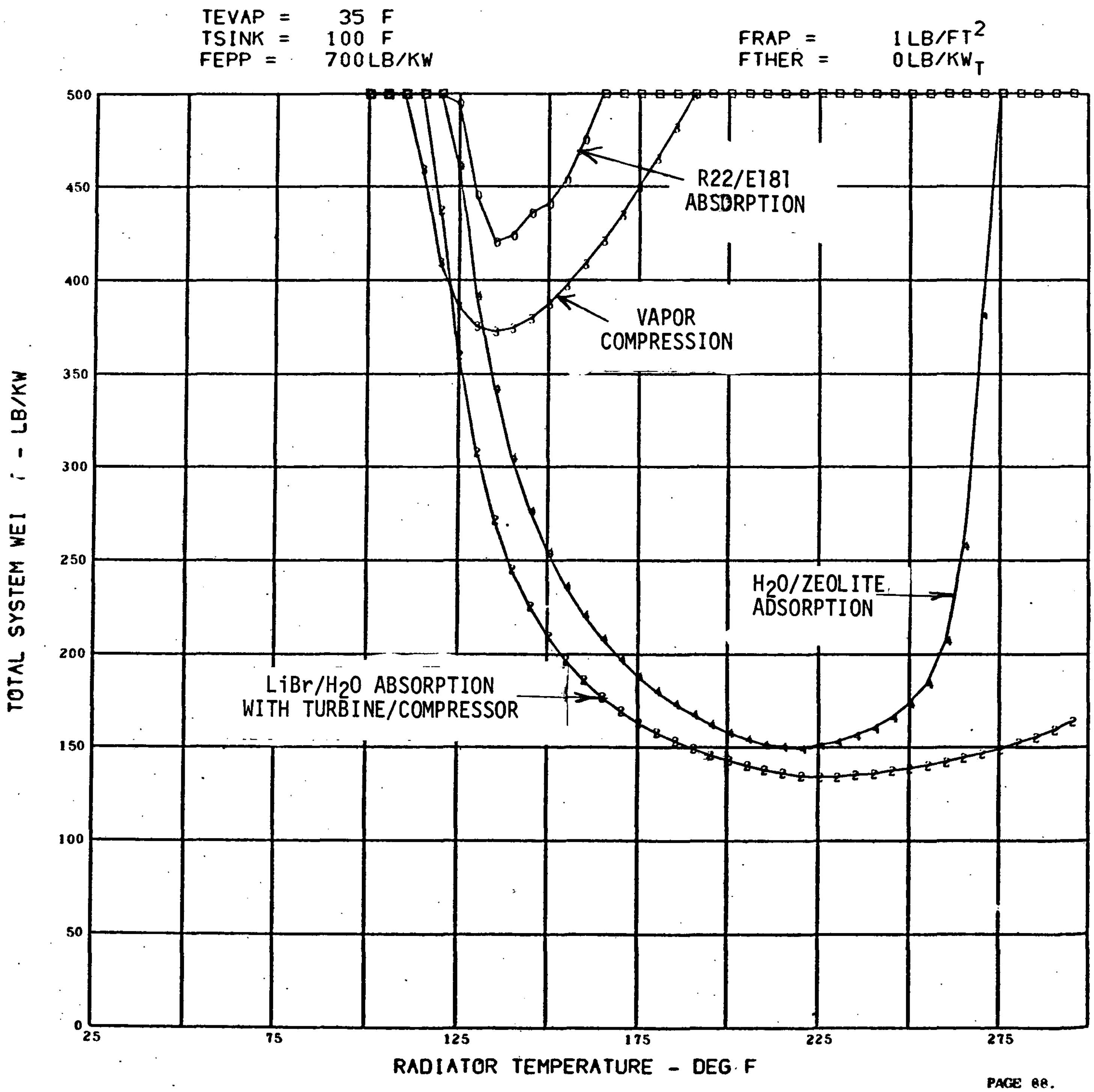

TEVAP = 35 F
TSINK = 100 F
FEPP = 700 LB/KW
FRAP = .1 LB/FT2
FTHER = 0 LB/KWT
R22/E181 ABSORPTION
VAPOR COMPRESSION
LiBr/H2O ABSORPTION
H2O/ZEOLITE ADSORPTION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEIGHT - LB/KW
RADIATOR TEMPERATURE - DEG F
0
50
100
150
200
250
300
350
400
450
500
25
75
125
175
225
275

PAGE 86.



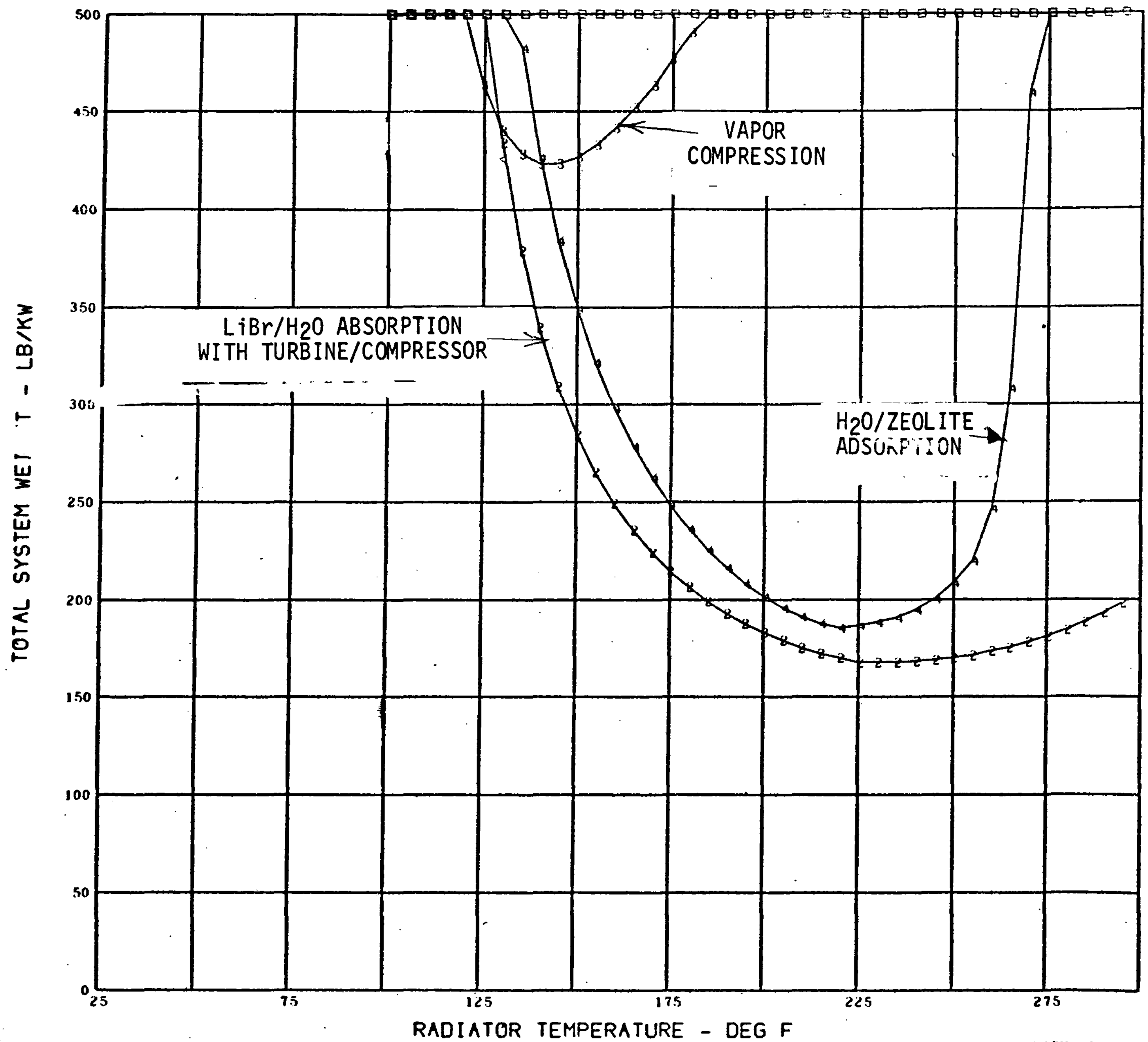
TEVAP = 35 F
TSINK = 100 F
FEPP = 700LB/KW
FRAP = .6LB/FT2
FTHER = 0LB/KWT
VAPOR COMPRESSION
R22/E181 ABSORPTION
H2O/ZEOLITE ADSORPTION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEI' - LB/KW
RADIATOR TEMPERATURE - DEG F
0
50
100
150
200
250
300
350
400
450
500
25
75
125
175
225
275

PAGE 87.


TEVAP = 35 F
TSINK = 100 F
FEPP = 700 LB/KW
FRAP = 1 LB/FT2
FTHER = 0 LB/KWT
R22/E181 ABSORPTION
VAPOR COMPRESSION
H2O/ZEOLITE ADSORPTION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEI - LB/KW
RADIATOR TEMPERATURE - DEG F
500
450
400
350
300
250
200
150
100
50
0
25
75
125
175
225
275
PAGE 88.

TEVAP = 35 F
TSINK = 100 F
FEPP = 700 LB/KW

FRAP = 1.5 LB/FT^2
FTHER = 0 LB/KW_T



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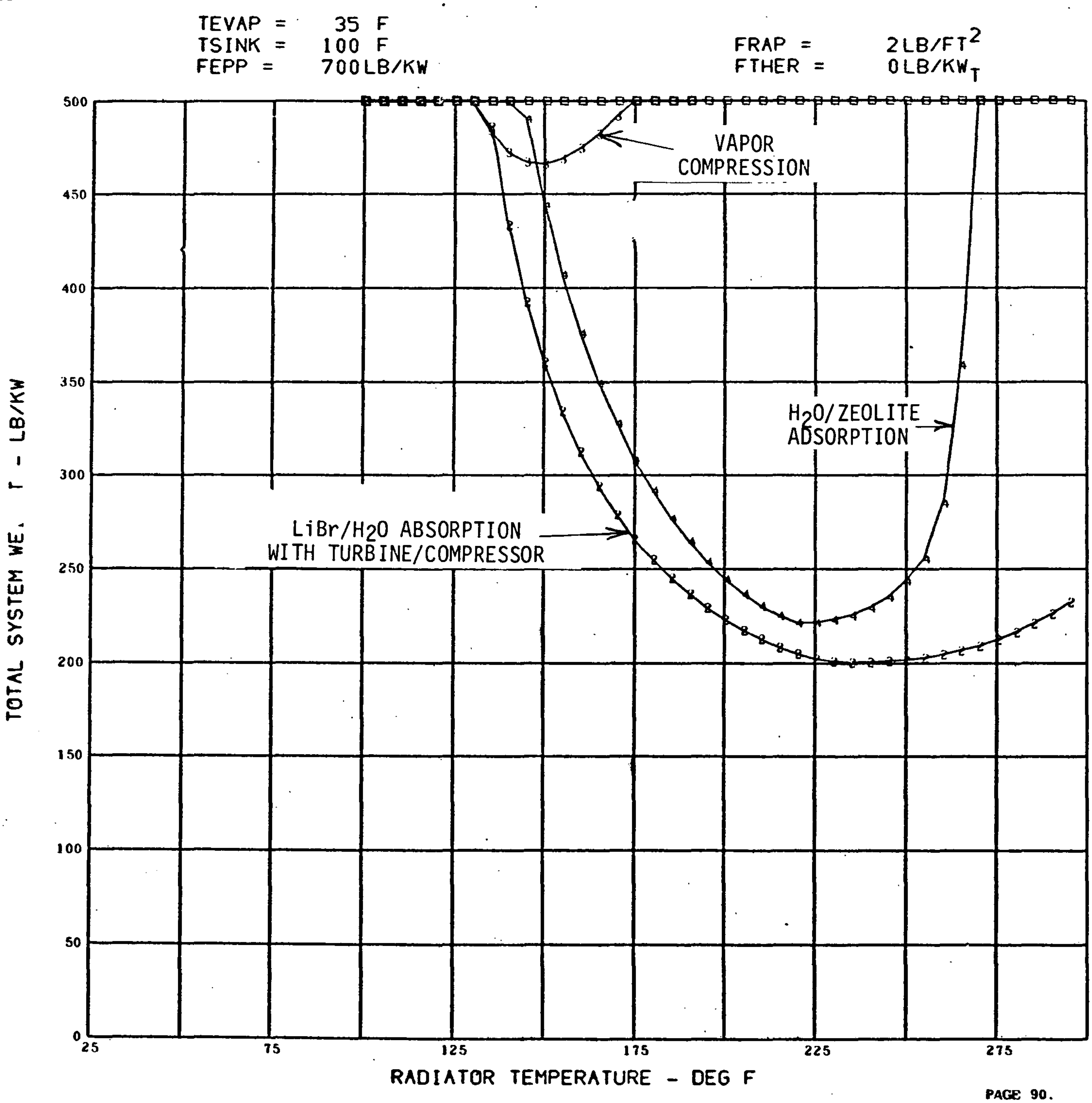

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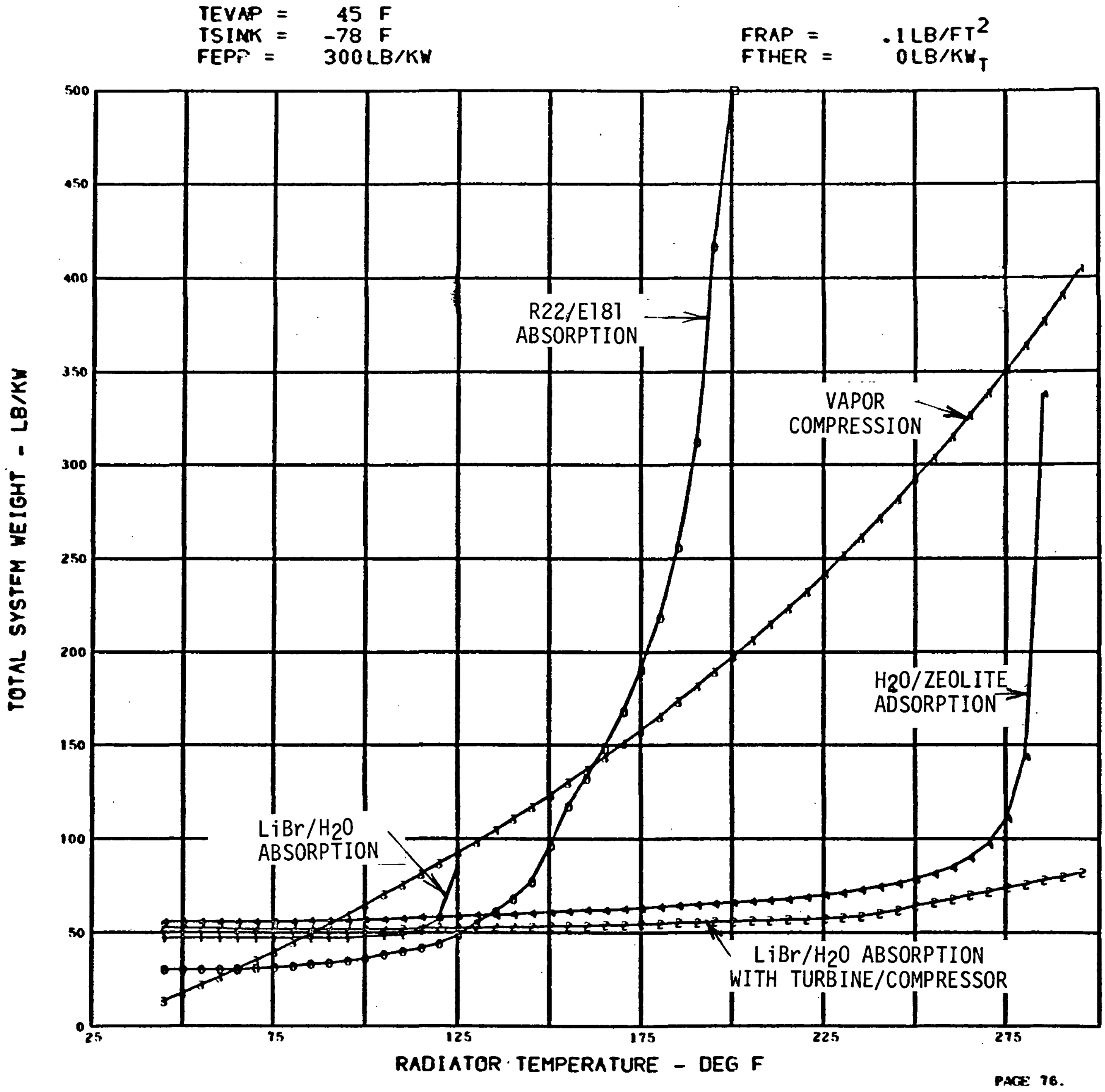
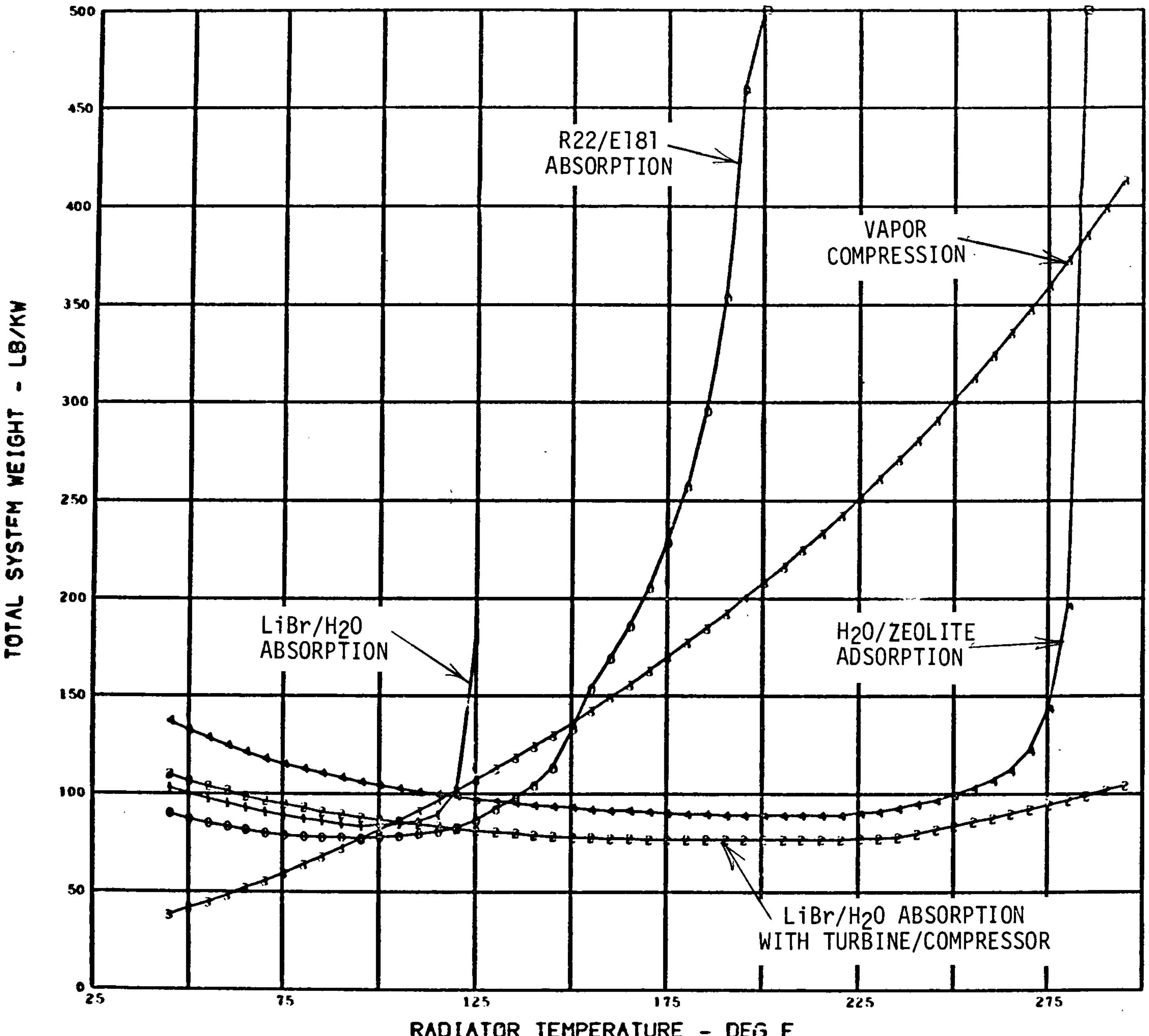
FEPP = 700 LB/KW
FRAP = 2 LB/FT2
FTHER = 0 LB/KWT
VAPOR COMPRESSION
H2O/ZEOLITE ADSORPTION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WE. T - LB/KW
RADIATOR TEMPERATURE - DEG F
500
450
400
350
300
250
200
150
100
50
0
25
75
125
175
225
275
PAGE 90.



TEVAP = 45 F
TSINK = -78 F
FEPP = 300LB/KW
FRAP = .1LB/FT2
FTHER = 0LB/KWT
TOTAL SYSTEM WEIGHT - LB/KW
RADIATOR TEMPERATURE - DEG F
R22/E181
ABSORPTION
VAPOR
COMPRESSION
H2O/ZEOLITE
ADSORPTION
LiBr/H2O
ABSORPTION
LiBr/H2O ABSORPTION
WITH TURBINE/COMPRESSOR
0
50
100
150
200
250
300
350
400
450
500
25
75
125
175
225
275
PAGE 76.

TEVAP = 45 F
TSINK = -78 F
FEPP = 300 LB/KW

FRAP = .6 LB/FT^2
FTHER = 0 LB/KW$_T$



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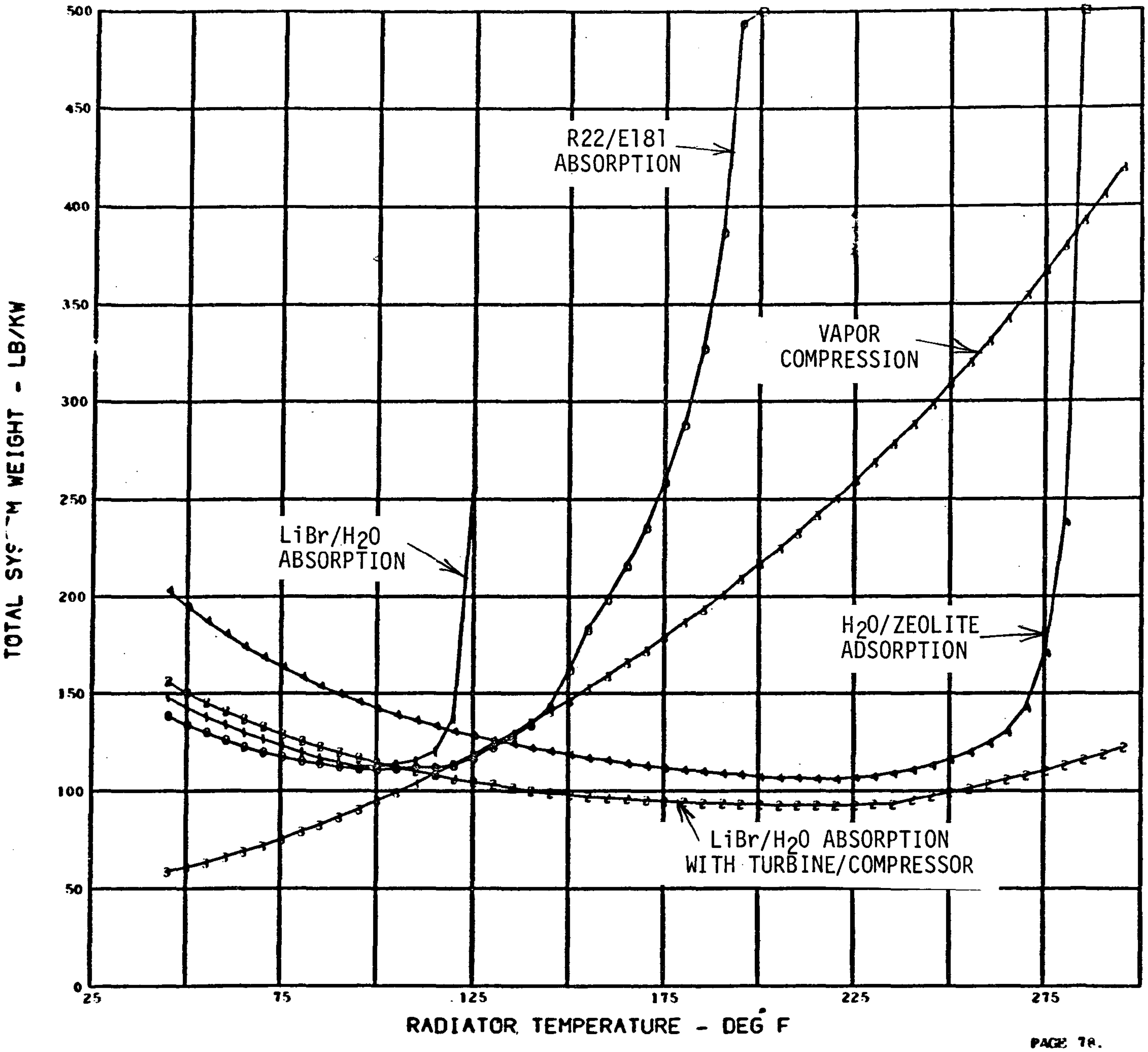
TEVAP = 45 F
TSINK = -78 F
FEPP = 300 LB/KW

FRAP = 1 LB/FT^2
FTHER = 0 LB/KW$_T$



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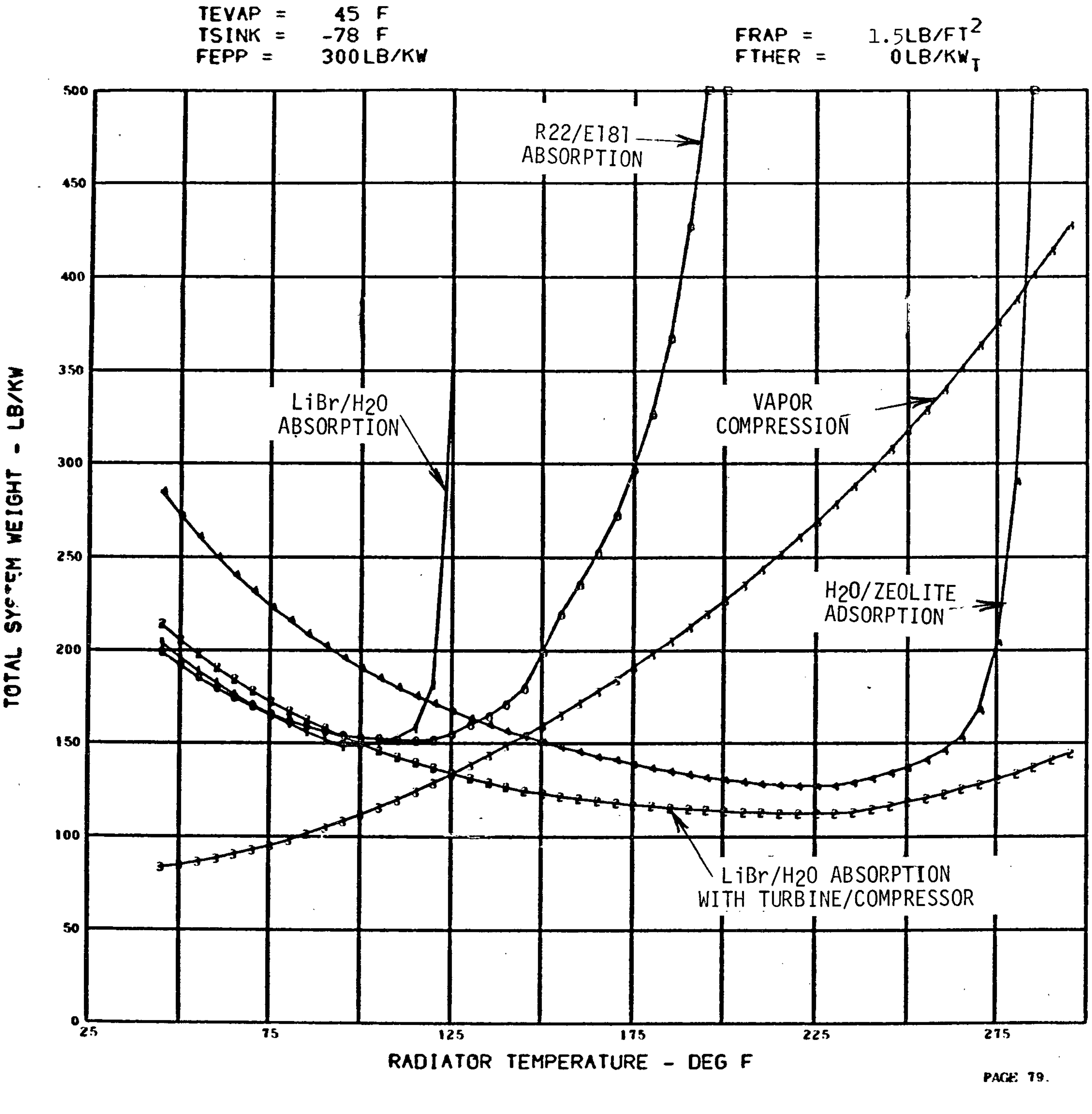
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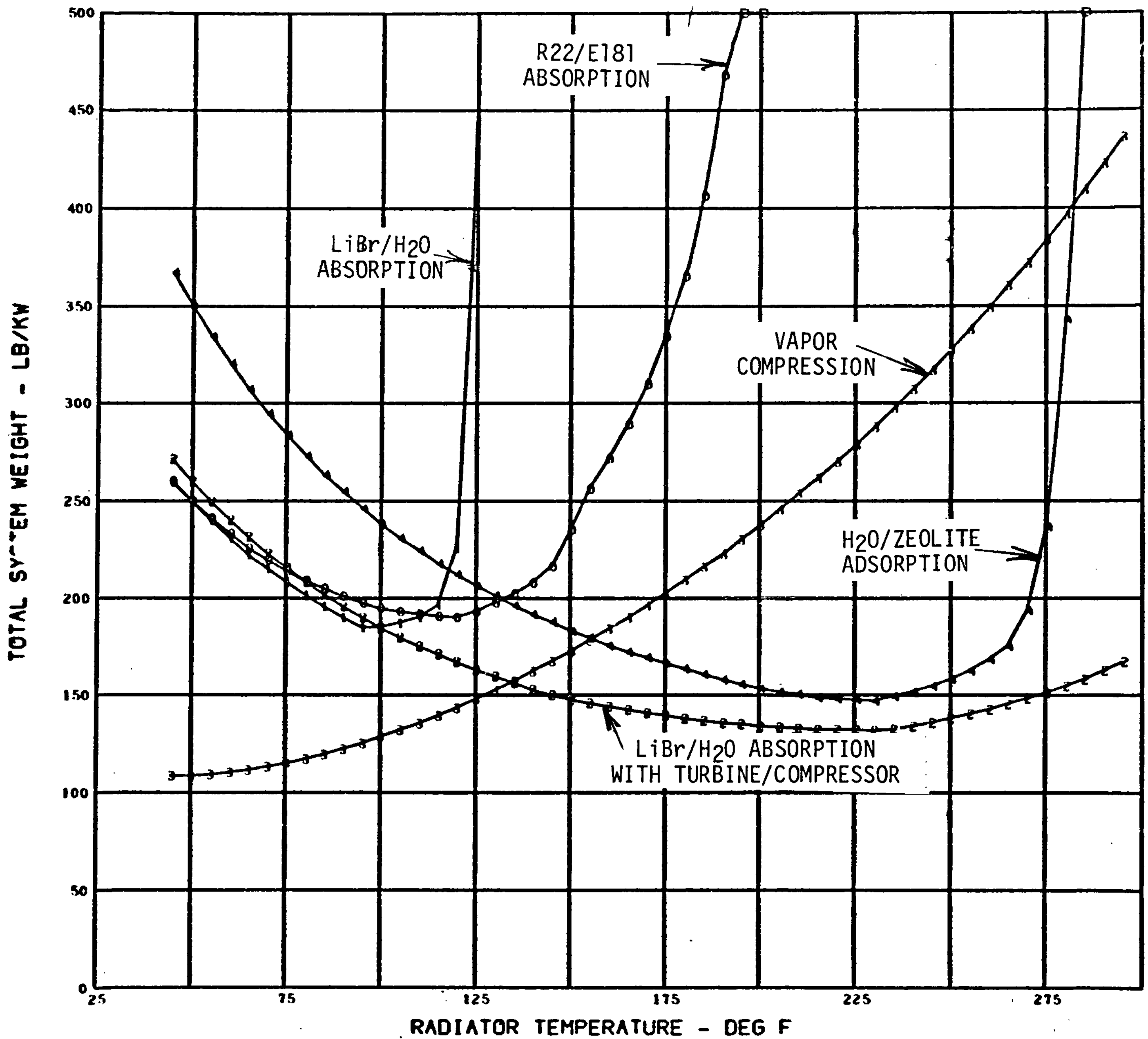
FEPP = 300LB/KW
FRAP = 1.5LB/FT2
FTHER = 0LB/KWT
R22/E181 ABSORPTION
LiBr/H2O ABSORPTION
VAPOR COMPRESSION
H2O/ZEOLITE ADSORPTION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEIGHT - LB/KW
RADIATOR TEMPERATURE - DEG F
0
50
100
150
200
250
300
350
400
450
500
25
75
125
175
225
275
PAGE 79.

TEVAP = 45 F
TSINK = -78 F
FEPP = 300 LB/KW

FRAP = 2 LB/FT^2
FTHER = 0 LB/KW$_T$



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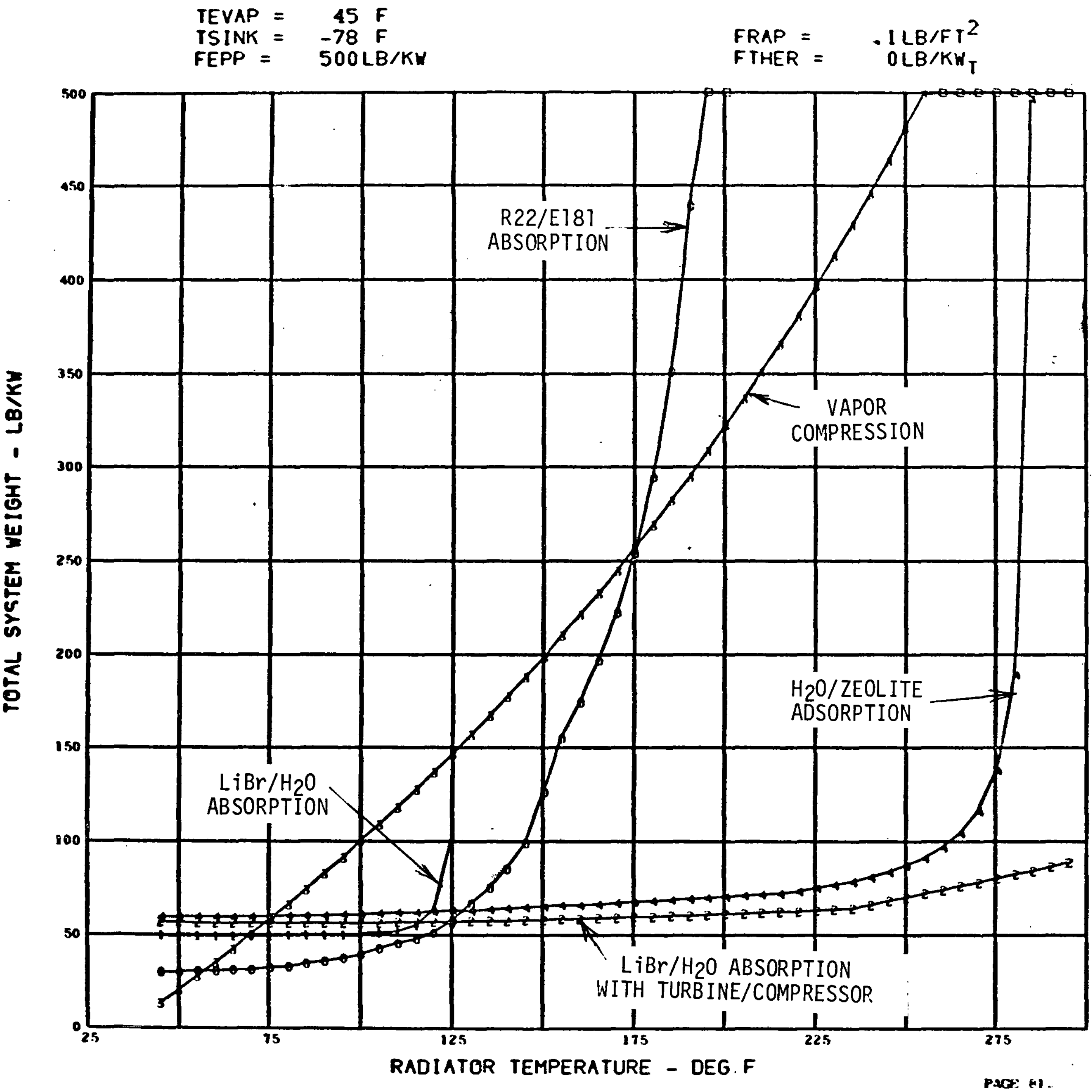
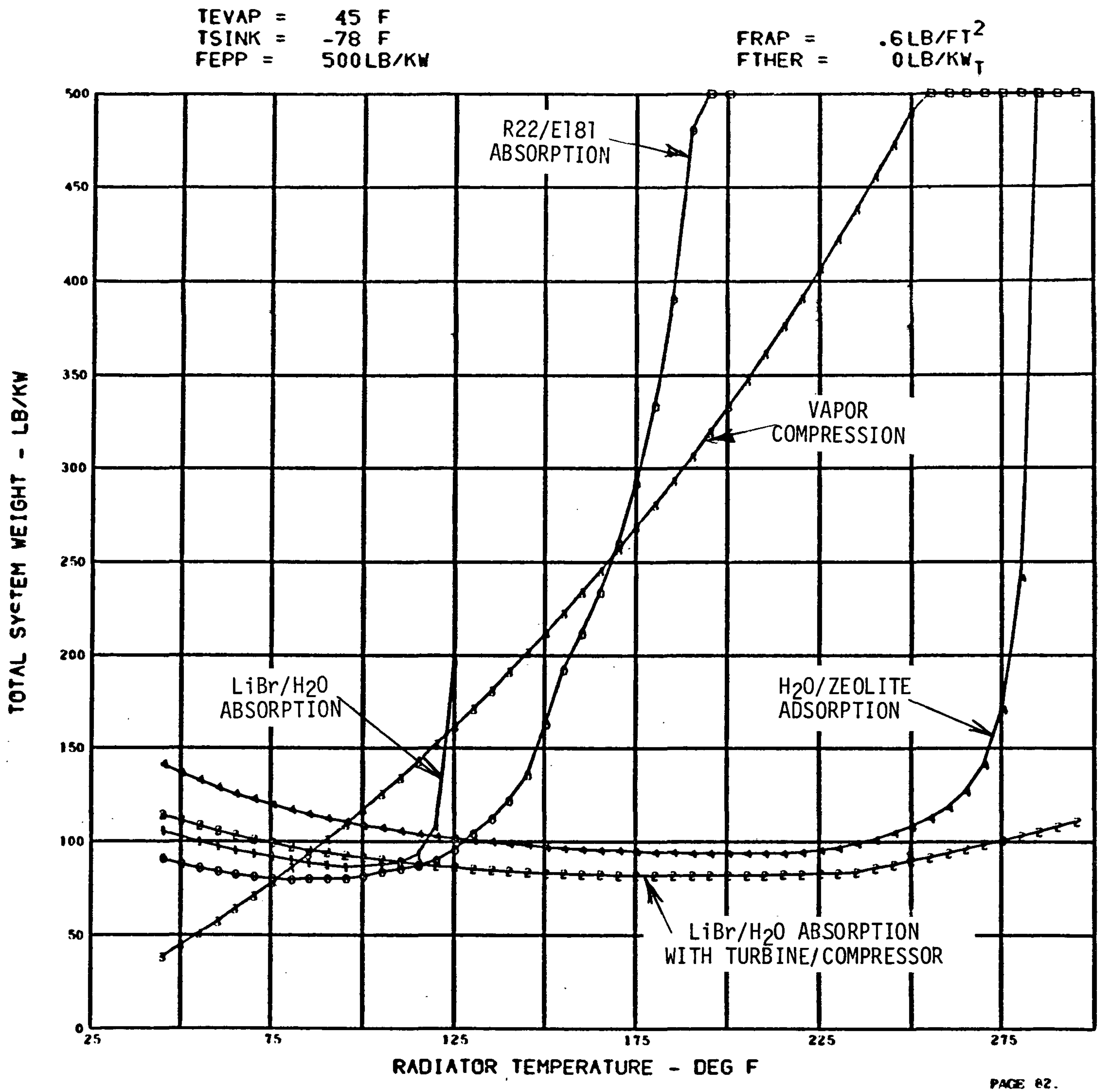


TEVAP = 45 F
TSINK = -78 F
FEPP = 500LB/KW
FRAP = .1LB/FT2
FTHER = 0LB/KWT
R22/E181 ABSORPTION
VAPOR COMPRESSION
H2O/ZEOLITE ADSORPTION
LiBr/H2O ABSORPTION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEIGHT - LB/KW
RADIATOR TEMPERATURE - DEG F
0
50
100
150
200
250
300
350
400
450
500
25
75
125
175
225
275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TSINK = -78 F
FEPP = 500LB/KW
FRAP = .6LB/FT2
FTHER = 0LB/KWT
R22/E181 ABSORPTION
VAPOR COMPRESSION
LiBr/H2O ABSORPTION
H2O/ZEOLITE ADSORPTION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEIGHT - LB/KW
RADIATOR TEMPERATURE - DEG F
0
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100
150
200
250
300
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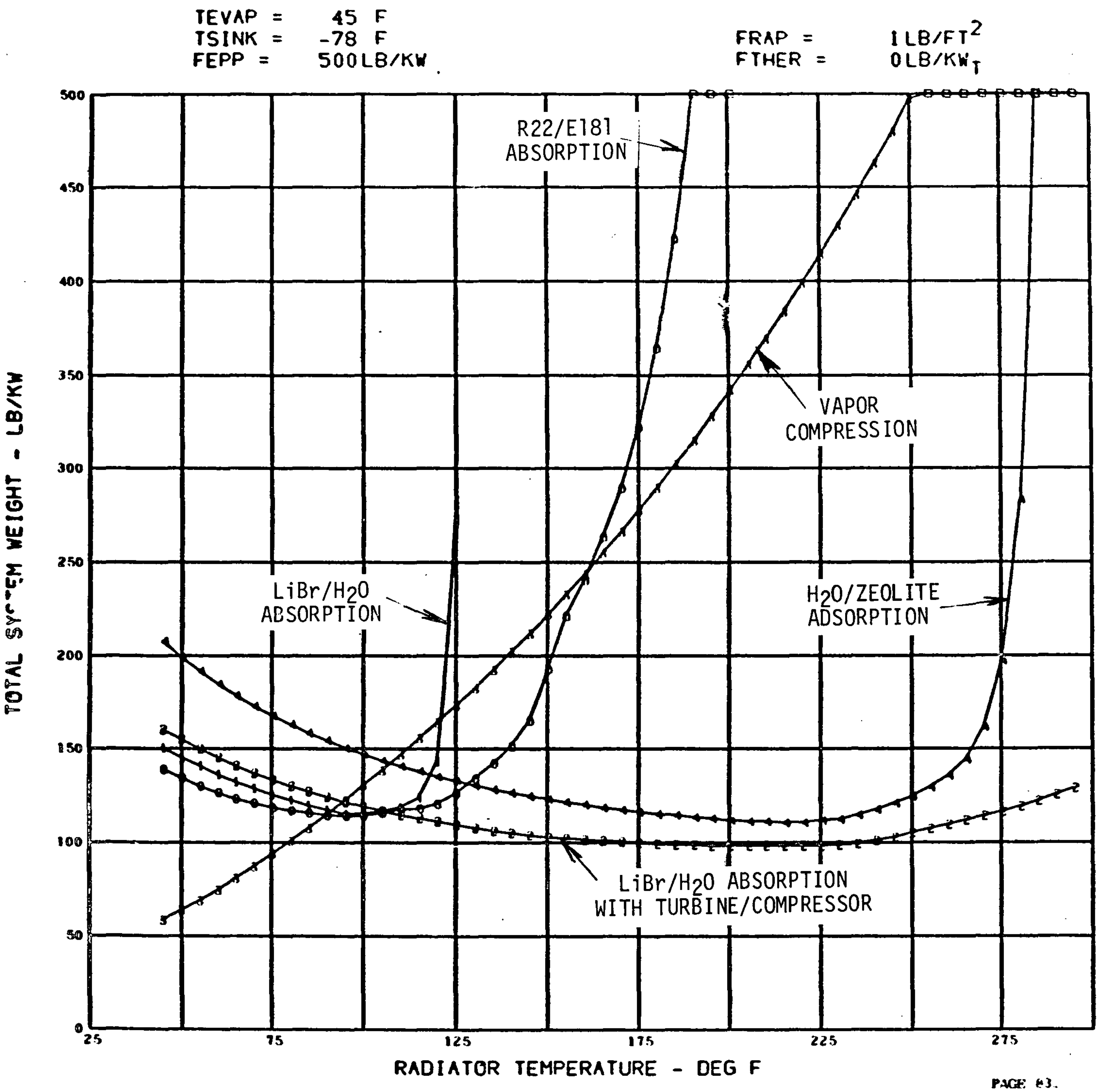
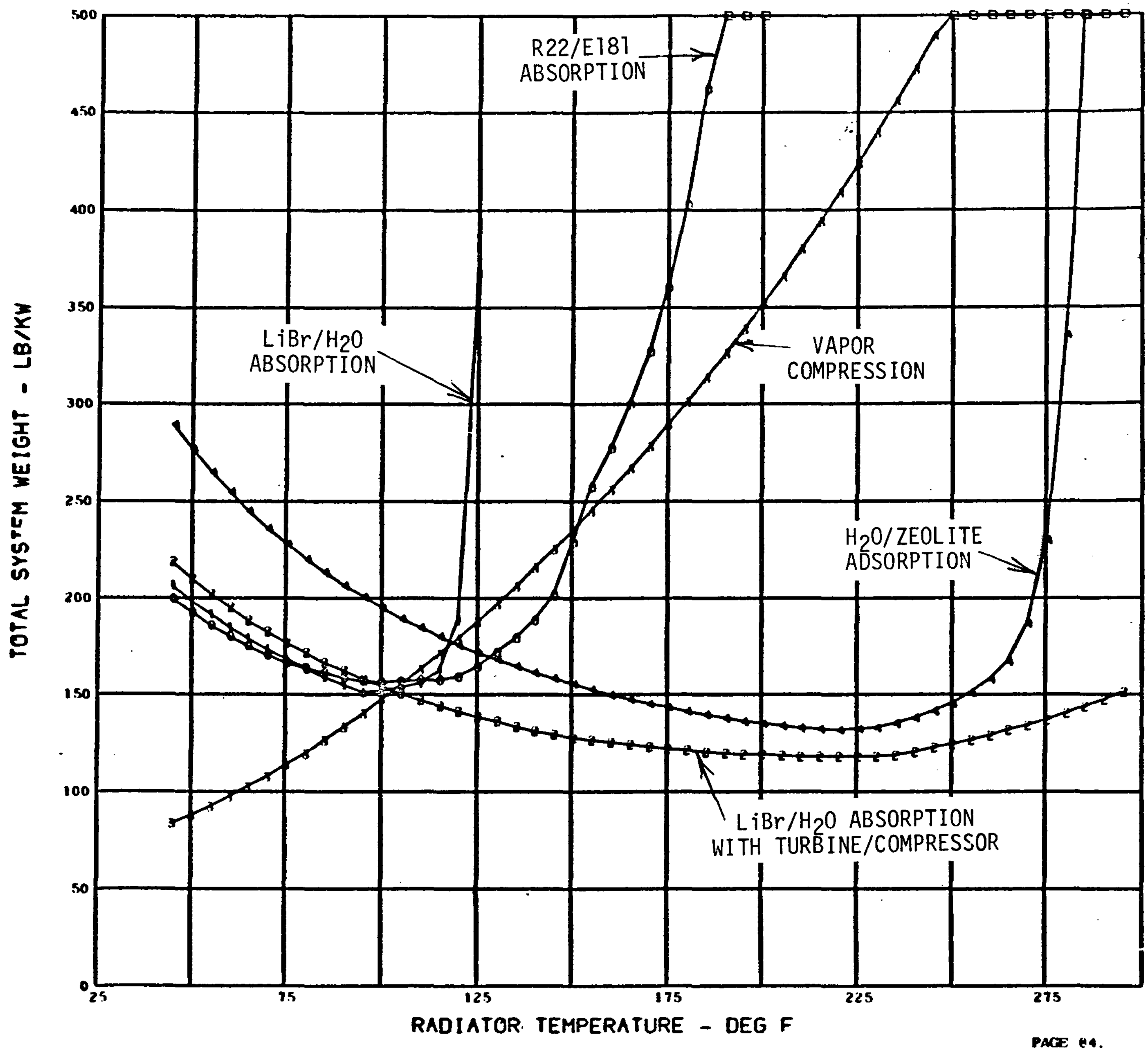
400
450
500
25
75
125
175
225
275
PAGE 82.

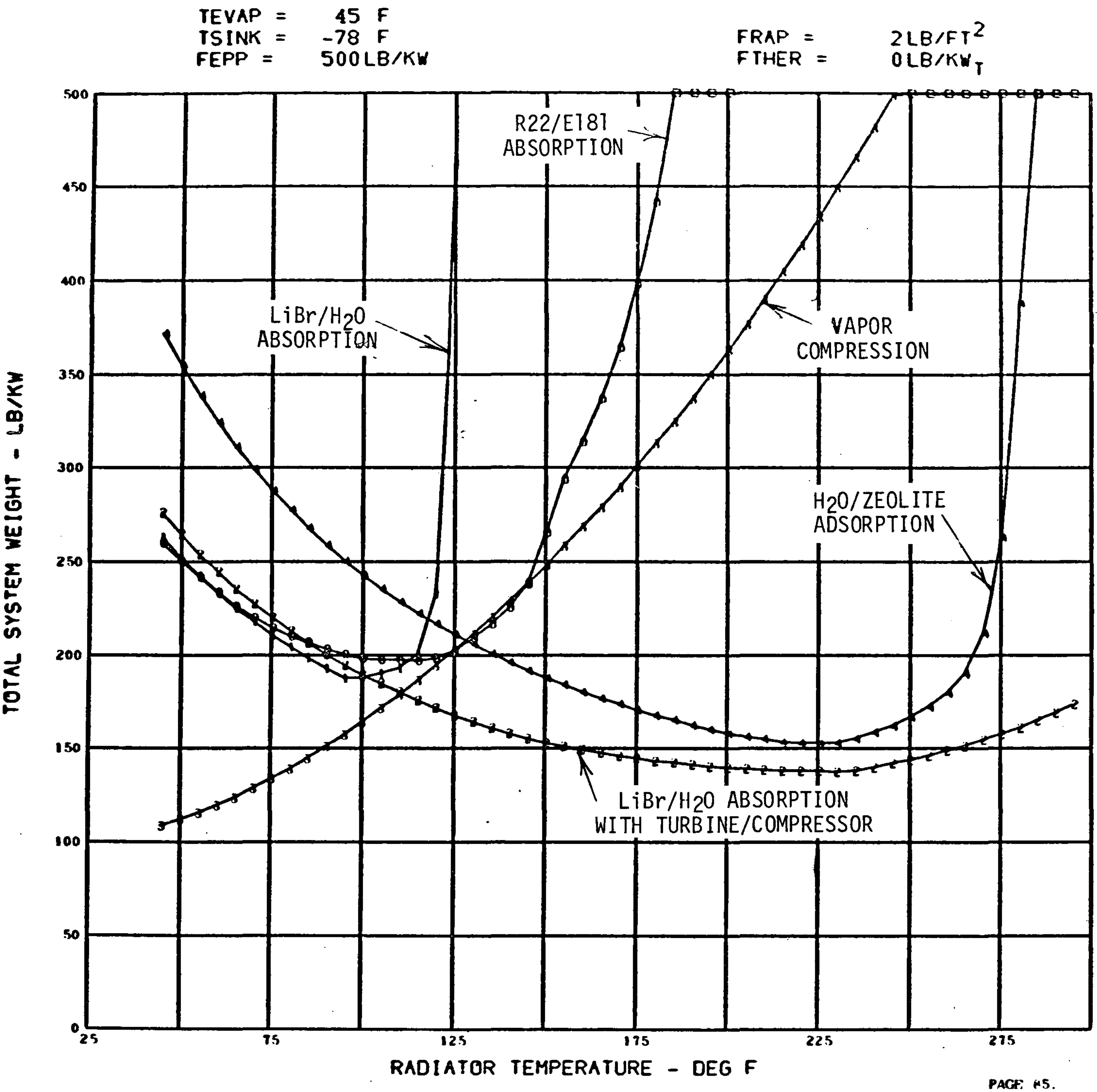


TEVAP = 45 F
TSINK = -78 F
FEPP = 500 LB/KW
FRAP = 1 LB/FT2
FTHER = 0 LB/KWT
R22/E181 ABSORPTION
VAPOR COMPRESSION
LiBr/H2O ABSORPTION
H2O/ZEOLITE ADSORPTION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEIGHT - LB/KW
RADIATOR TEMPERATURE - DEG F
0
50
100
150
200
250
300
350
400
450
500
25
75
125
175
225
275
PAGE 83.

TEVAP = 45 F
TSINK = -78 F
FEPP = 500LB/KW

FRAP = 1.5LB/FT^2
FTHER = 0LB/KW$_T$




TEVAP = 45 F
TSINK = -78 F
FEPP = 500LB/KW
FRAP = 2LB/FT^2
FTHER = 0LB/KW$_T$
TOTAL SYSTEM WEIGHT - LB/KW
RADIATOR TEMPERATURE - DEG F
R22/E181 ABSORPTION
LiBr/H$_2$O ABSORPTION
VAPOR COMPRESSION
H$_2$O/ZEOLITE ADSORPTION
LiBr/H$_2$O ABSORPTION WITH TURBINE/COMPRESSOR
0
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400
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175
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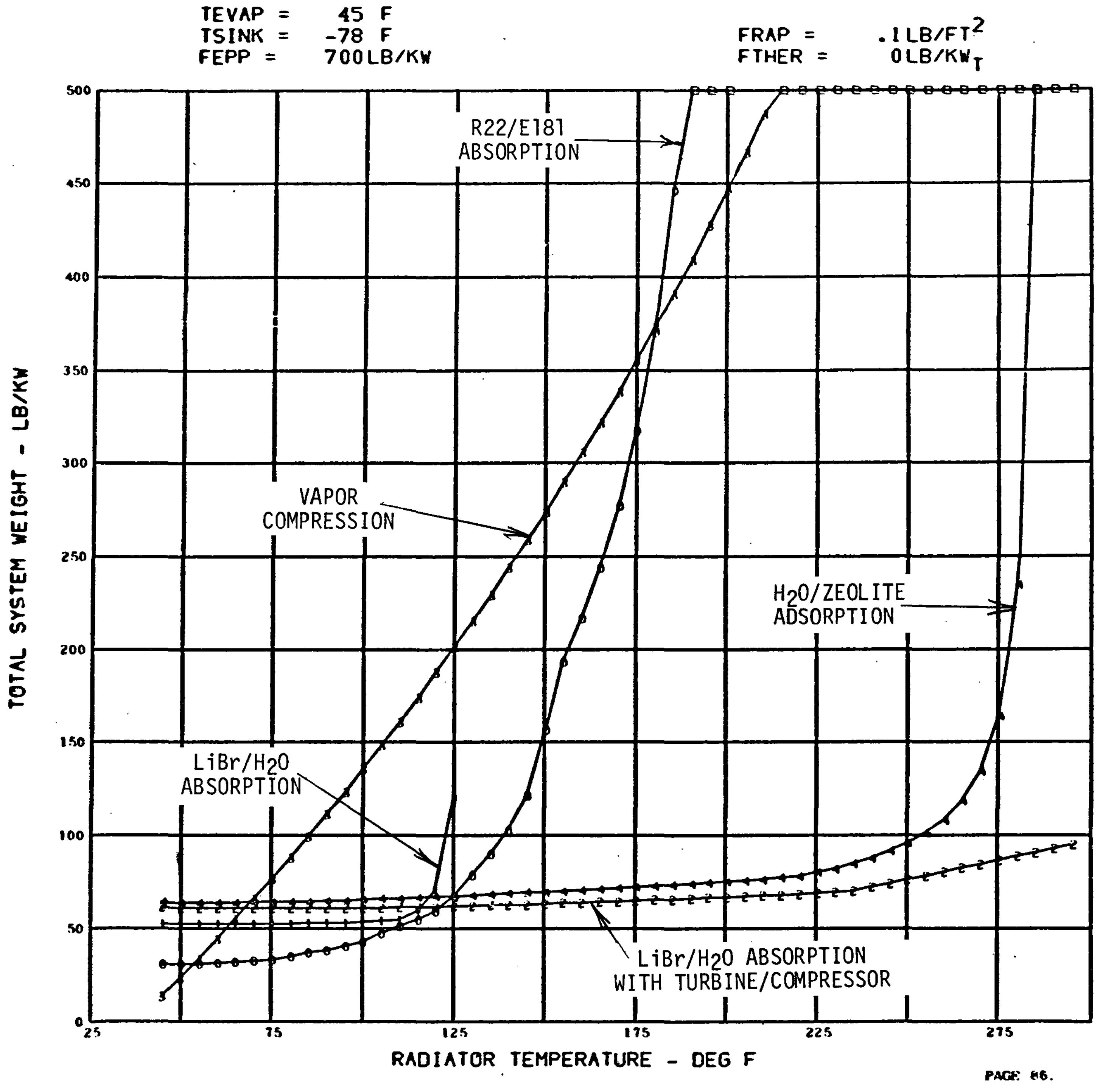
275
PAGE 85.



TEVAP = 45 F
TSINK = -78 F
FEPP = 700 LB/KW
FRAP = .1 LB/FT^2
FTHER = 0 LB/KW$_T$
R22/E181 ABSORPTION
VAPOR COMPRESSION
H_2O/ZEOLITE ADSORPTION
LiBr/H_2O ABSORPTION
LiBr/H_2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEIGHT - LB/KW
RADIATOR TEMPERATURE - DEG F
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100
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200
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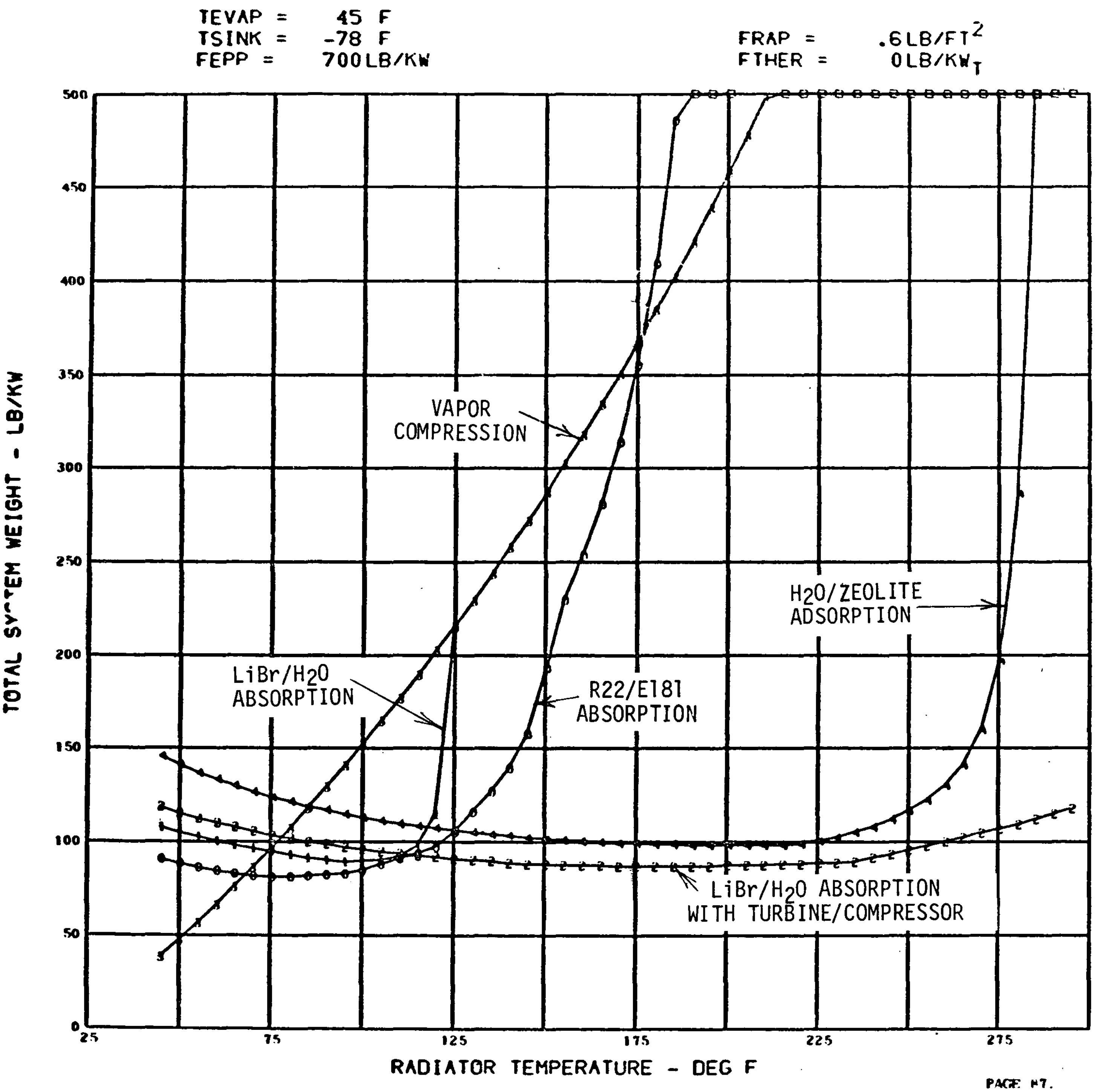
175
225
275
PAGE 86.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TSINK = -78 F
FEPP = 700LB/KW
FRAP = .6LB/FT2
FTHER = 0LB/KWT
TOTAL SYSTEM WEIGHT - LB/KW
RADIATOR TEMPERATURE - DEG F
VAPOR COMPRESSION
H2O/ZEOLITE ADSORPTION
LiBr/H2O ABSORPTION
R22/E181 ABSORPTION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
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500
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125
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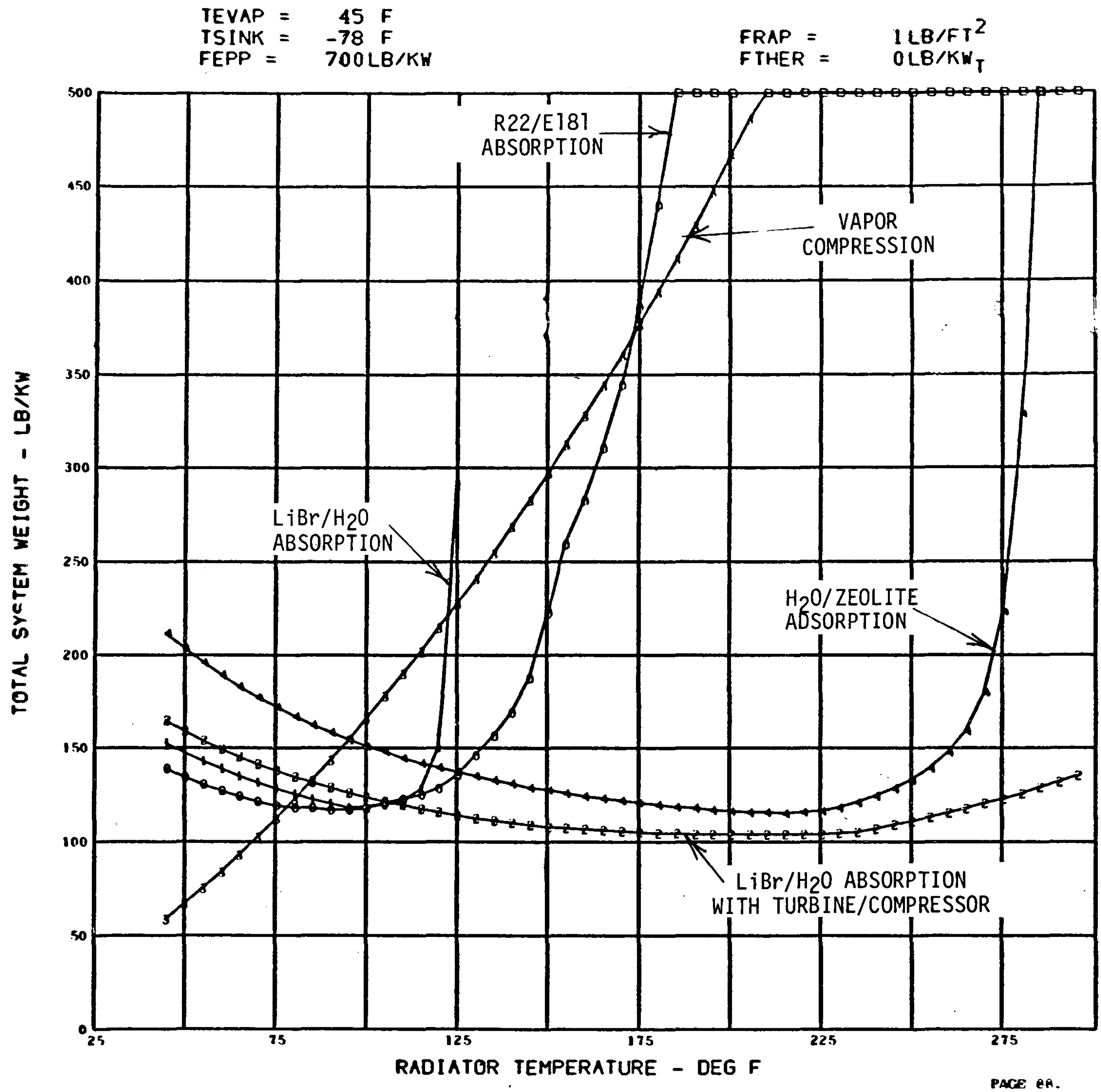
225
275
PAGE 87.



TEVAP = 45 F
TSINK = -78 F
FEPP = 700LB/KW
FRAP = 1LB/FT2
FTHER = 0LB/KWT
R22/E181 ABSORPTION
VAPOR COMPRESSION
LiBr/H2O ABSORPTION
H2O/ZEOLITE ABSORPTION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEIGHT - LB/KW
RADIATOR TEMPERATURE - DEG F
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200
250
300
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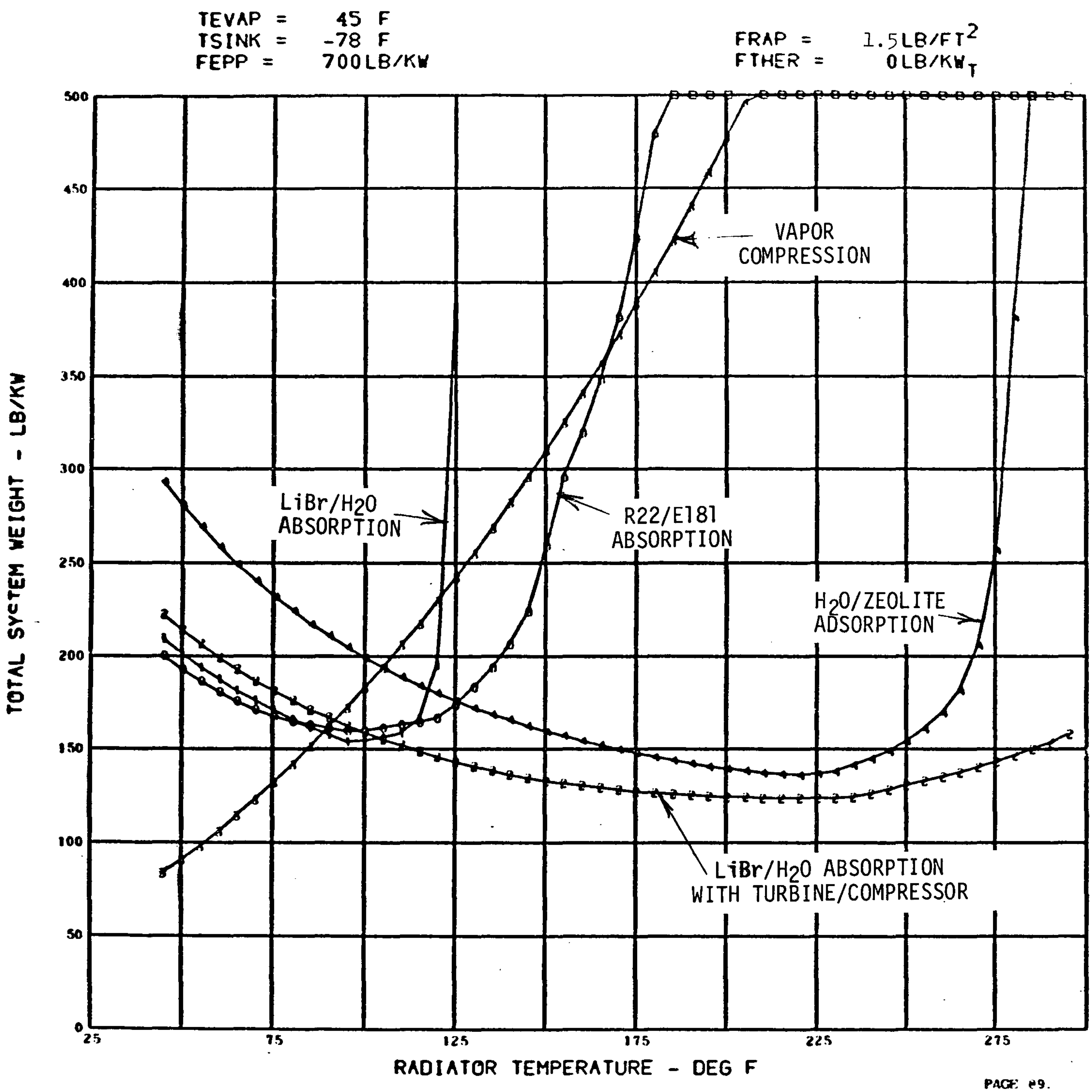
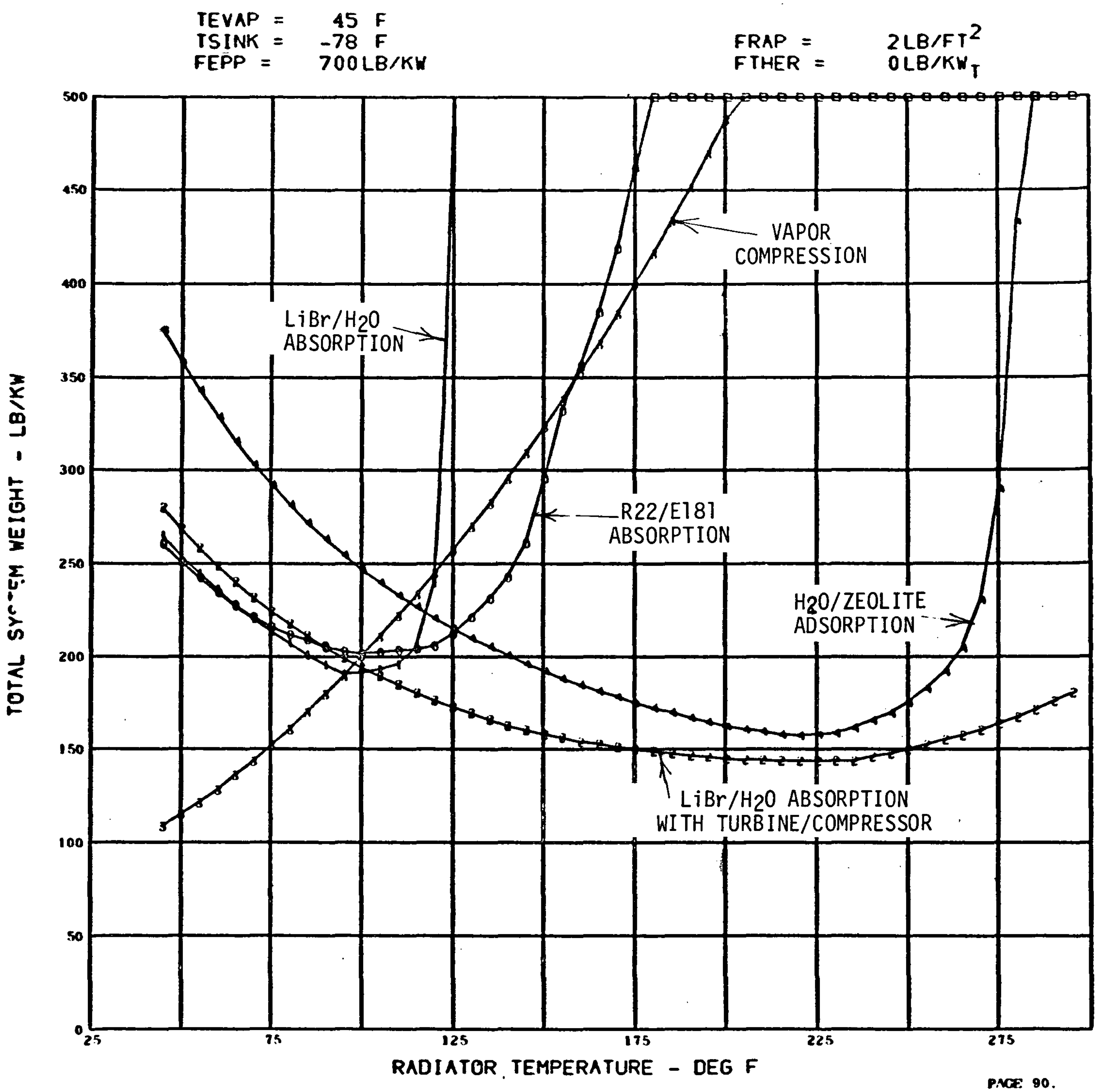
125
175
225
275
PAGE 88.



TEVAP = 45 F
TSINK = -78 F
FEPP = 700LB/KW
FRAP = 1.5LB/FT2
FTHER = 0LB/KWT
VAPOR COMPRESSION
LiBr/H2O ABSORPTION
R22/E181 ABSORPTION
H2O/ZEOLITE ADSORPTION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEIGHT - LB/KW
RADIATOR TEMPERATURE - DEG F
0
50
100
150
200
250
300
350
400
450
500
25
75
125
175
225
275
PAGE 89.


TEVAP = 45 F
TSINK = -78 F
FEPP = 700 LB/KW
FRAP = 2 LB/FT2
FTHER = 0 LB/KWT
VAPOR COMPRESSION
LiBr/H2O ABSORPTION
R22/E181 ABSORPTION
H2O/ZEOLITE ADSORPTION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEIGHT - LB/KW
RADIATOR TEMPERATURE - DEG F
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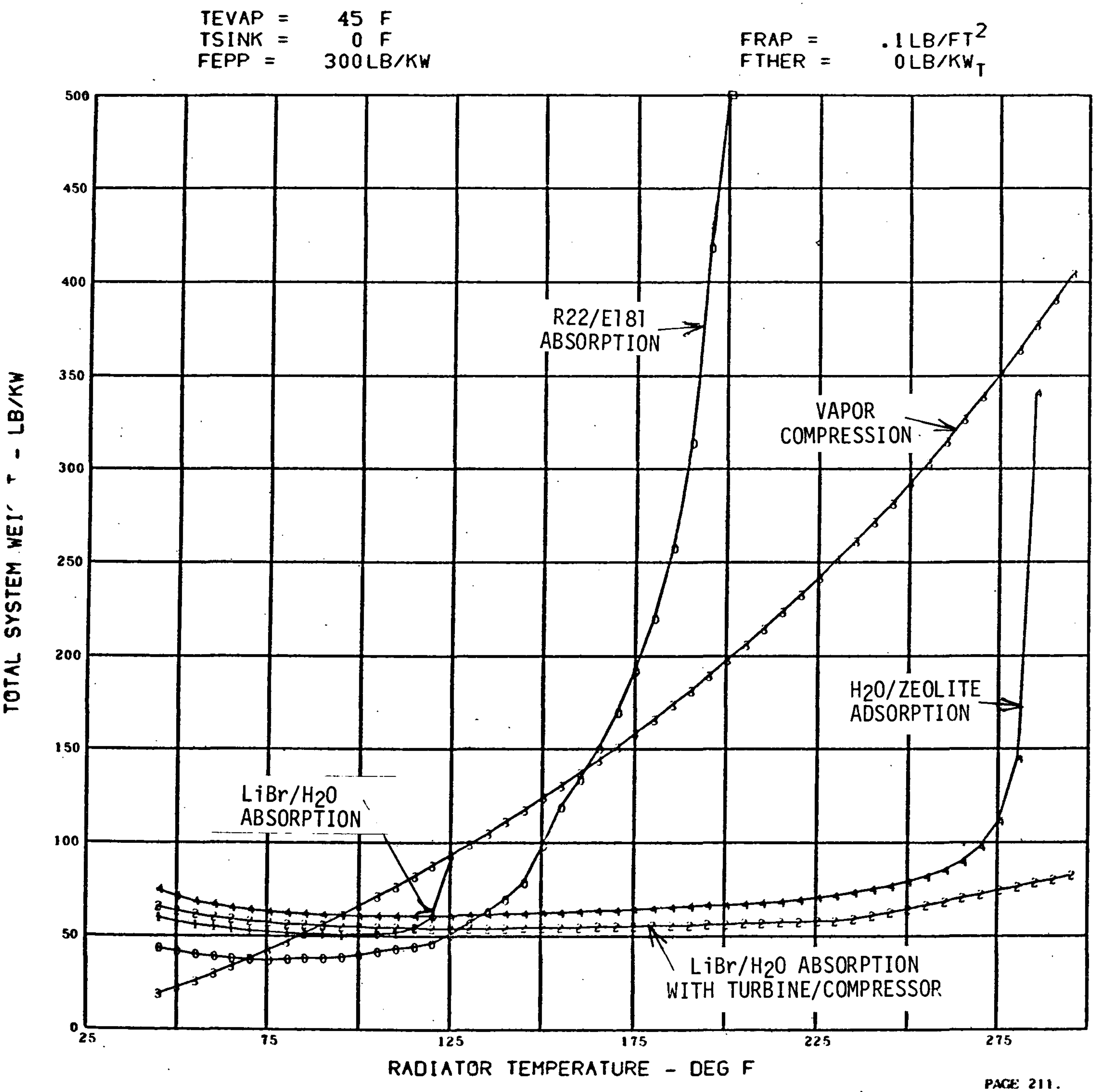
100
150
200
250
300
350
400
450
500
25
75
125
175
225
275
PAGE 90.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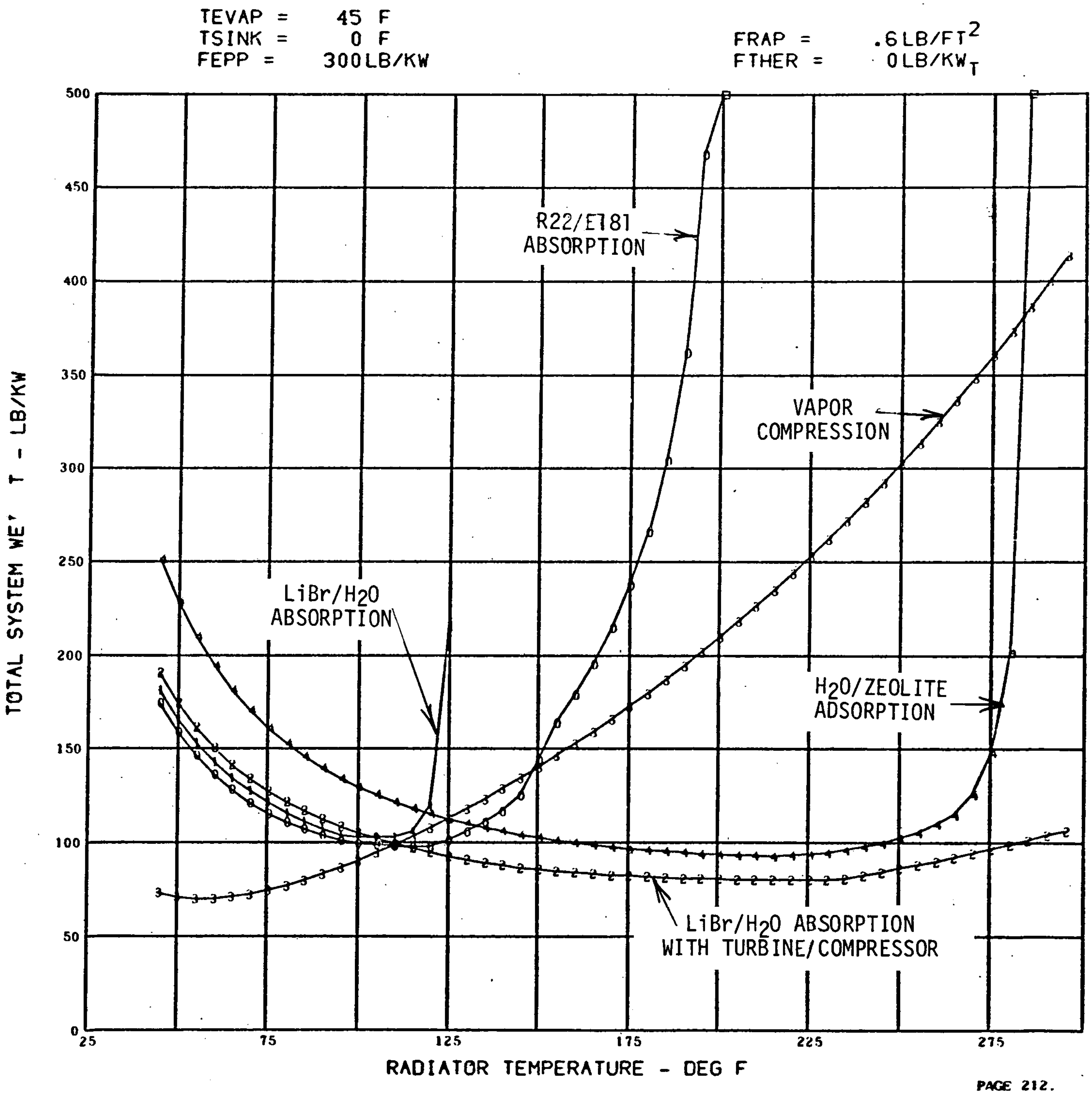
TSINK = 0 F
FEPP = 300LB/KW
FRAP = .1LB/FT2
FTHER = 0LB/KWT
TOTAL SYSTEM WEI T - LB/KW
RADIATOR TEMPERATURE - DEG F
R22/E181 ABSORPTION
VAPOR COMPRESSION
H2O/ZEOLITE ADSORPTION
LiBr/H2O ABSORPTION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
0
50
100
150
200
250
300
350
400
450
500
25
75
125
175
225
275
PAGE 211.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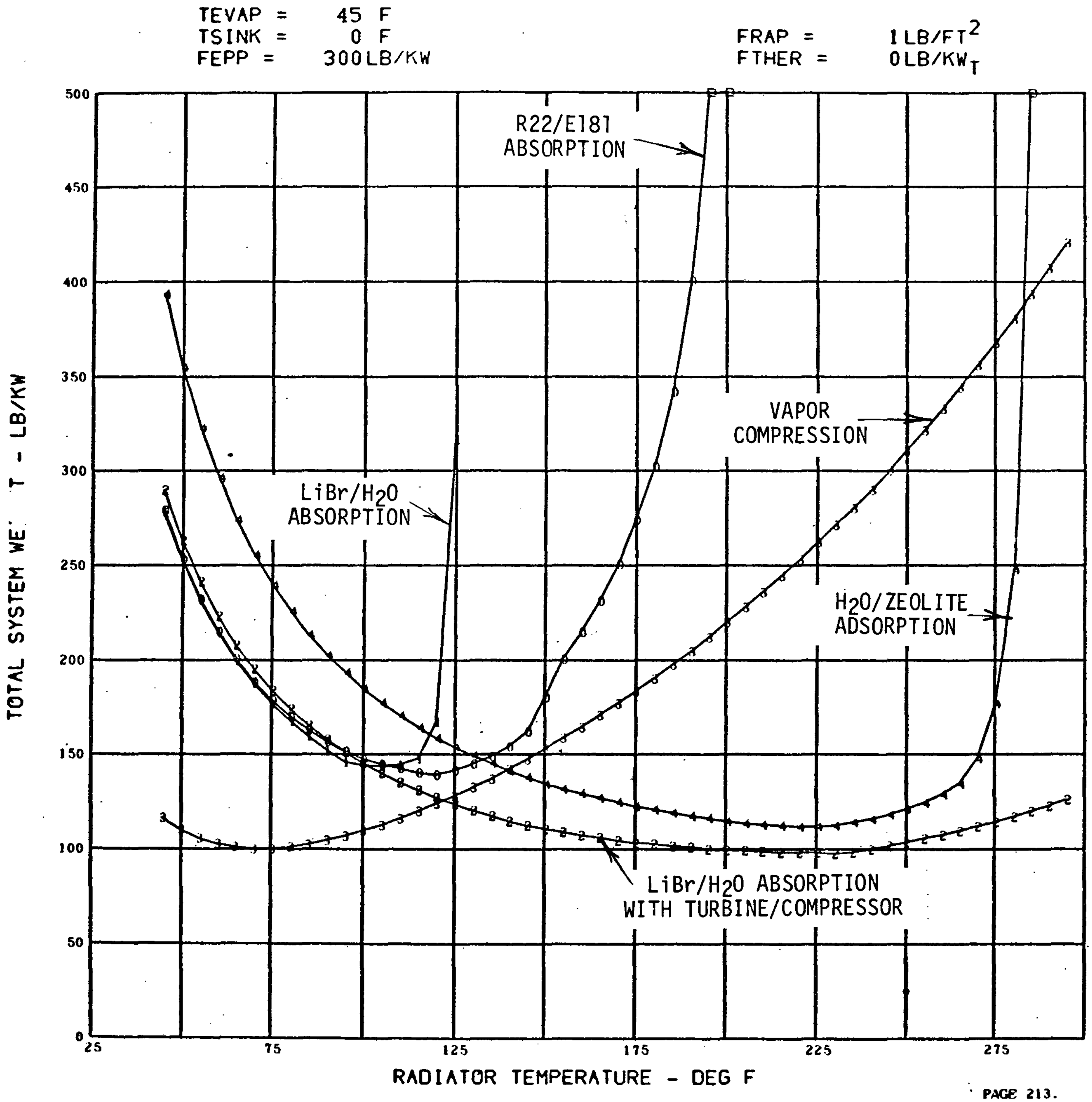
TSINK = 0 F
FEPP = 300LB/KW
FRAP = .6LB/FT2
FTHER = 0LB/KWT
R22/E181 ABSORPTION
VAPOR COMPRESSION
LiBr/H2O ABSORPTION
H2O/ZEOLITE ADSORPTION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WE' T - LB/KW
RADIATOR TEMPERATURE - DEG F
500
450
400
350
300
250
200
150
100
50
0
25
75
125
175
225
275
PAGE 212.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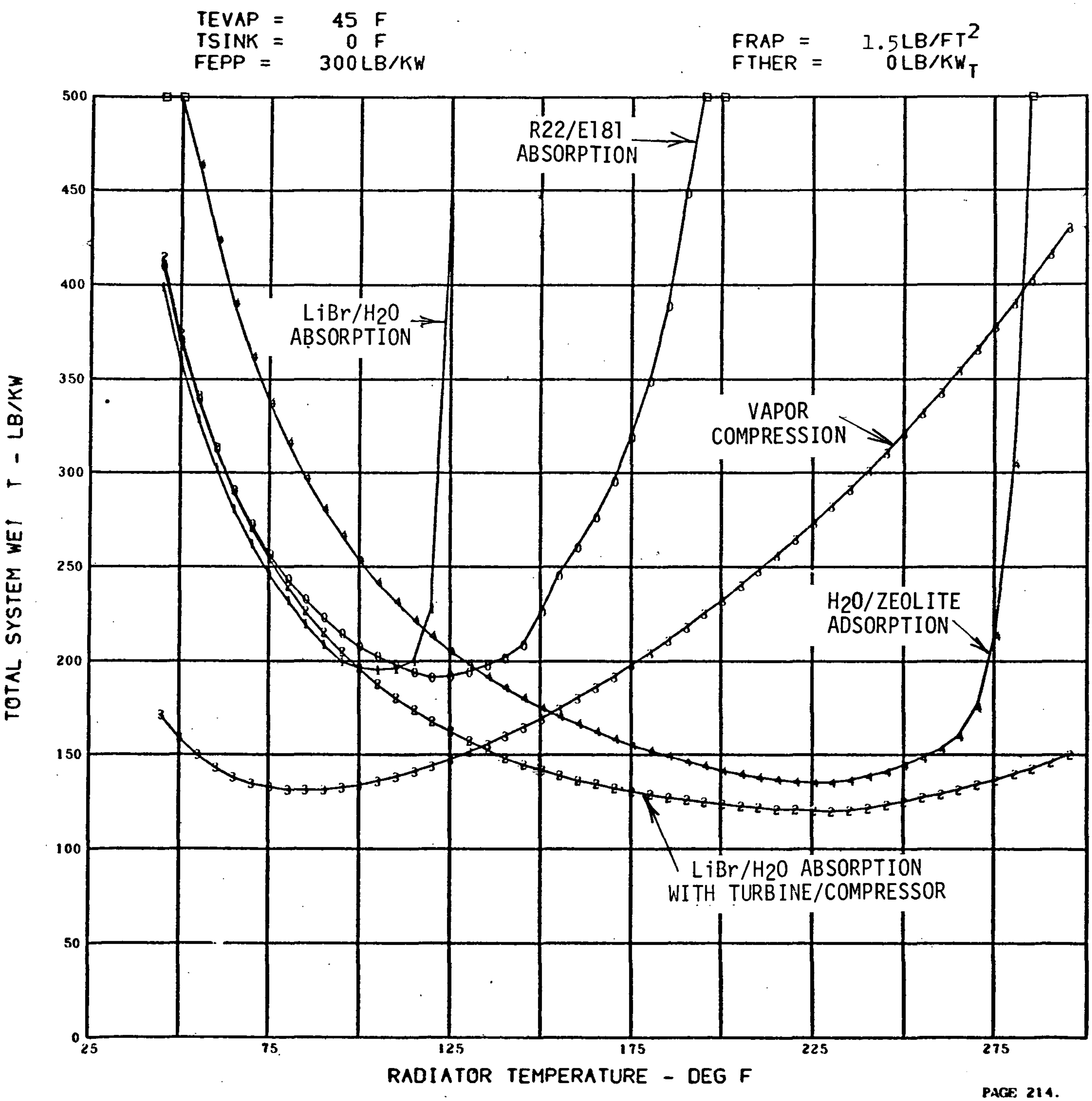
TSINK = 0 F
FEPP = 300LB/KW
FRAP = 1LB/FT2
FTHER = 0LB/KWT
R22/E181 ABSORPTION
VAPOR COMPRESSION
LiBr/H2O ABSORPTION
H2O/ZEOLITE ADSORPTION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WE T - LB/KW
RADIATOR TEMPERATURE - DEG F
500
450
400
350
300
250
200
150
100
50
0
25
75
125
175
225
275
PAGE 213.



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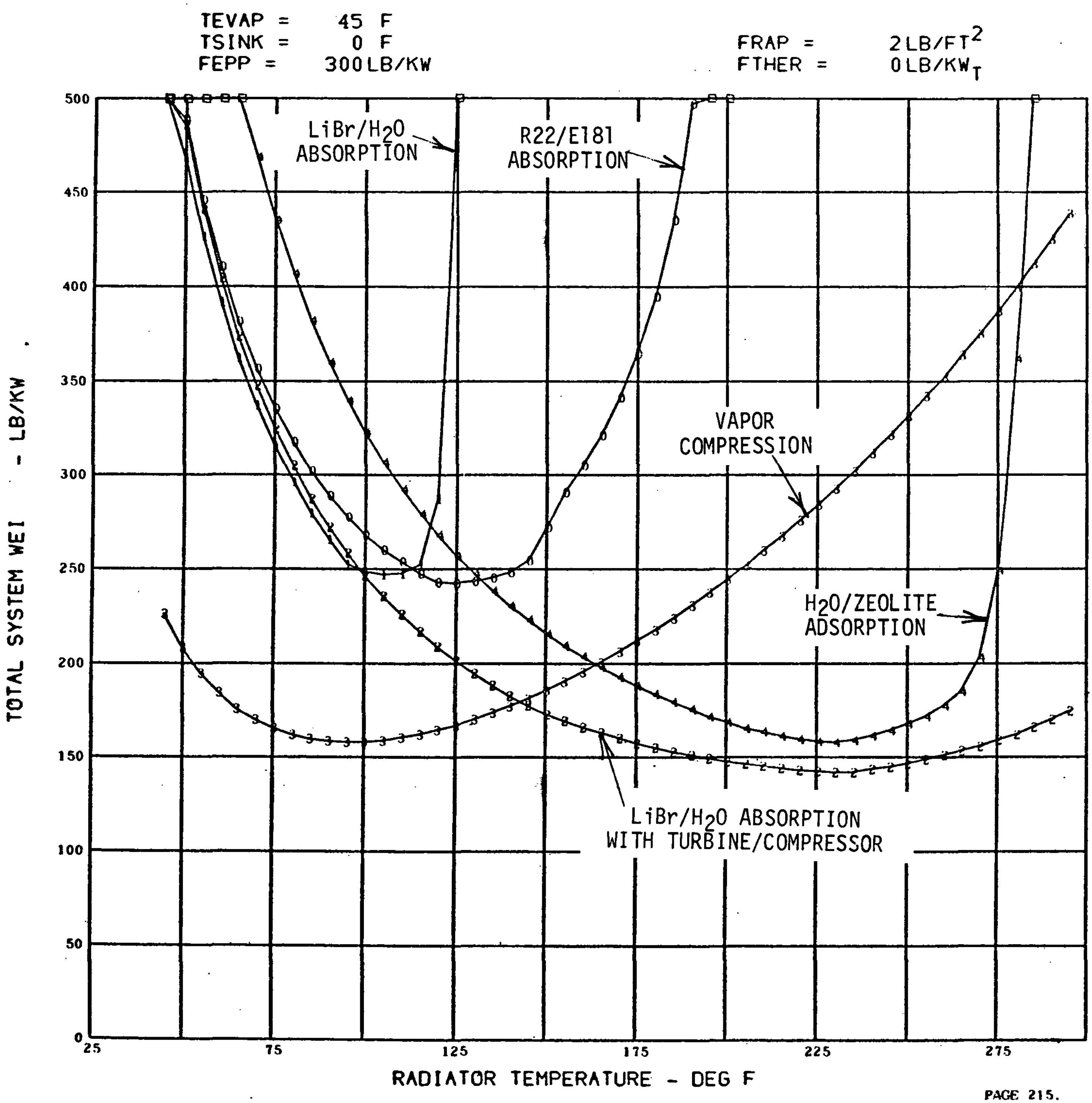
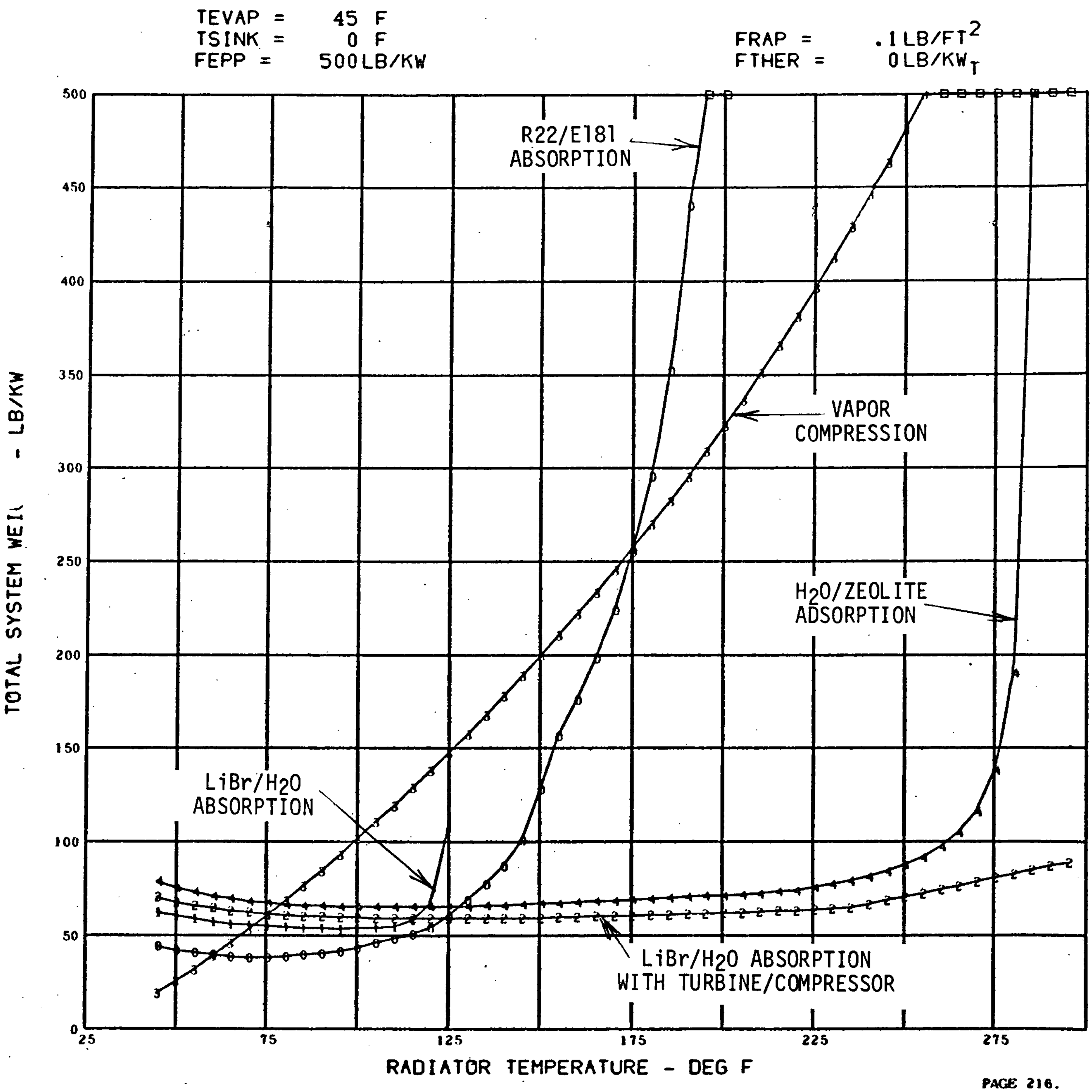
TSINK = 0 F
FEPP = 300 LB/KW
FRAP = 1.5 LB/FT2
FTHER = 0 LB/KWT
R22/E181 ABSORPTION
LiBr/H2O ABSORPTION
VAPOR COMPRESSION
H2O/ZEOLITE ADSORPTION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEIGHT - LB/KW
RADIATOR TEMPERATURE - DEG F
500
450
400
350
300
250
200
150
100
50
0
25
75
125
175
225
275
PAGE 214.


TEVAP = 45 F
TSINK = 0 F
FEPP = 300 LB/KW
FRAP = 2 LB/FT2
FTHER = 0 LB/KWT
TOTAL SYSTEM WEI - LB/KW
RADIATOR TEMPERATURE - DEG F
LiBr/H2O ABSORPTION
R22/E181 ABSORPTION
VAPOR COMPRESSION
H2O/ZEOLITE ADSORPTION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
PAGE 215.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TSINK = 0 F
FEPP = 500 LB/KW
FRAP = .1 LB/FT^2
FTHER = 0 LB/KW$_T$
R22/E181 ABSORPTION
VAPOR COMPRESSION
H_2O/ZEOLITE ADSORPTION
LiBr/H_2O ABSORPTION
LiBr/H_2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEI(- LB/KW
RADIATOR TEMPERATURE - DEG F
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50
100
150
200
250
300
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400
450
500
25
75
125
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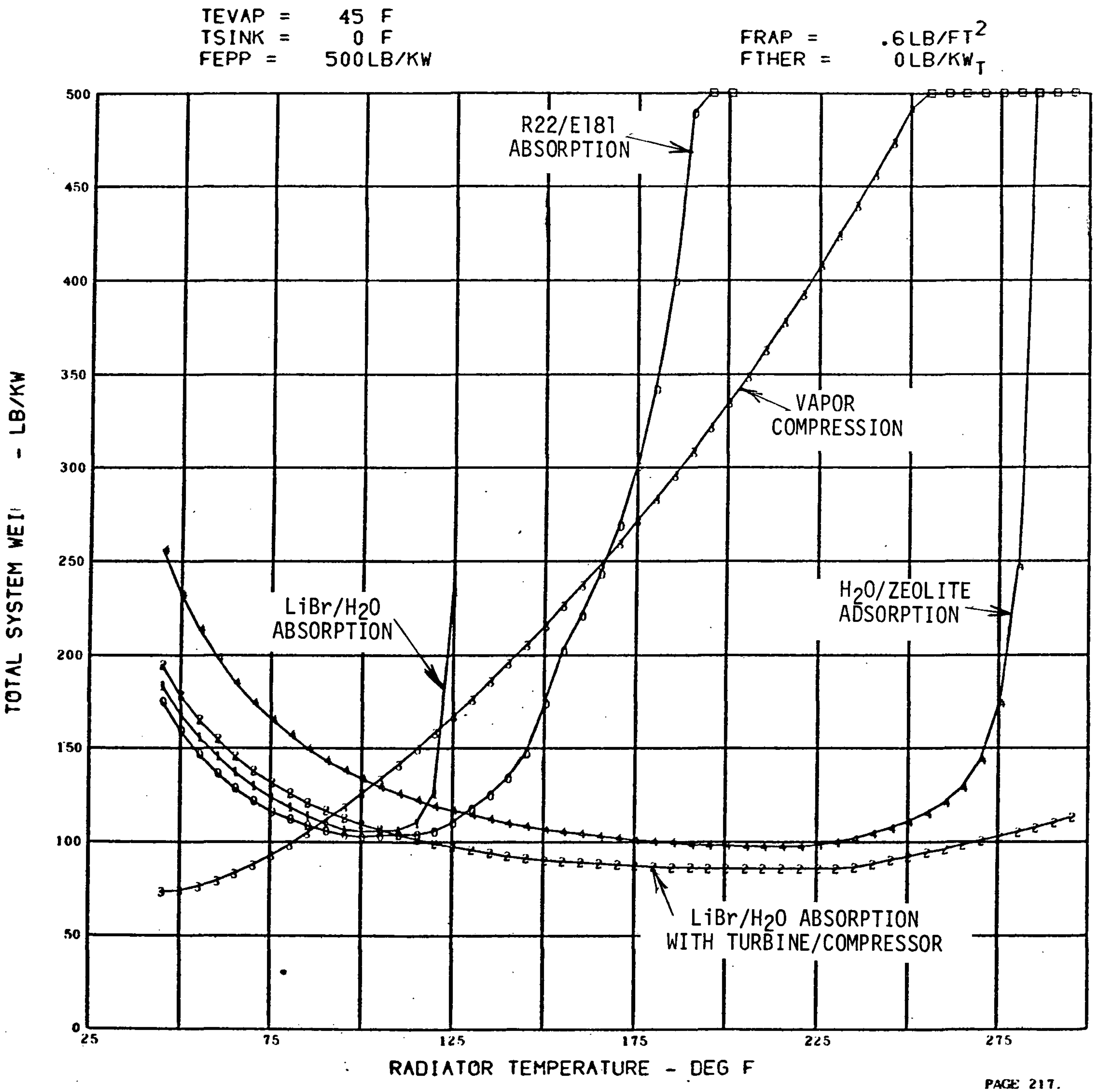
225
275
PAGE 216.



TEVAP = 45 F
TSINK = 0 F
FEPP = 500LB/KW
FRAP = .6LB/FT2
FTHER = 0LB/KWT
R22/E181
ABSORPTION
VAPOR
COMPRESSION
LiBr/H2O
ABSORPTION
H2O/ZEOLITE
ADSORPTION
LiBr/H2O ABSORPTION
WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEI - LB/KW
RADIATOR TEMPERATURE - DEG F
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150
200
250
300
350
400
450
500
25
75
125
175
225
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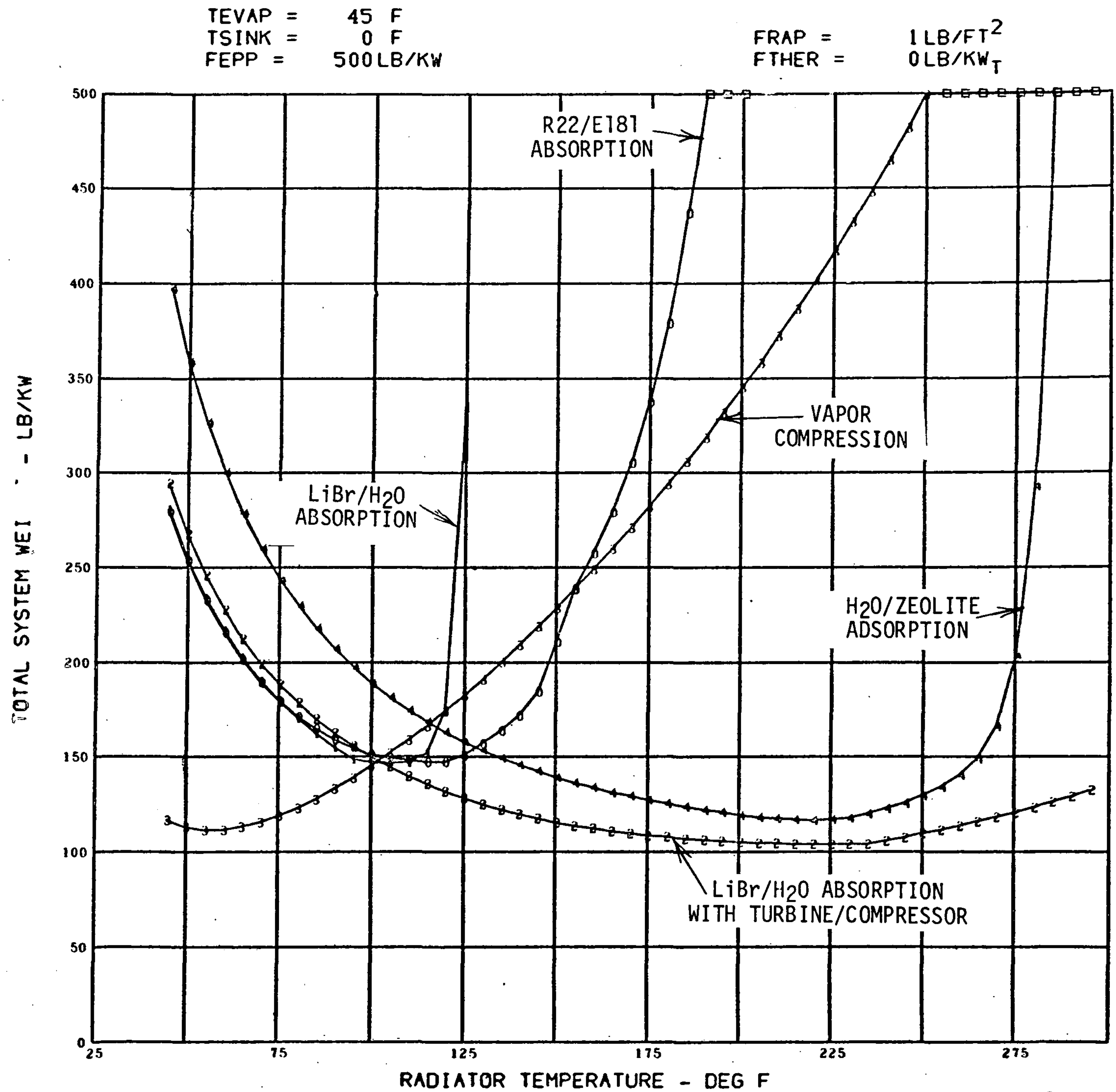
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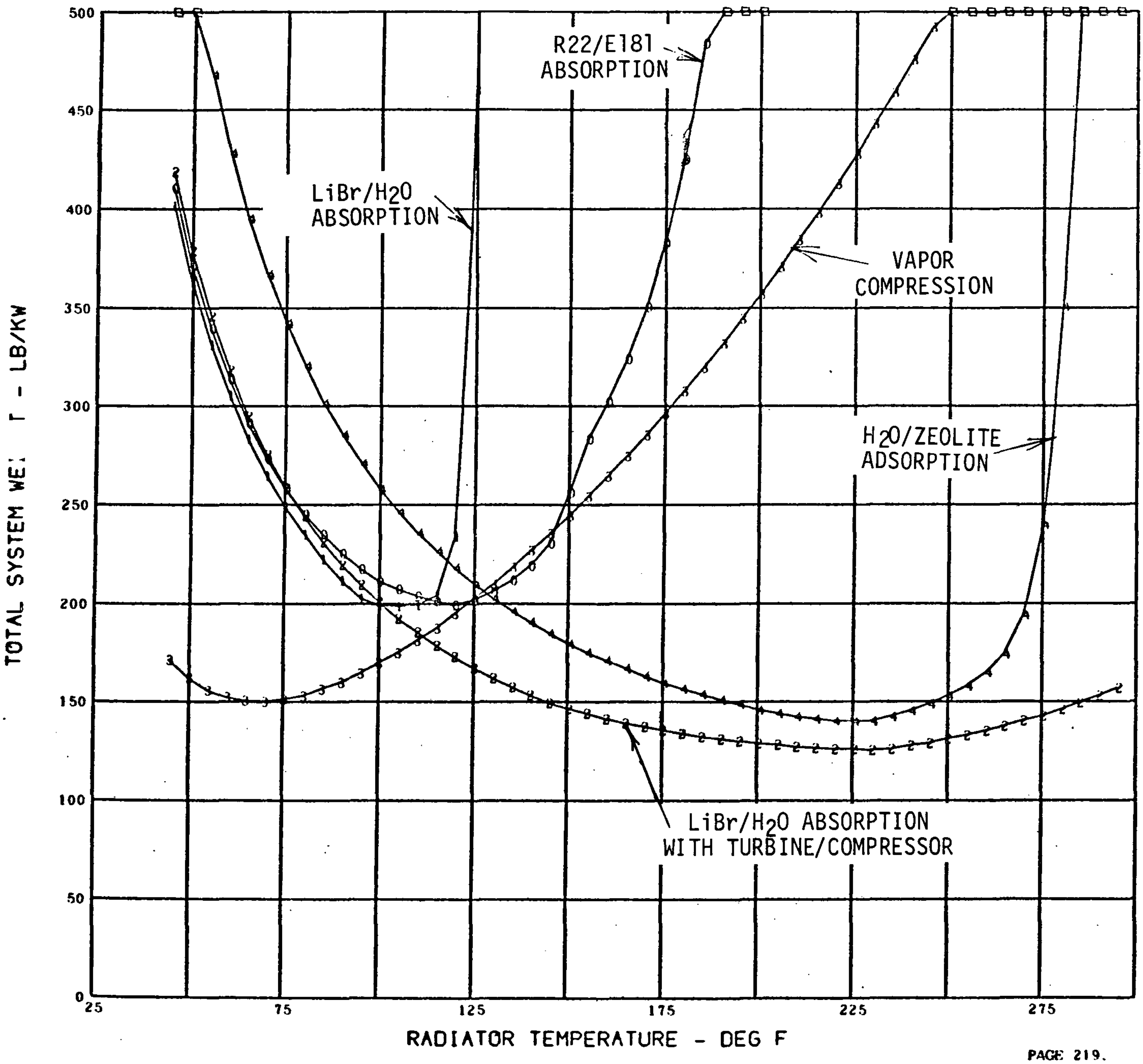
FTHER = 0 LB/KWT
R22/E181 ABSORPTION
VAPOR COMPRESSION
LiBr/H2O ABSORPTION
H2O/ZEOLITE ADSORPTION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEI - LB/KW
RADIATOR TEMPERATURE - DEG F
0
50
100
150
200
250
300
350
400
450
500
25
75
125
175
225
275

PAGE 218.

TEVAP = 45 F
TSINK = 0 F
FEPP = 500LB/KW

FRAP = 1.5LB/FT^2
FTHER = 0LB/KW$_T$



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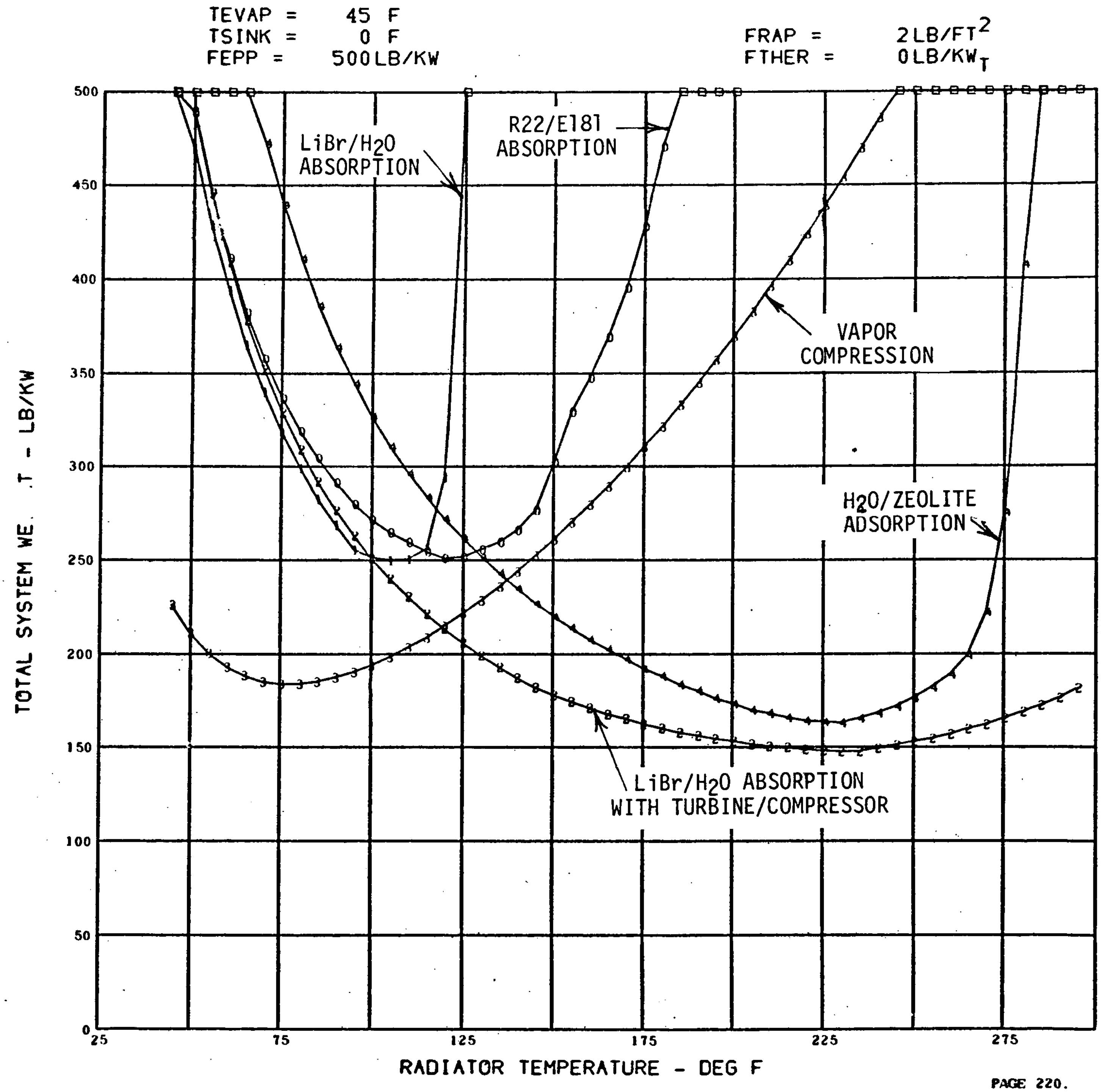
TEVAP = 45 F
TSINK = 0 F
FEPP = 500LB/KW
FRAP = 2LB/FT2
FTHER = 0LB/KWT
LiBr/H2O ABSORPTION
R22/E181 ABSORPTION
VAPOR COMPRESSION
H2O/ZEOLITE ADSORPTION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WE. T - LB/KW
RADIATOR TEMPERATURE - DEG F
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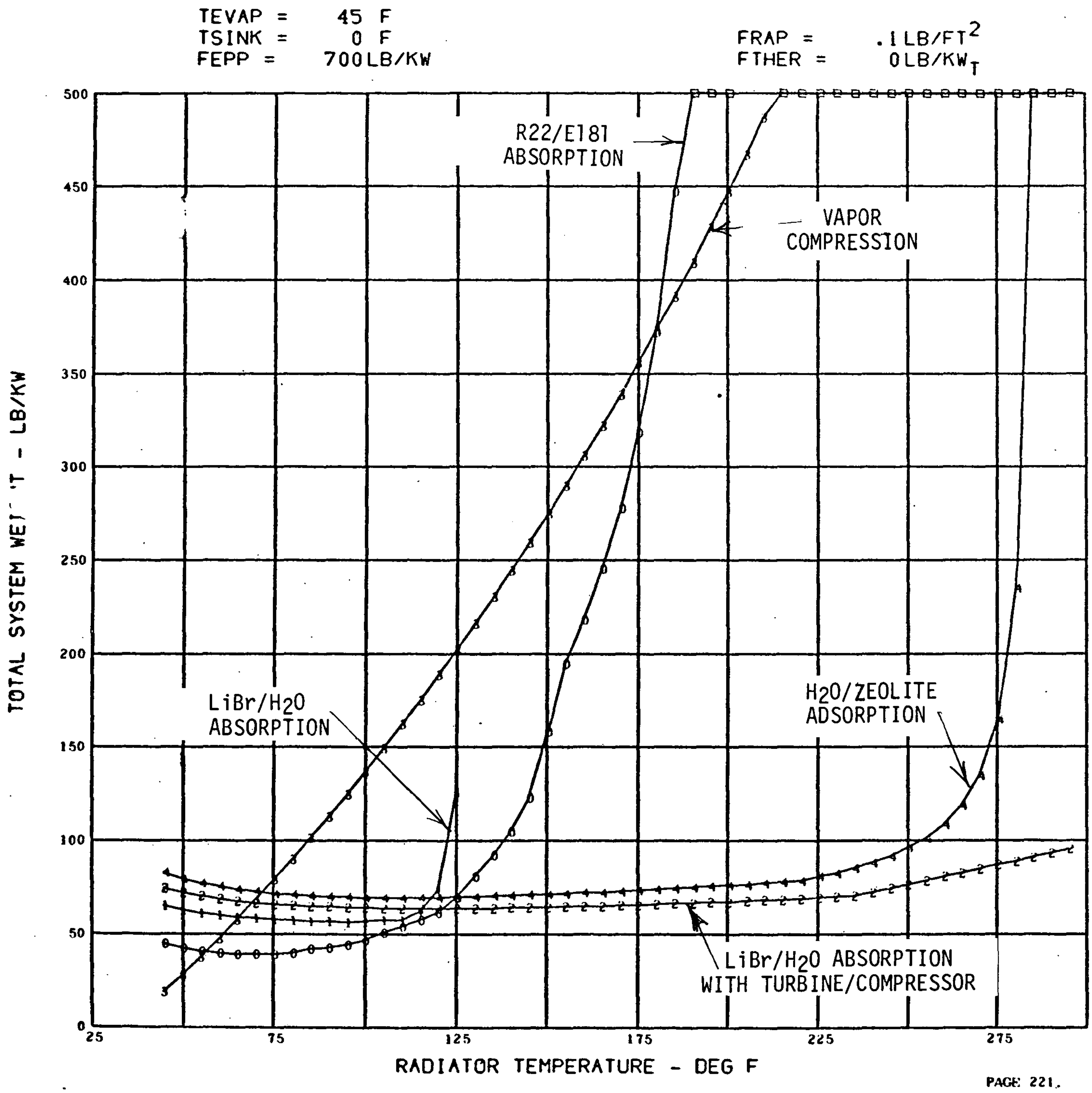
PAGE 220.


TEVAP = 45 F
TSINK = 0 F
FEPP = 700 LB/KW
FRAP = .1 LB/FT2
FTHER = 0 LB/KWT
R22/E181 ABSORPTION
VAPOR COMPRESSION
LiBr/H2O ABSORPTION
H2O/ZEOLITE ADSORPTION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEI 'T - LB/KW
RADIATOR TEMPERATURE - DEG F
0
50
100
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75
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175
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275
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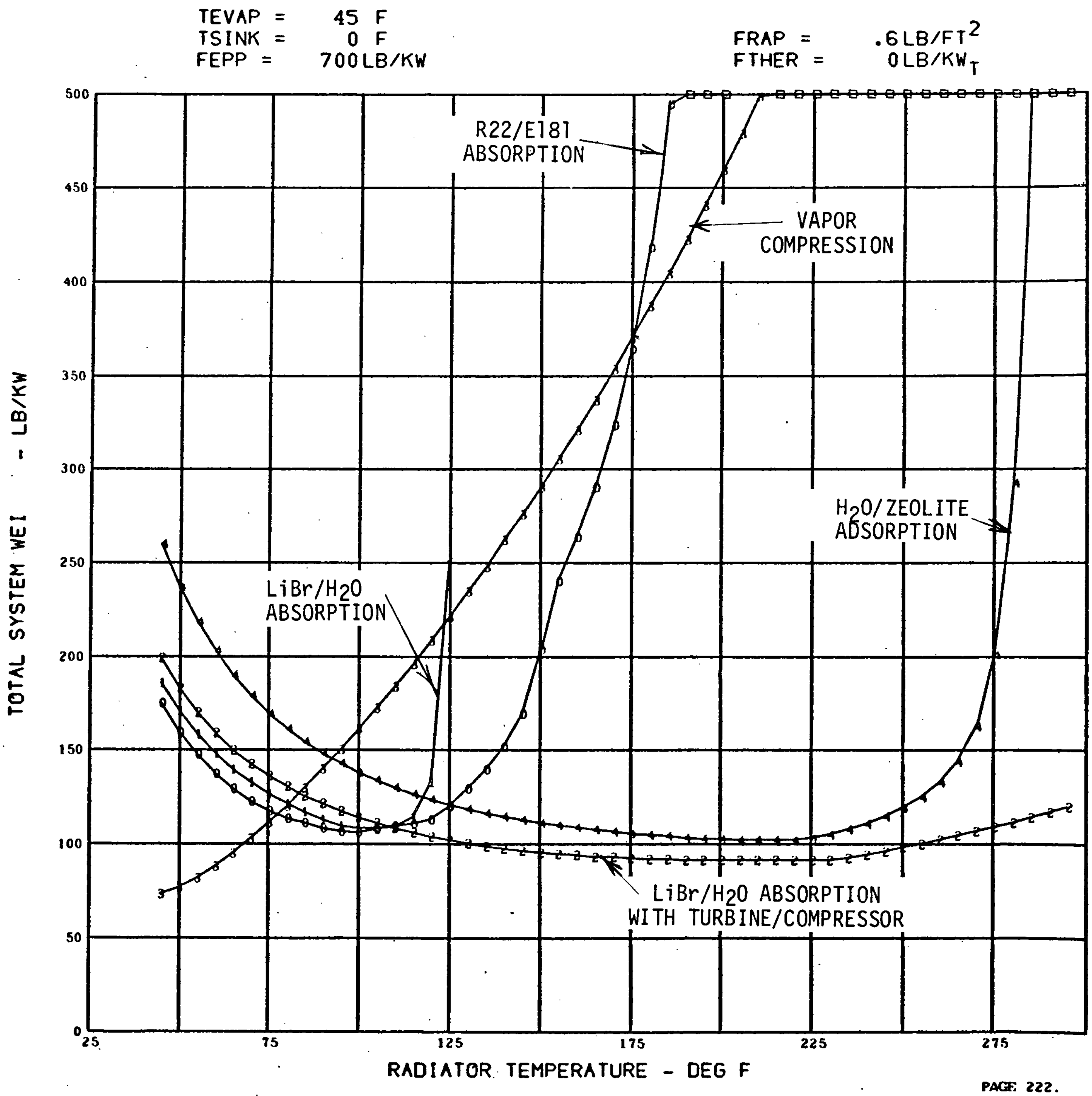
TEVAP = 45 F
TSINK = 0 F
FEPP = 700 LB/KW
FRAP = .6 LB/FT^2
FTHER = 0 LB/KW$_T$
R22/E181 ABSORPTION
VAPOR COMPRESSION
H_2O/ZEOLITE ABSORPTION
LiBr/H_2O ABSORPTION
LiBr/H_2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEI - LB/KW
RADIATOR TEMPERATURE - DEG F
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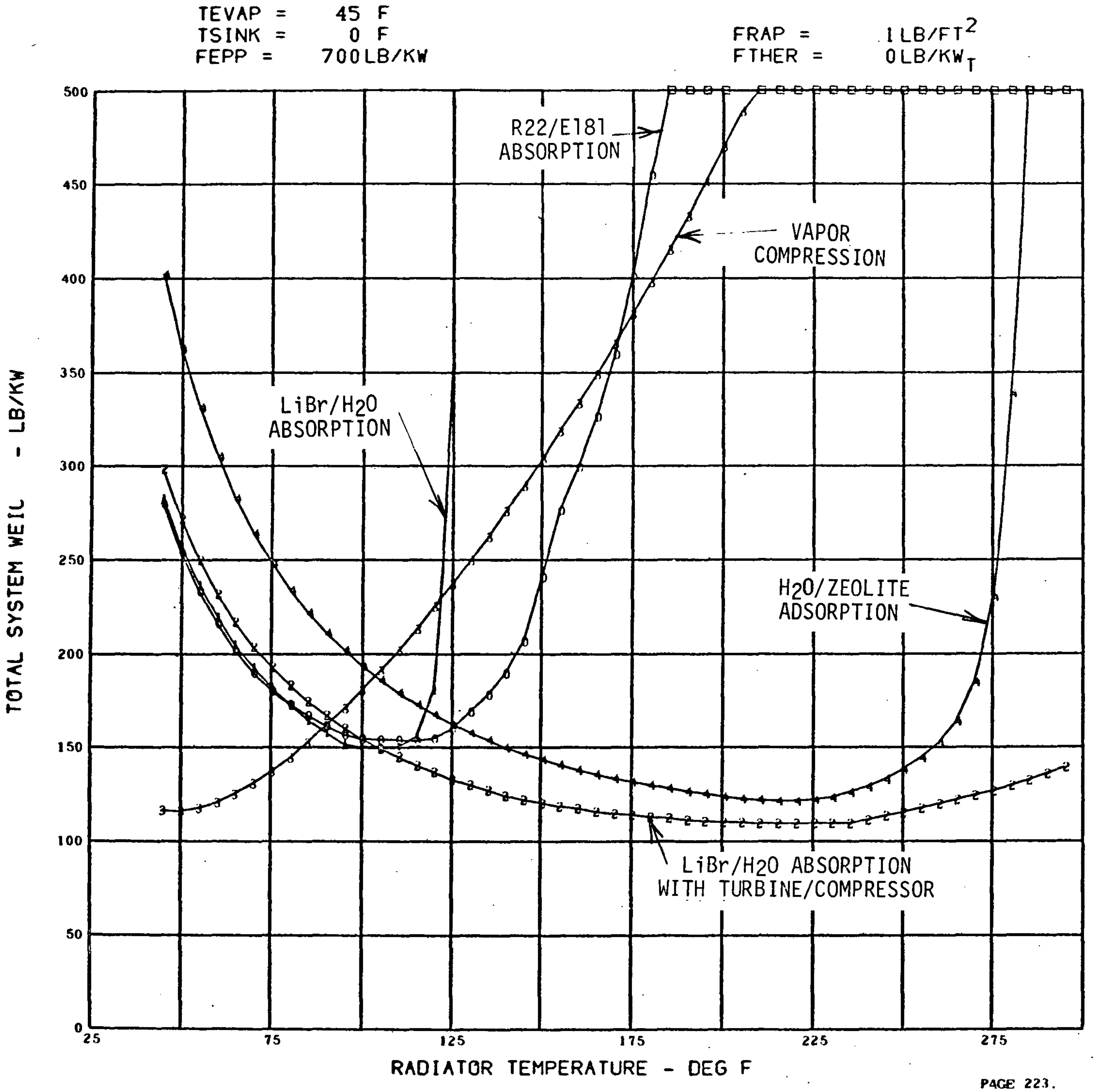

TEVAP = 45 F
TSINK = 0 F
FEPP = 700 LB/KW
FRAP = 1 LB/FT2
FTHER = 0 LB/KWT
R22/E181 ABSORPTION
VAPOR COMPRESSION
LiBr/H2O ABSORPTION
H2O/ZEOLITE ADSORPTION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEIL - LB/KW
RADIATOR TEMPERATURE - DEG F
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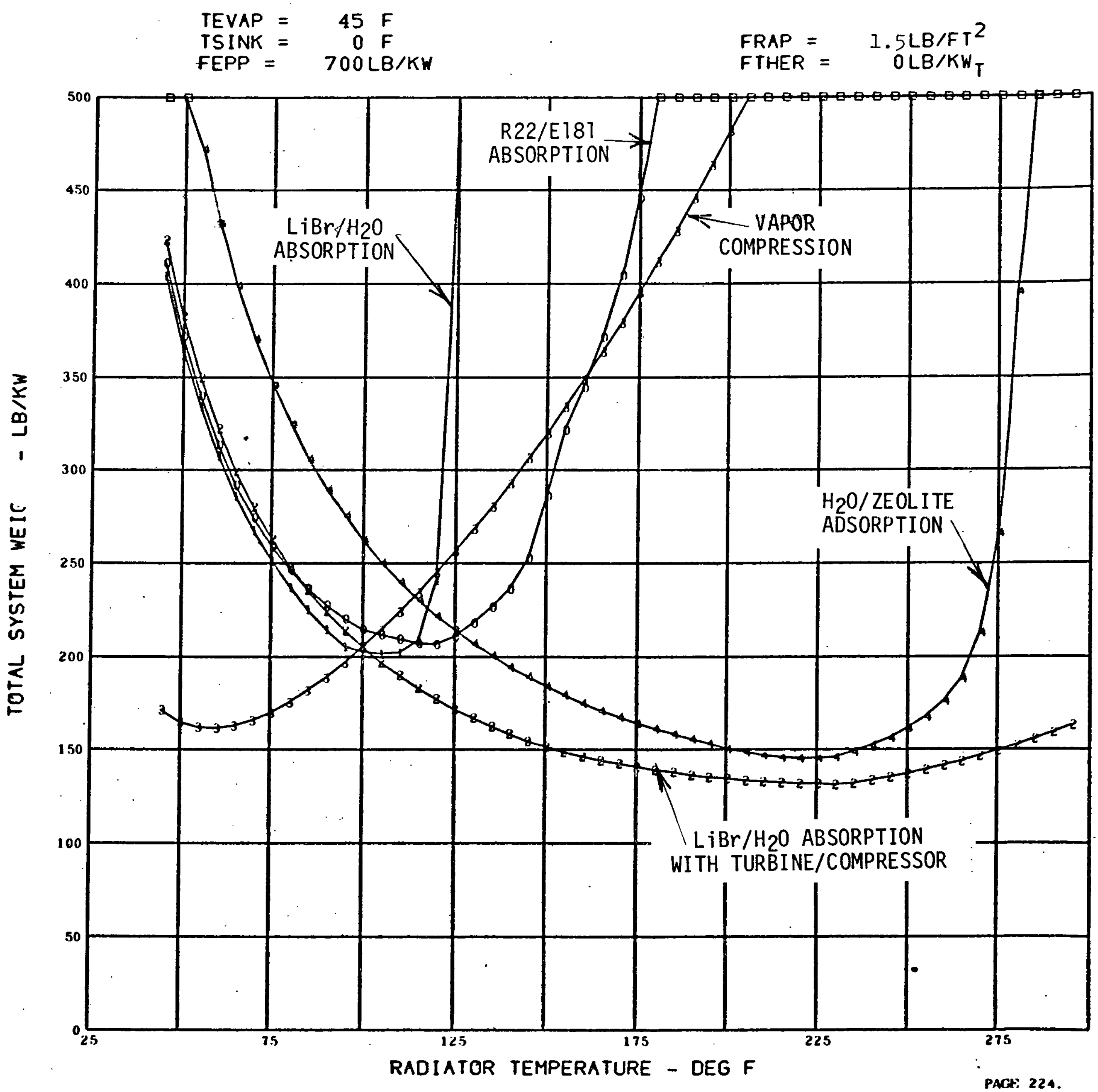
TEVAP = 45 F
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FRAP = 1.5 LB/FT2
FTHER = 0 LB/KWT
R22/E181 ABSORPTION
LiBr/H2O ABSORPTION
VAPOR COMPRESSION
H2O/ZEOLITE ADSORPTION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEIG - LB/KW
RADIATOR TEMPERATURE - DEG F
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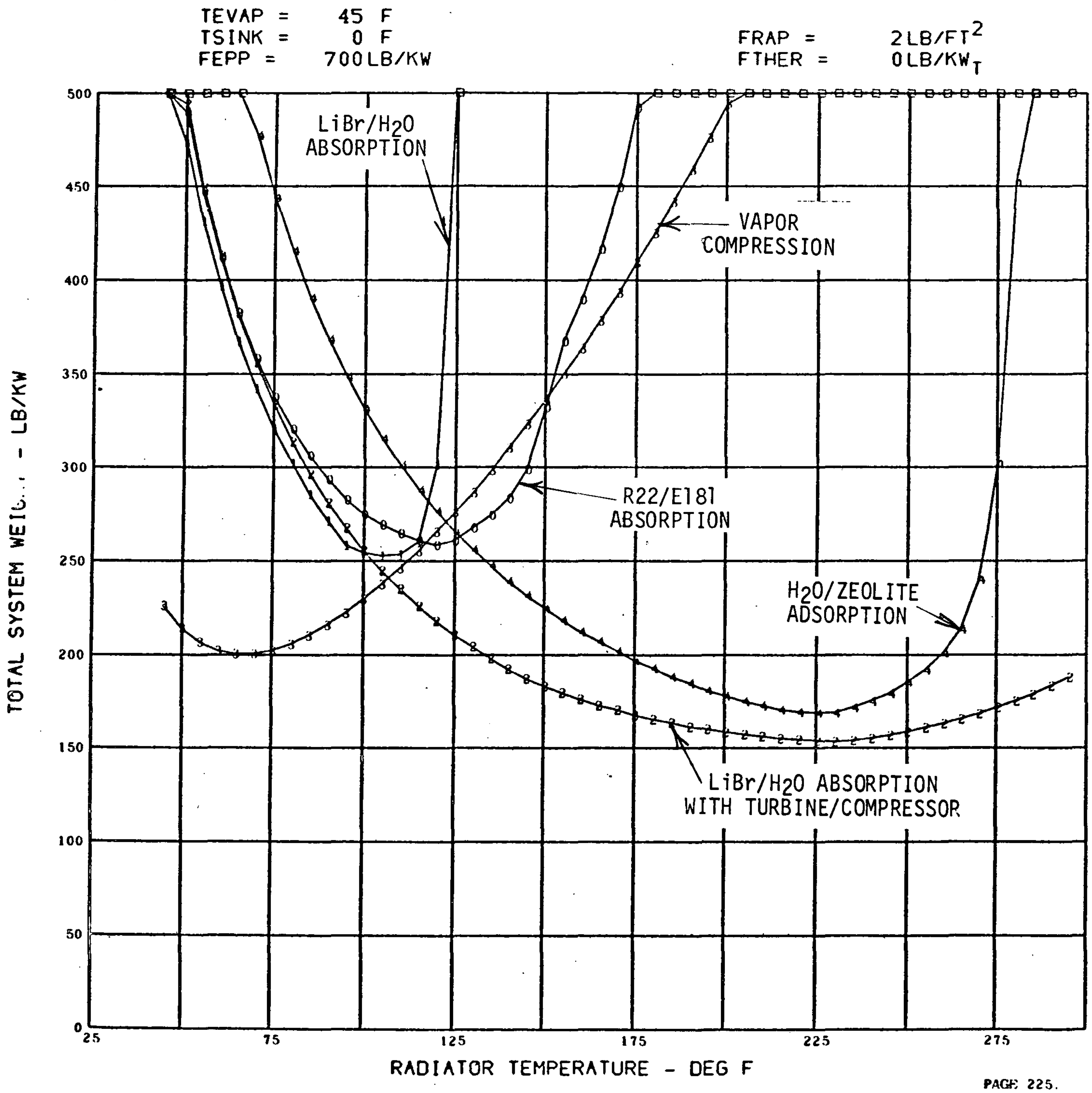

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FRAP = 2 LB/FT^2
FTHER = 0 LB/KW$_T$
LiBr/H$_2$O ABSORPTION
VAPOR COMPRESSION
R22/E181 ABSORPTION
H$_2$O/ZEOLITE ADSORPTION
LiBr/H$_2$O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEIG... - LB/KW
RADIATOR TEMPERATURE - DEG F
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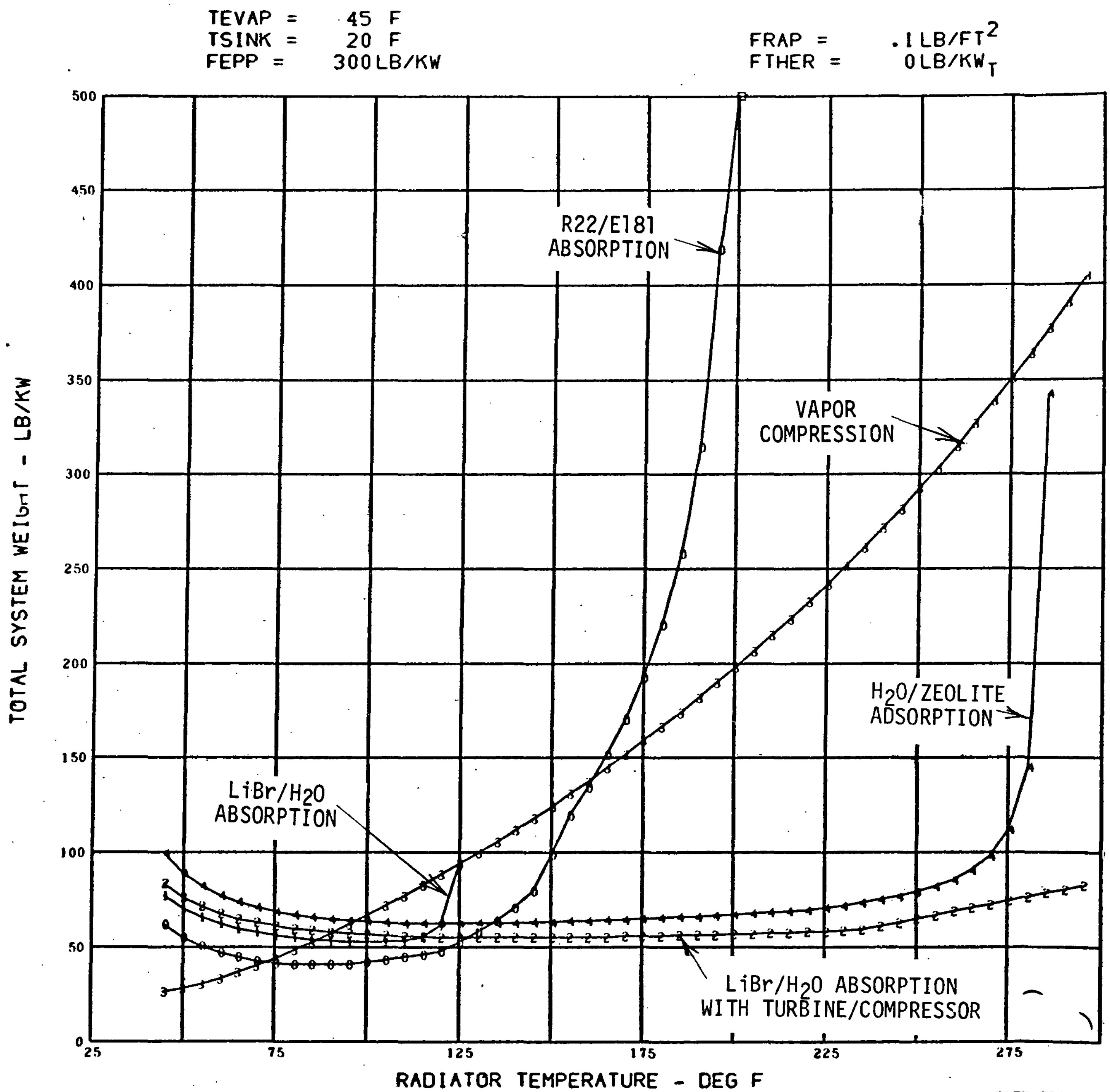

TEVAP = 45 F
TSINK = 20 F
FEPP = 300 LB/KW
FRAP = .1 LB/FT2
FTHER = 0 LB/KWT
R22/E181 ABSORPTION
VAPOR COMPRESSION
H2O/ZEOLITE ADSORPTION
LiBr/H2O ABSORPTION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEIGHT - LB/KW
RADIATOR TEMPERATURE - DEG F
500
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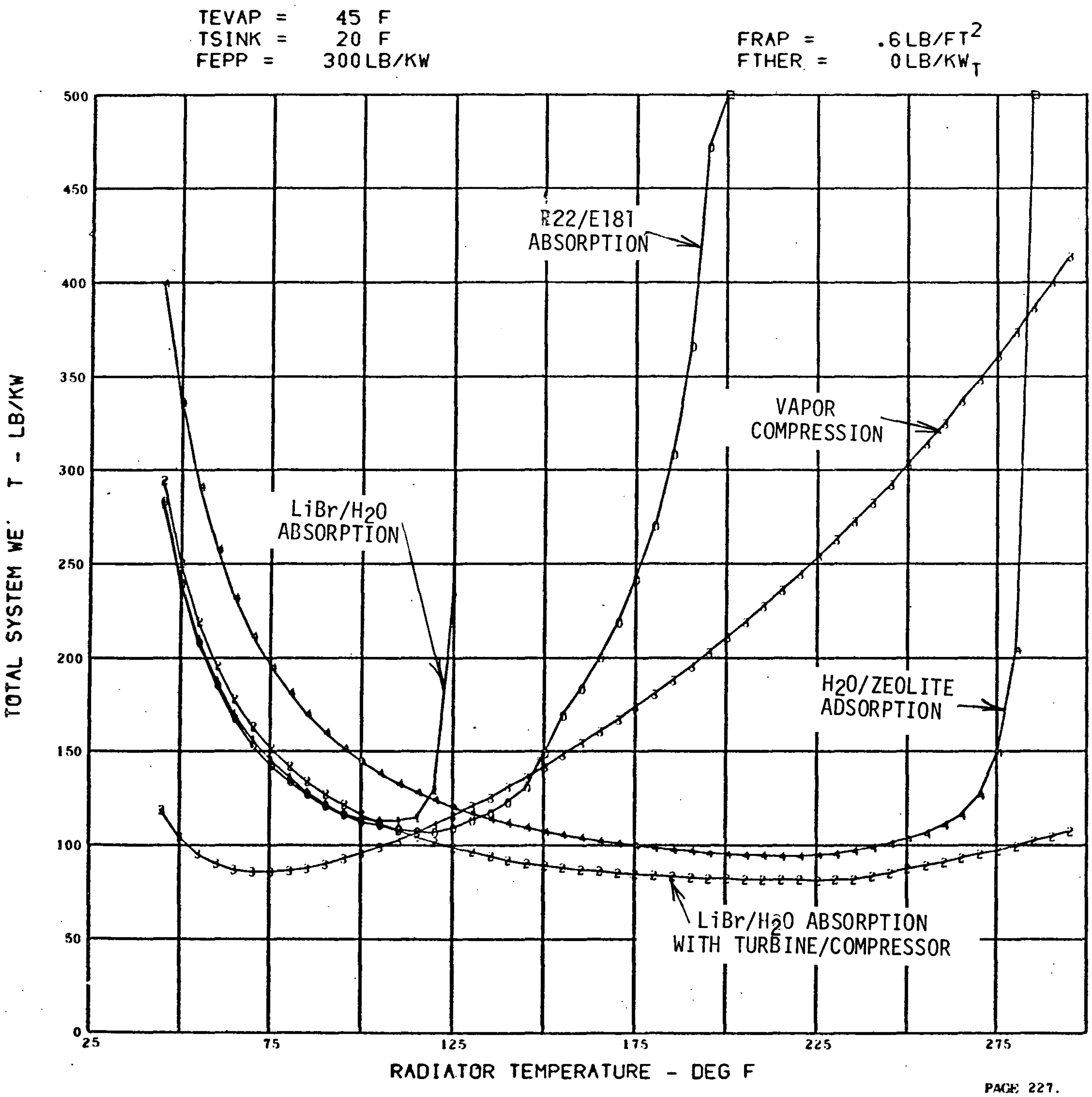

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FEPP = 300 LB/KW
FRAP = .6 LB/FT2
FTHER = 0 LB/KWT
R22/E181 ABSORPTION
VAPOR COMPRESSION
LiBr/H2O ABSORPTION
H2O/ZEOLITE ADSORPTION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEIGHT - LB/KW
RADIATOR TEMPERATURE - DEG F
500
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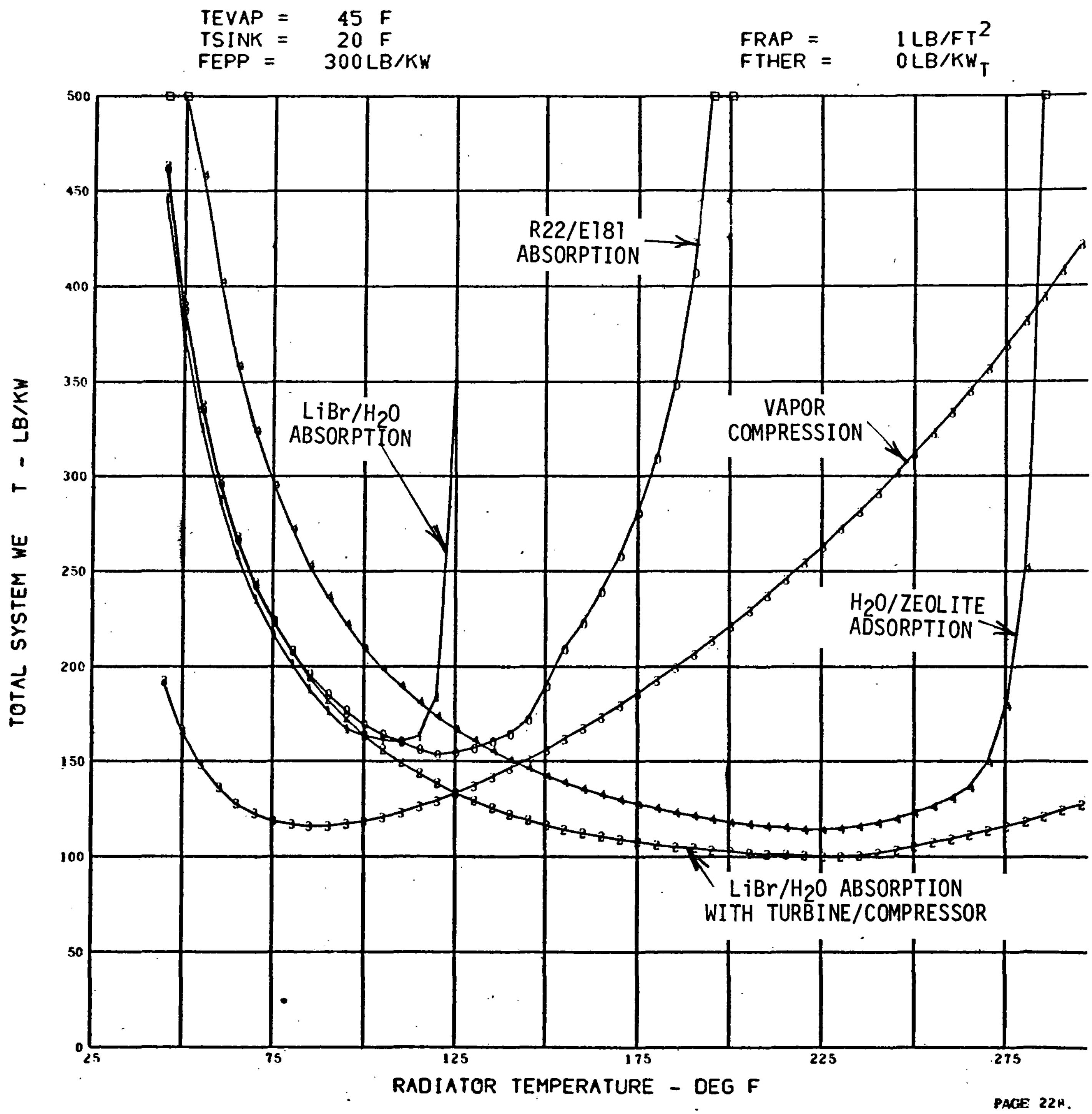
PAGE 227.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FEPP = 300 LB/KW
FRAP = 1 LB/FT²
FTHER = 0 LB/KWT
R22/E181 ABSORPTION
LiBr/H2O ABSORPTION
VAPOR COMPRESSION
H2O/ZEOLITE ADSORPTION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WE T - LB/KW
RADIATOR TEMPERATURE - DEG F
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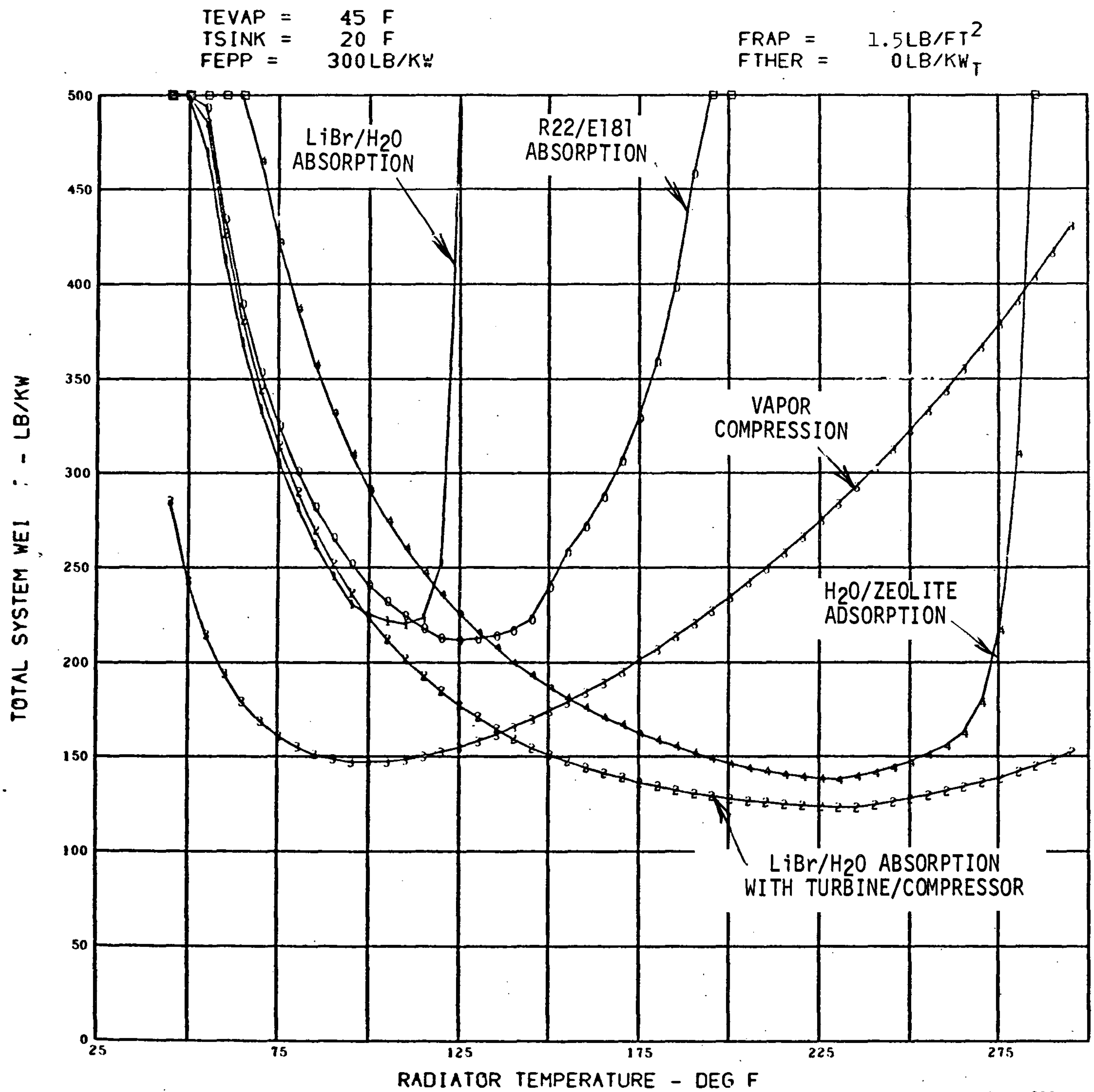

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FRAP = 1.5 LB/FT2
FTHER = 0 LB/KWT
LiBr/H2O ABSORPTION
R22/E181 ABSORPTION
VAPOR COMPRESSION
H2O/ZEOLITE ADSORPTION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEI - LB/KW
RADIATOR TEMPERATURE - DEG F
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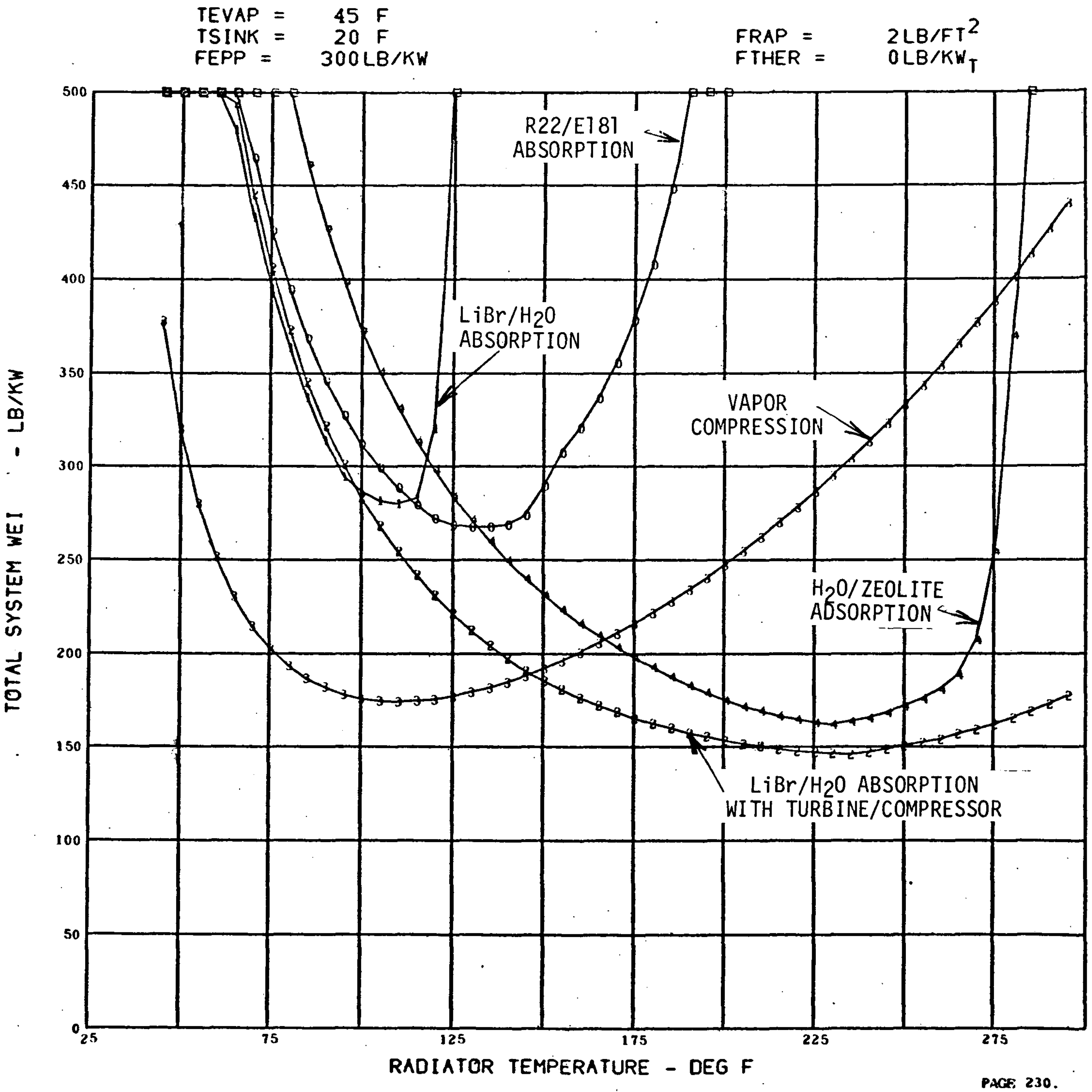
PAGE 229.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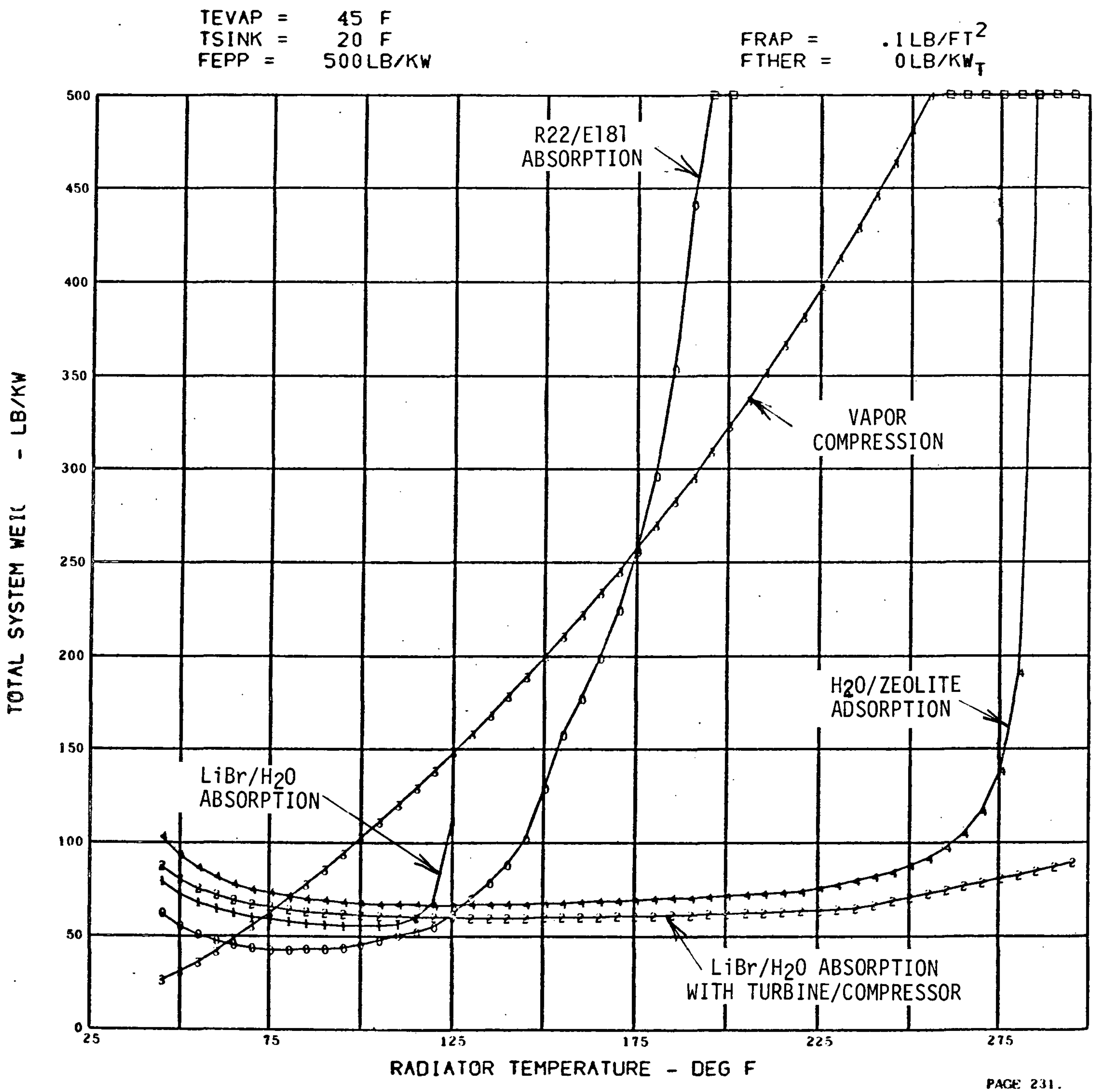
FRAP = 2LB/FT2
FTHER = 0LB/KWT
R22/E181 ABSORPTION
LiBr/H2O ABSORPTION
VAPOR COMPRESSION
H2O/ZEOLITE ADSORPTION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEI - LB/KW
RADIATOR TEMPERATURE - DEG F
PAGE 230.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FRAP = .1LB/FT2
FTHER = 0LB/KWT
R22/E181 ABSORPTION
VAPOR COMPRESSION
H2O/ZEOLITE ADSORPTION
LiBr/H2O ABSORPTION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEIC - LB/KW
RADIATOR TEMPERATURE - DEG F
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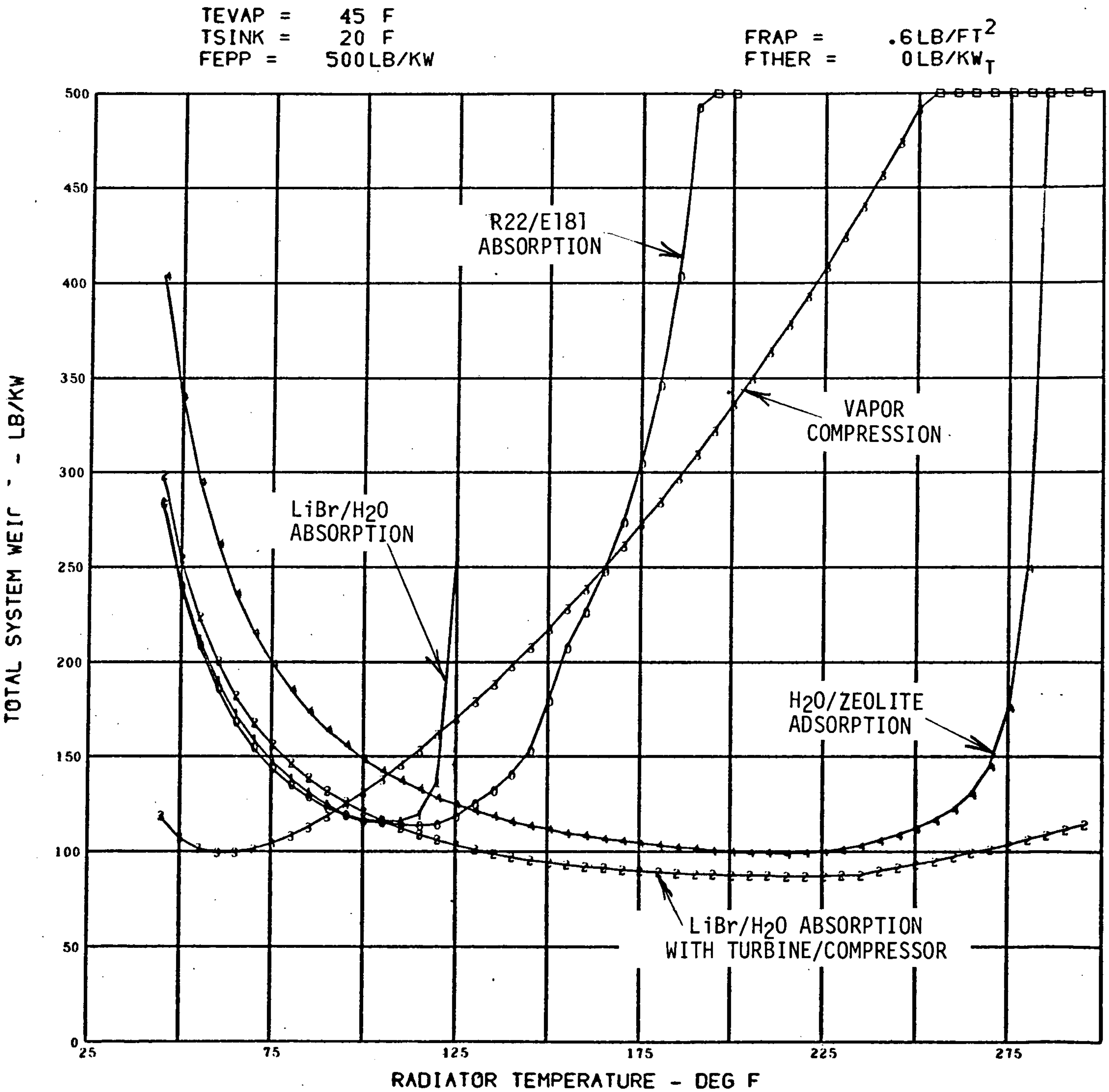
PAGE 231.



TEVAP = 45 F
TSINK = 20 F
FEPP = 500 LB/KW
FRAP = .6 LB/FT2
FTHER = 0 LB/KWT
R22/E181 ABSORPTION
VAPOR COMPRESSION
LiBr/H2O ABSORPTION
H2O/ZEOLITE ADSORPTION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEIGHT - LB/KW
RADIATOR TEMPERATURE - DEG F
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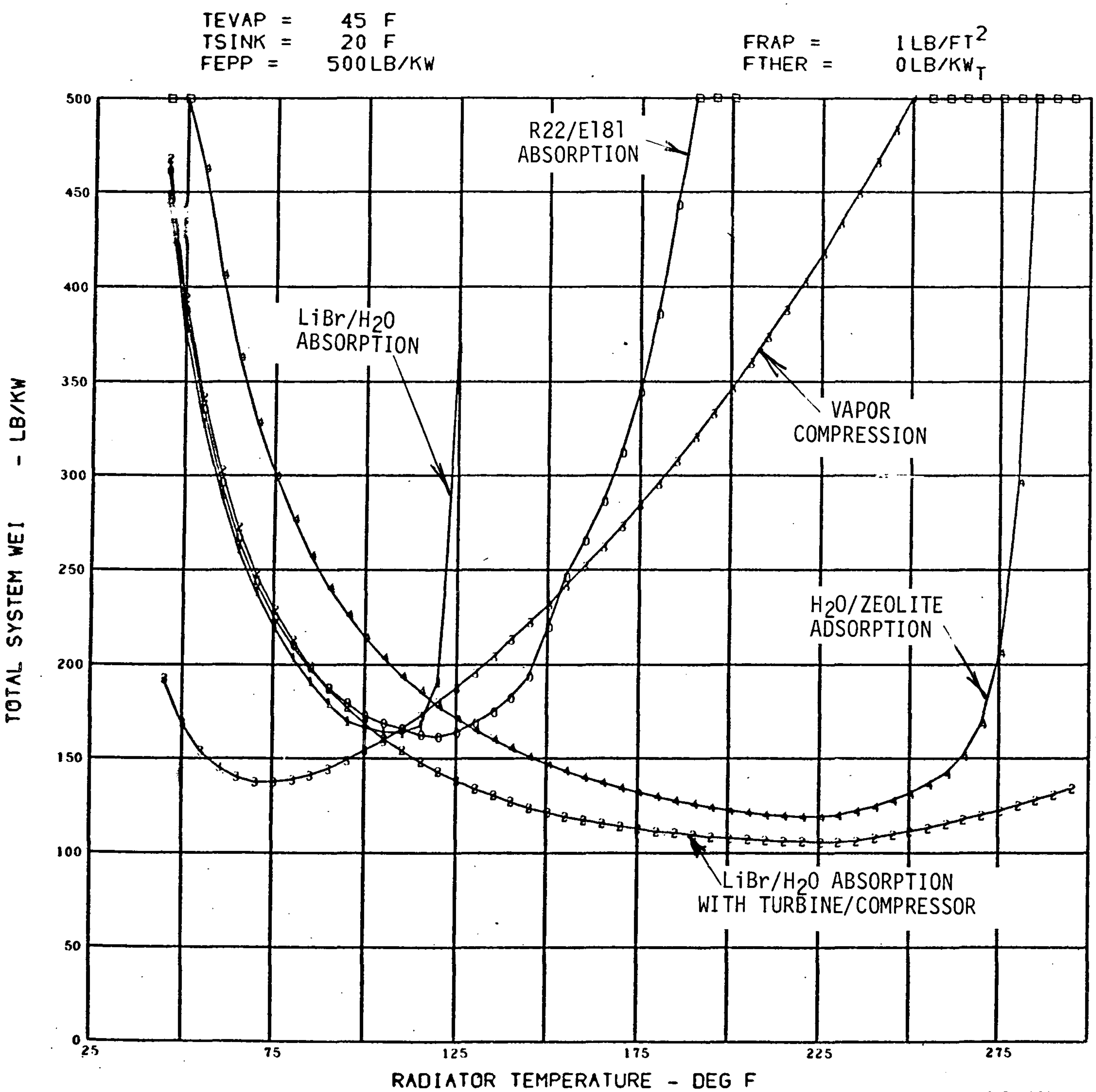
PAGE 232



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TSINK = 20 F
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FRAP = 1LB/FT2
FTHER = 0LB/KWT
R22/E181 ABSORPTION
LiBr/H2O ABSORPTION
VAPOR COMPRESSION
H2O/ZEOLITE ADSORPTION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEI - LB/KW
RADIATOR TEMPERATURE - DEG F
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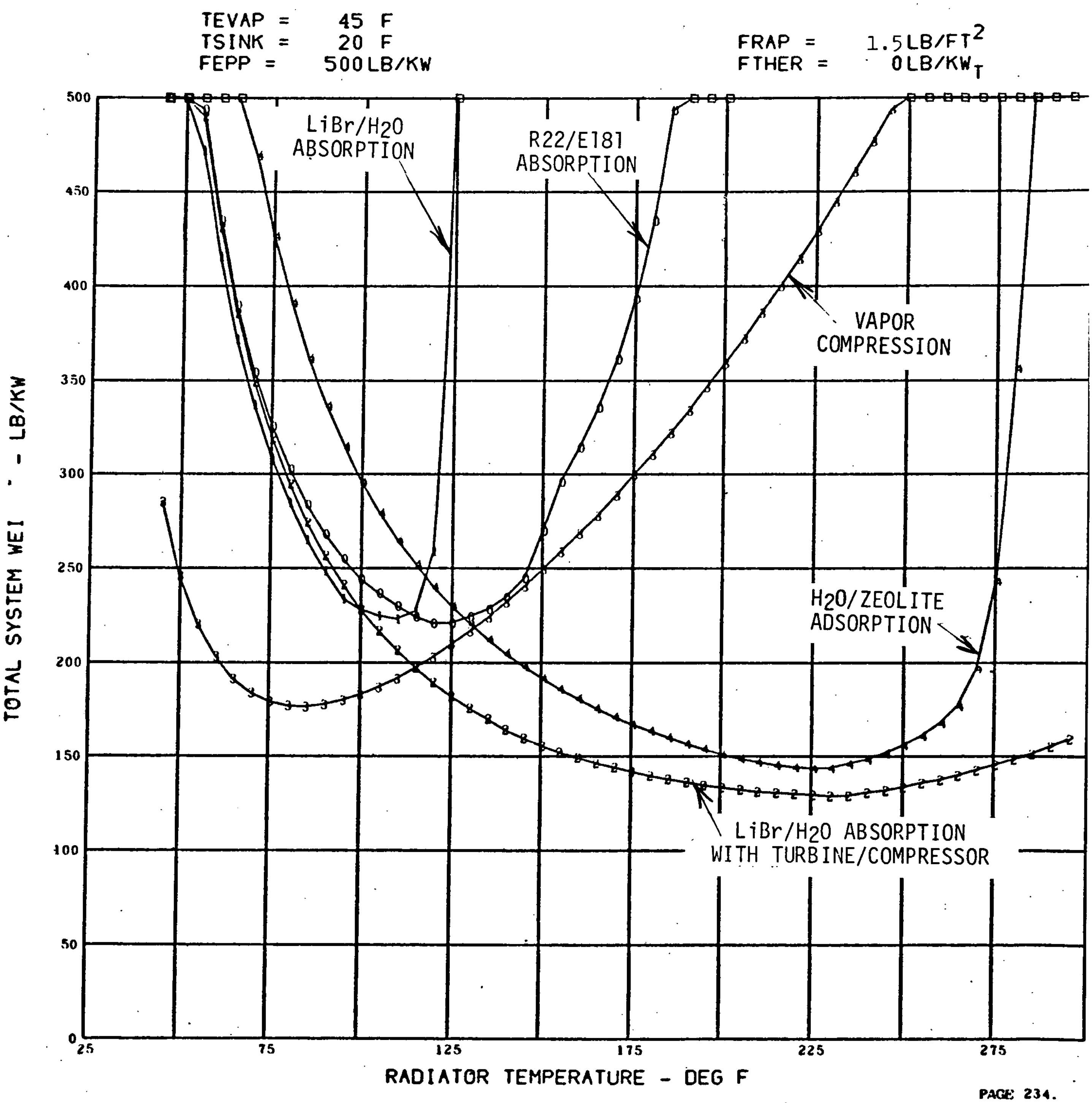

TEVAP = 45 F
TSINK = 20 F
FEPP = 500LB/KW
FRAP = 1.5LB/FT2
FTHER = 0LB/KWT
LiBr/H2O ABSORPTION
R22/E181 ABSORPTION
VAPOR COMPRESSION
H2O/ZEOLITE ADSORPTION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEI ' - LB/KW
RADIATOR TEMPERATURE - DEG F
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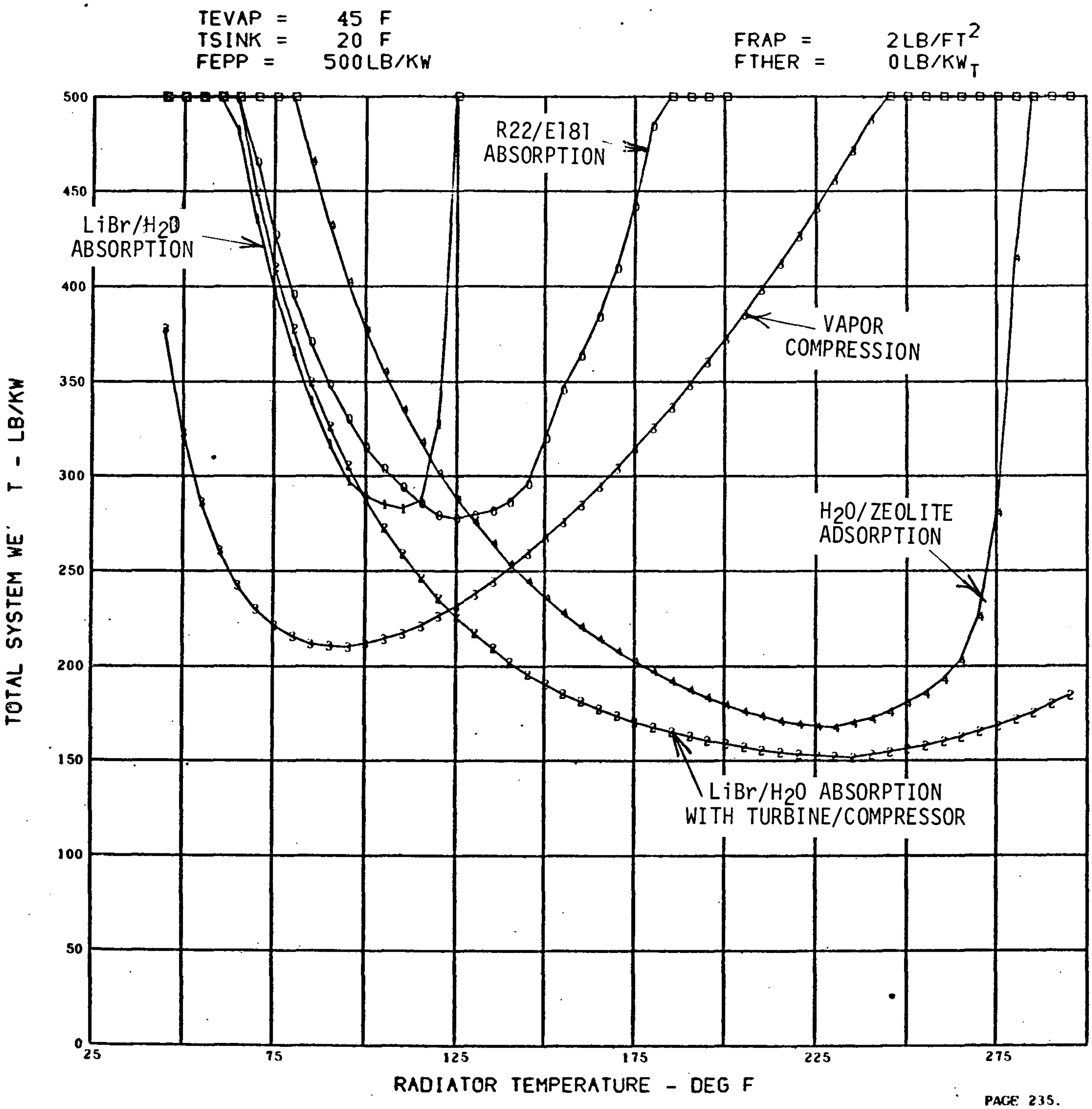
PAGE 234.


TEVAP = 45 F
TSINK = 20 F
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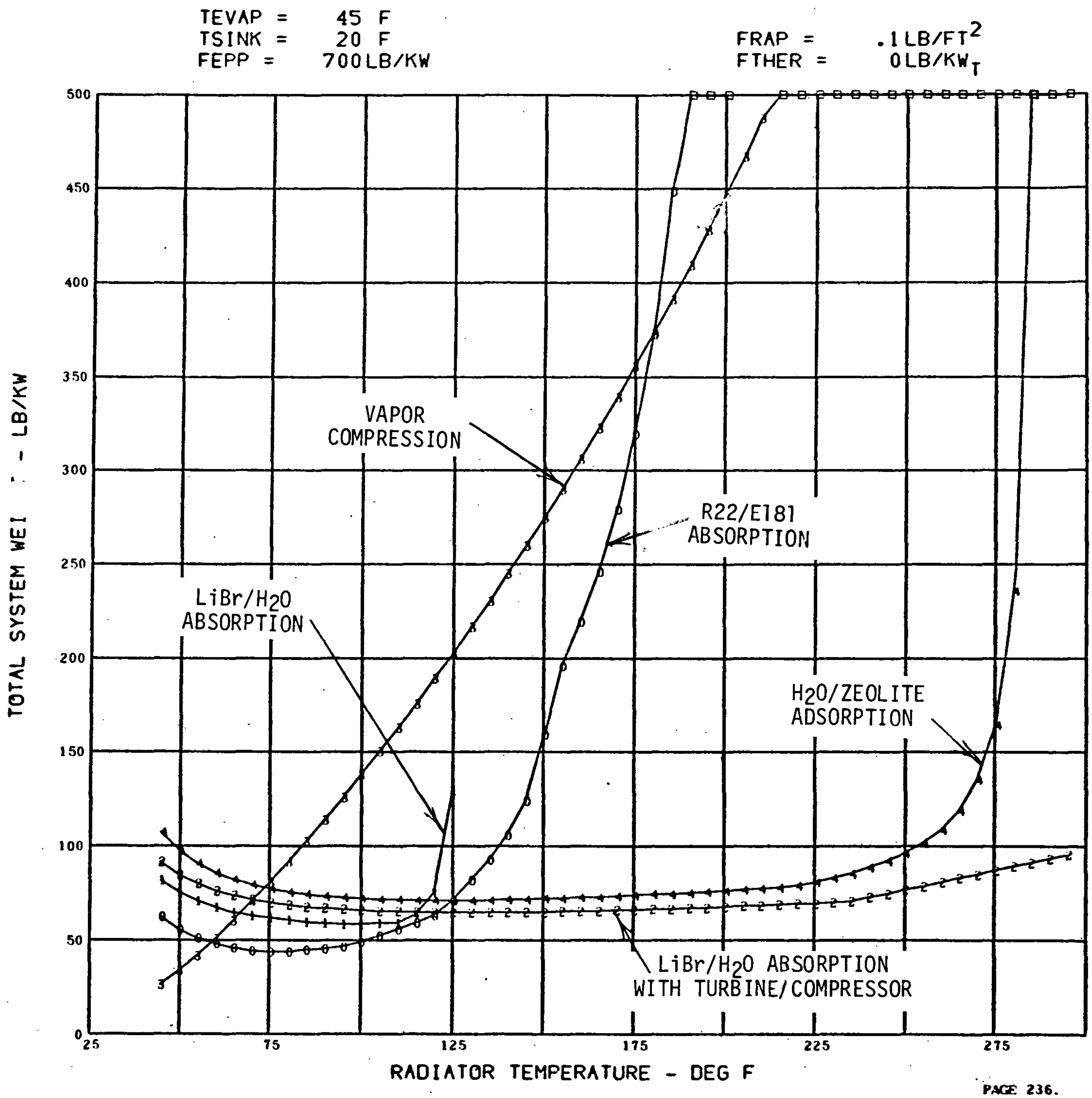
FRAP = 2LB/FT^2
FTHER = 0LB/KW$_T$
R22/E181 ABSORPTION
LiBr/H_2O ABSORPTION
VAPOR COMPRESSION
H_2O/ZEOLITE ADSORPTION
LiBr/H_2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WE T - LB/KW
RADIATOR TEMPERATURE - DEG F
PAGE 235.


TEVAP = 45 F
TSINK = 20 F
FEPP = 700 LB/KW
FRAP = .1 LB/FT²
FTHER = 0 LB/KW_T
VAPOR COMPRESSION
R22/E181 ABSORPTION
LiBr/H2O ABSORPTION
H2O/ZEOLITE ADSORPTION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEI ' - LB/KW
RADIATOR TEMPERATURE - DEG F
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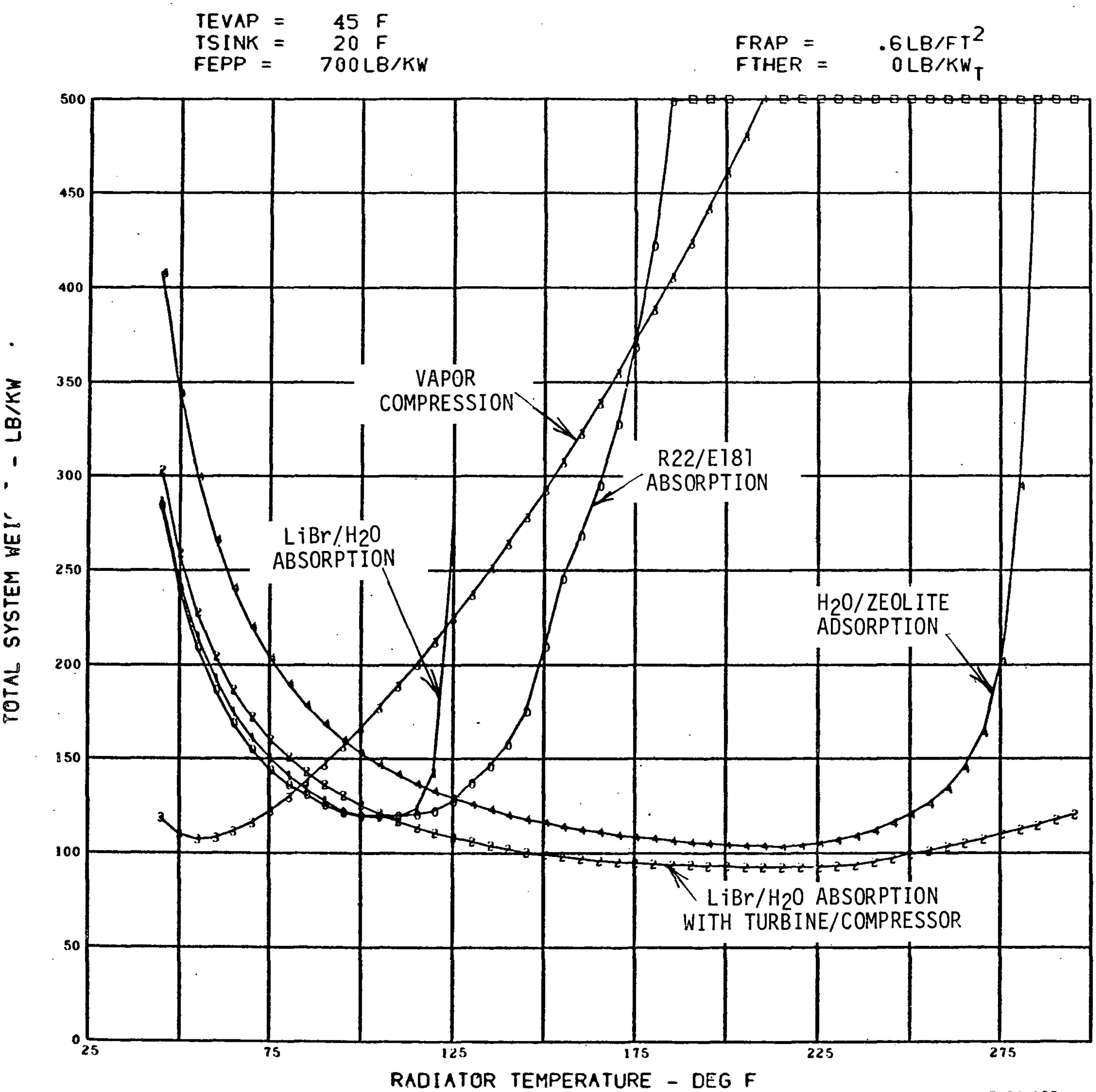
PAGE 236.



TEVAP = 45 F
TSINK = 20 F
FEPP = 700LB/KW
FRAP = .6LB/FT2
FTHER = 0LB/KWT
VAPOR COMPRESSION
R22/E181 ABSORPTION
LiBr/H2O ABSORPTION
H2O/ZEOLITE ADSORPTION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEI' - LB/KW
RADIATOR TEMPERATURE - DEG F
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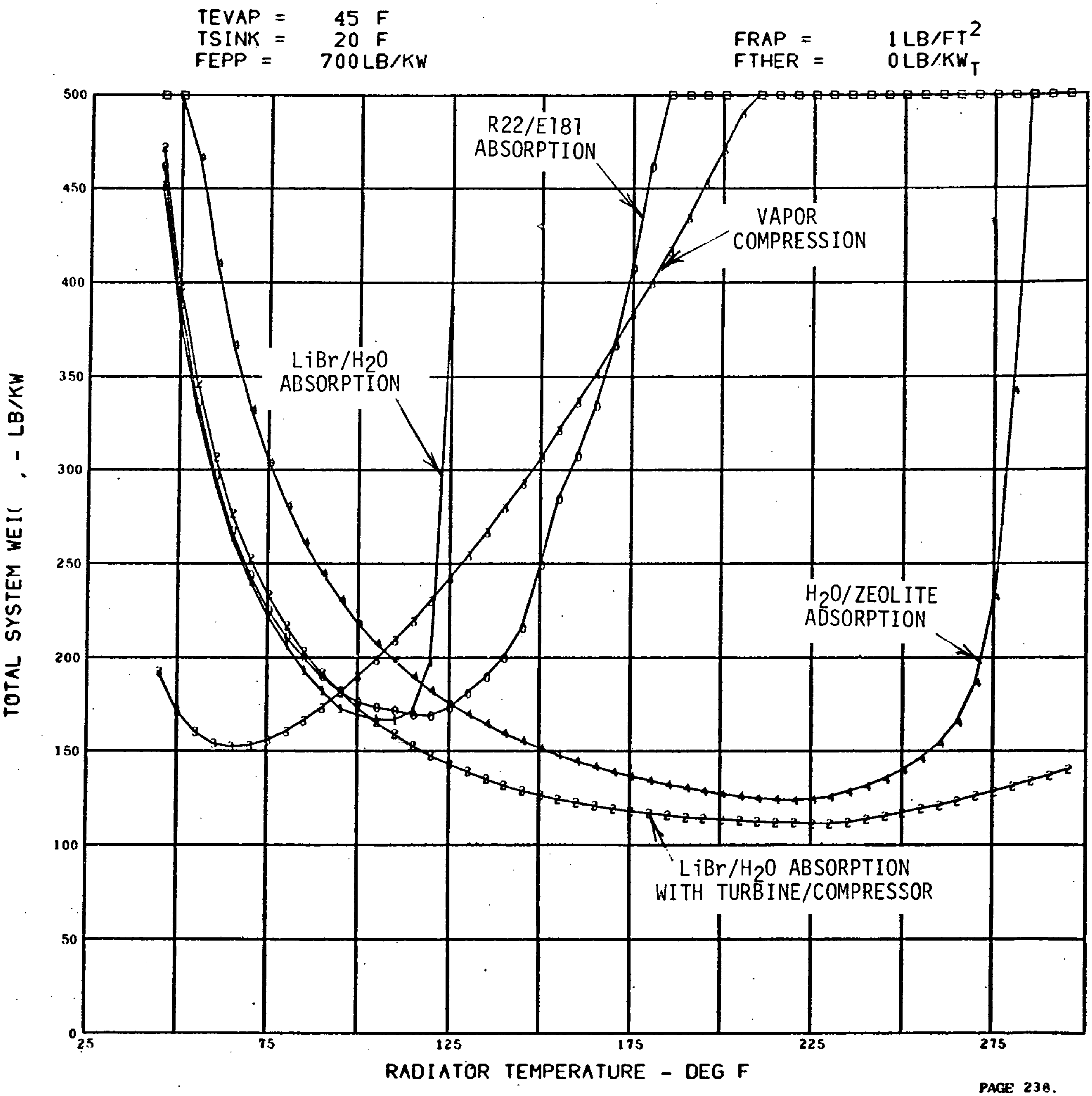
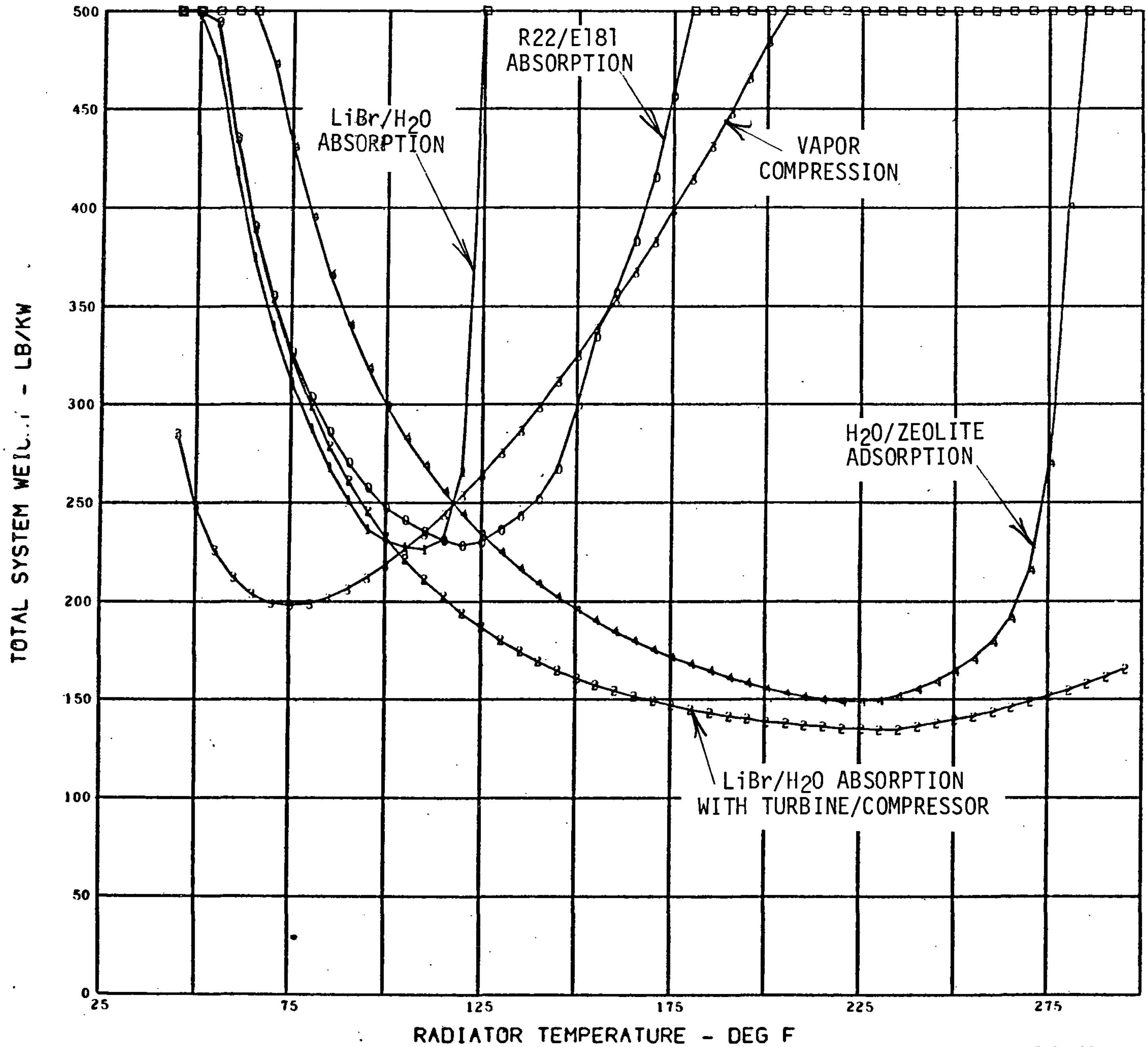

TEVAP = 45 F
TSINK = 20 F
FEPP = 700 LB/KW
FRAP = 1 LB/FT^2
FTHER = 0 LB/KW$_T$
R22/E181 ABSORPTION
VAPOR COMPRESSION
LiBr/H_2O ABSORPTION
H_2O/ZEOLITE ADSORPTION
LiBr/H_2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEI(, - LB/KW
RADIATOR TEMPERATURE - DEG F
PAGE 238.

TEVAP = 45 F
TSINK = 20 F
FEPP = 700 LB/KW

FRAP = 1.5 LB/FT^2
FTHER = 0 LB/KW$_T$



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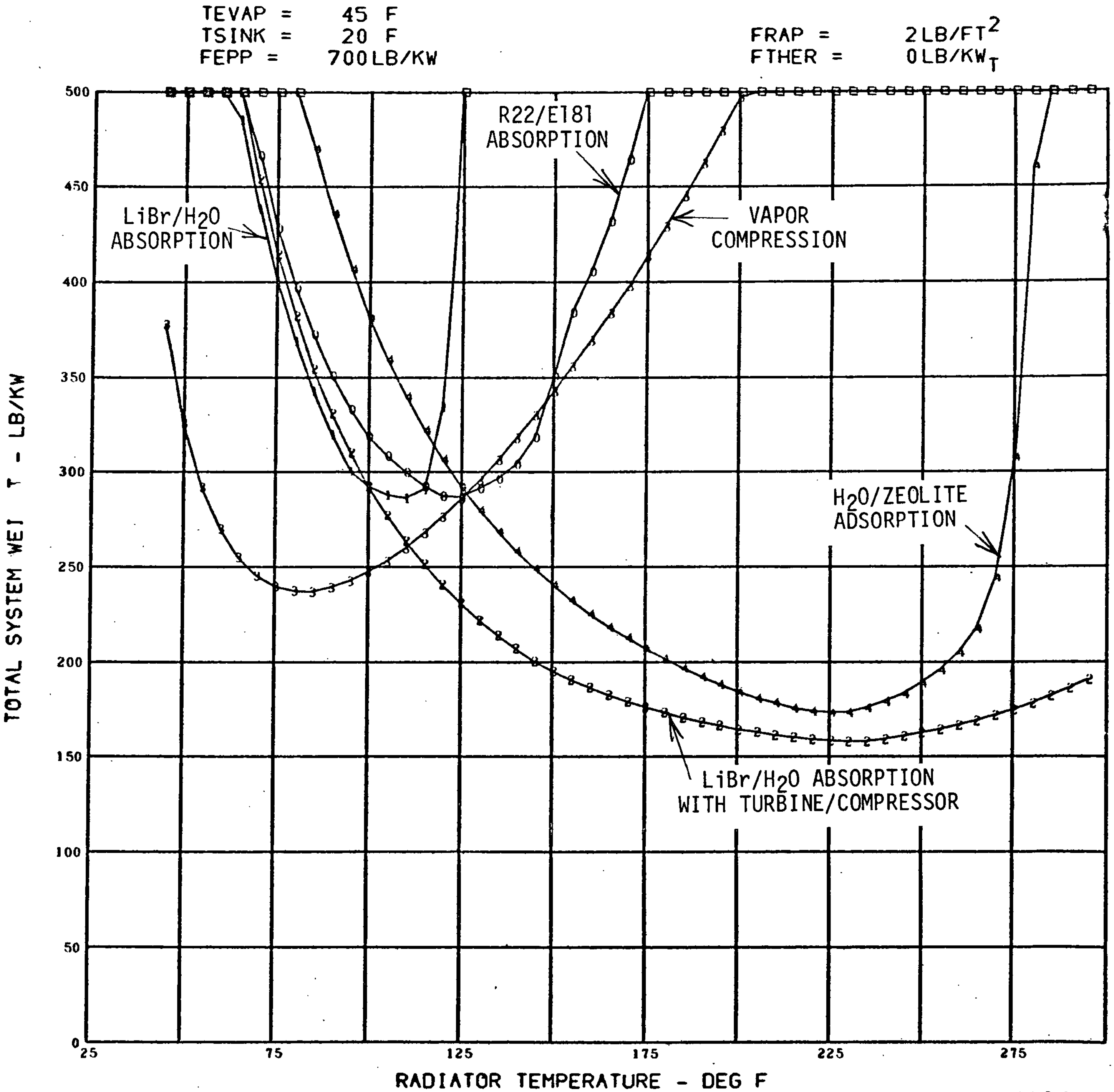
TEVAP = 45 F
TSINK = 20 F
FEPP = 700LB/KW
FRAP = 2LB/FT2
FTHER = 0LB/KWT
R22/E181 ABSORPTION
LiBr/H2O ABSORPTION
VAPOR COMPRESSION
H2O/ZEOLITE ADSORPTION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEIGHT - LB/KW
RADIATOR TEMPERATURE - DEG F
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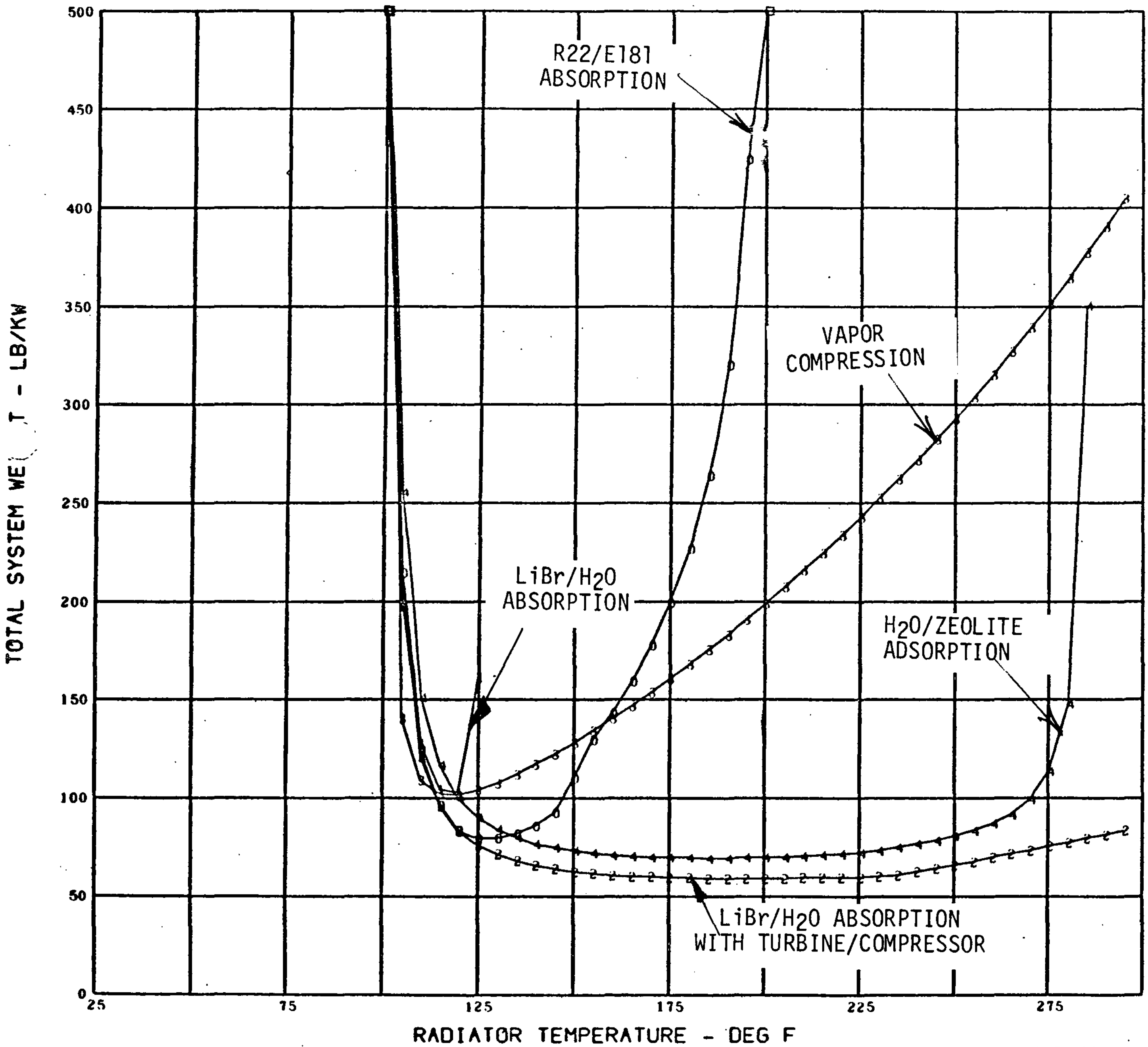
TEVAP = 45 F
TSINK = 100 F
FEPP = 300 LB/KW

FRAP = .1 LB/FT^2
FTHER = 0 LB/KW$_T$



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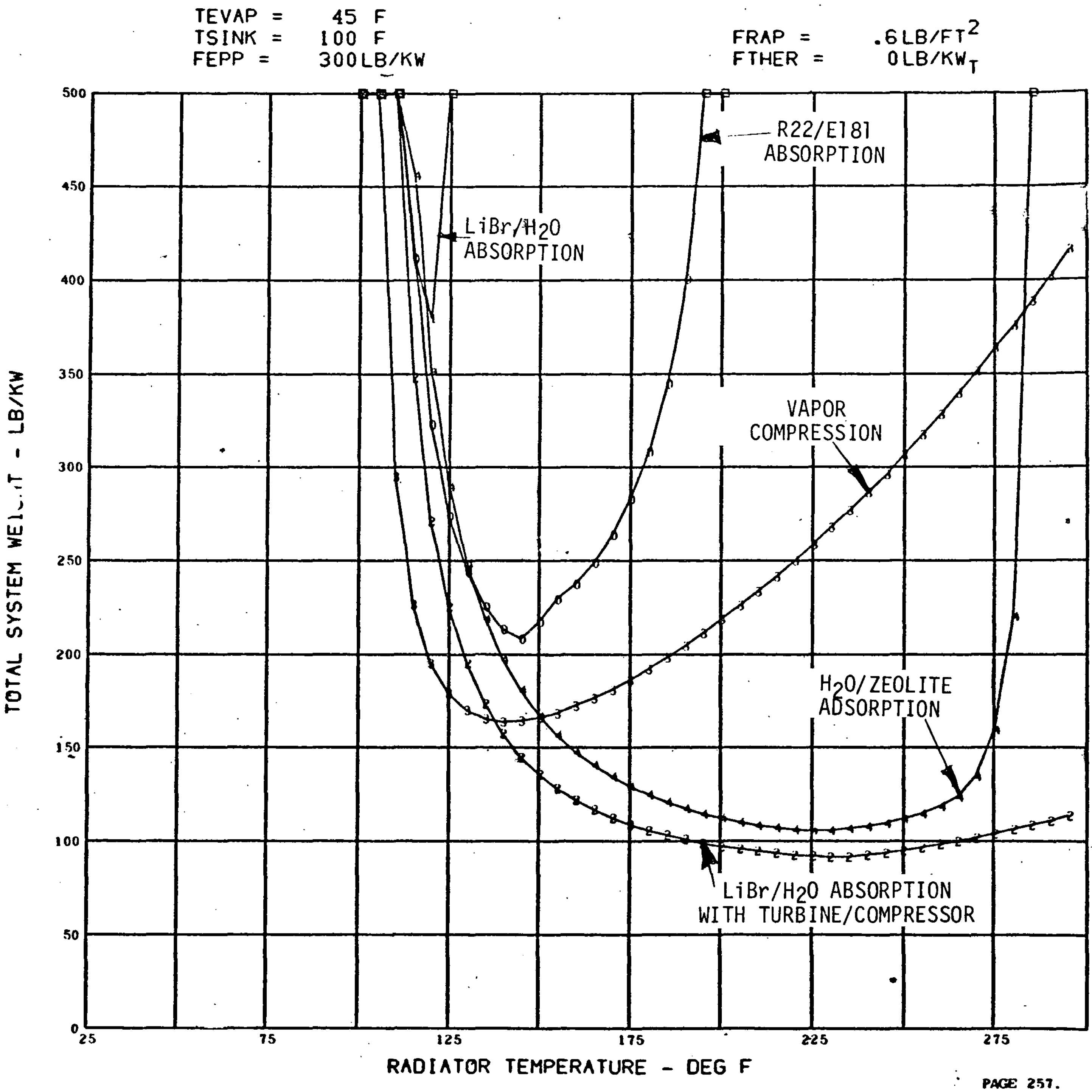

TEVAP = 45 F
TSINK = 100 F
FEPP = 300 LB/KW
FRAP = .6 LB/FT2
FTHER = 0 LB/KWT
R22/E181 ABSORPTION
LiBr/H2O ABSORPTION
VAPOR COMPRESSION
H2O/ZEOLITE ADSORPTION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEIGHT - LB/KW
RADIATOR TEMPERATURE - DEG F
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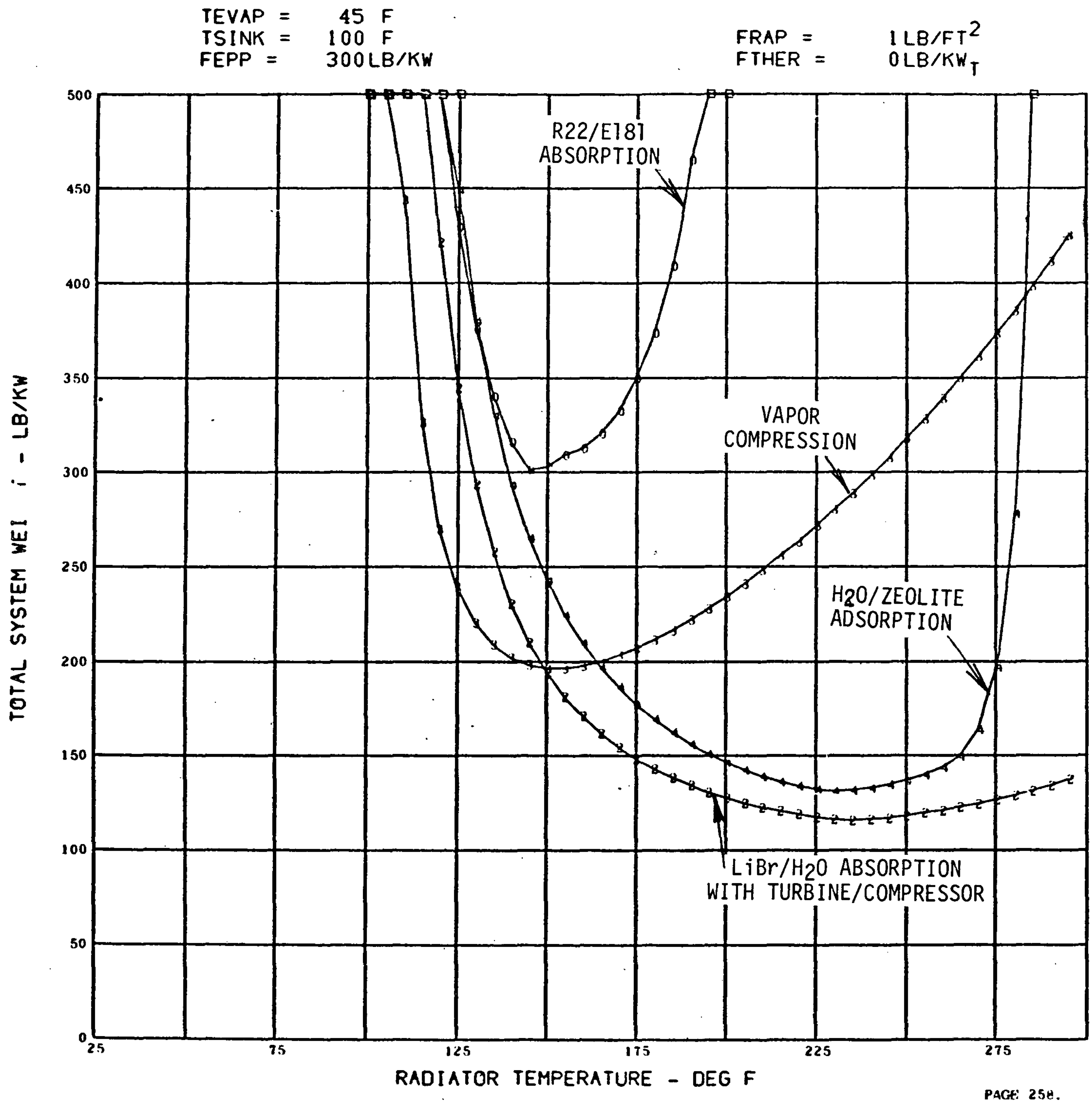

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TSINK = 100 F
FEPP = 300 LB/KW
FRAP = 1 LB/FT2
FTHER = 0 LB/KWT
R22/E181 ABSORPTION
VAPOR COMPRESSION
H2O/ZEOLITE ADSORPTION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEI - LB/KW
RADIATOR TEMPERATURE - DEG F
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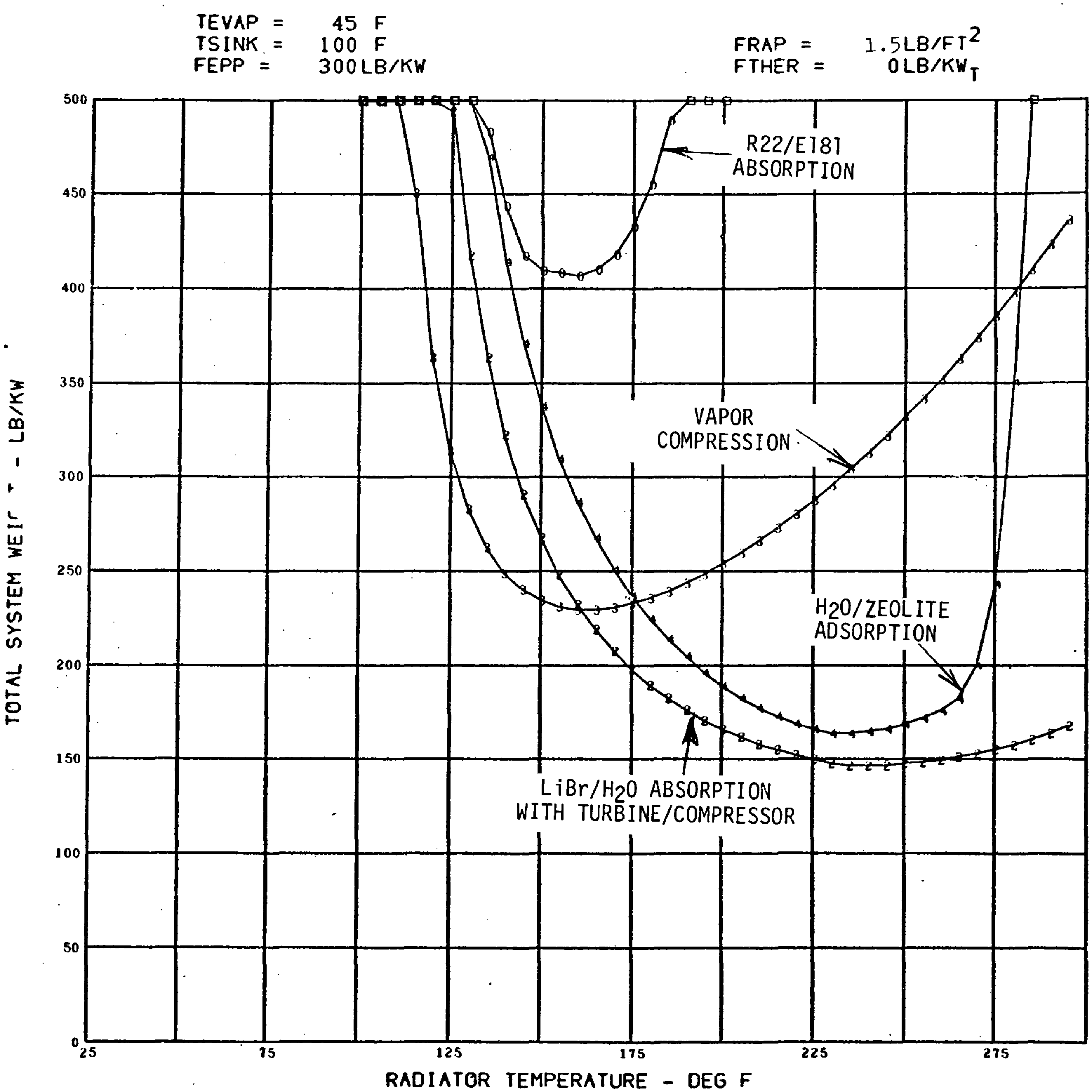

TEVAP = 45 F
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FEPP = 300 LB/KW
FRAP = 1.5 LB/FT2
FTHER = 0 LB/KWT
R22/E181 ABSORPTION
VAPOR COMPRESSION
H2O/ZEOLITE ADSORPTION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEIGHT - LB/KW
RADIATOR TEMPERATURE - DEG F
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TEVAP = 45 F
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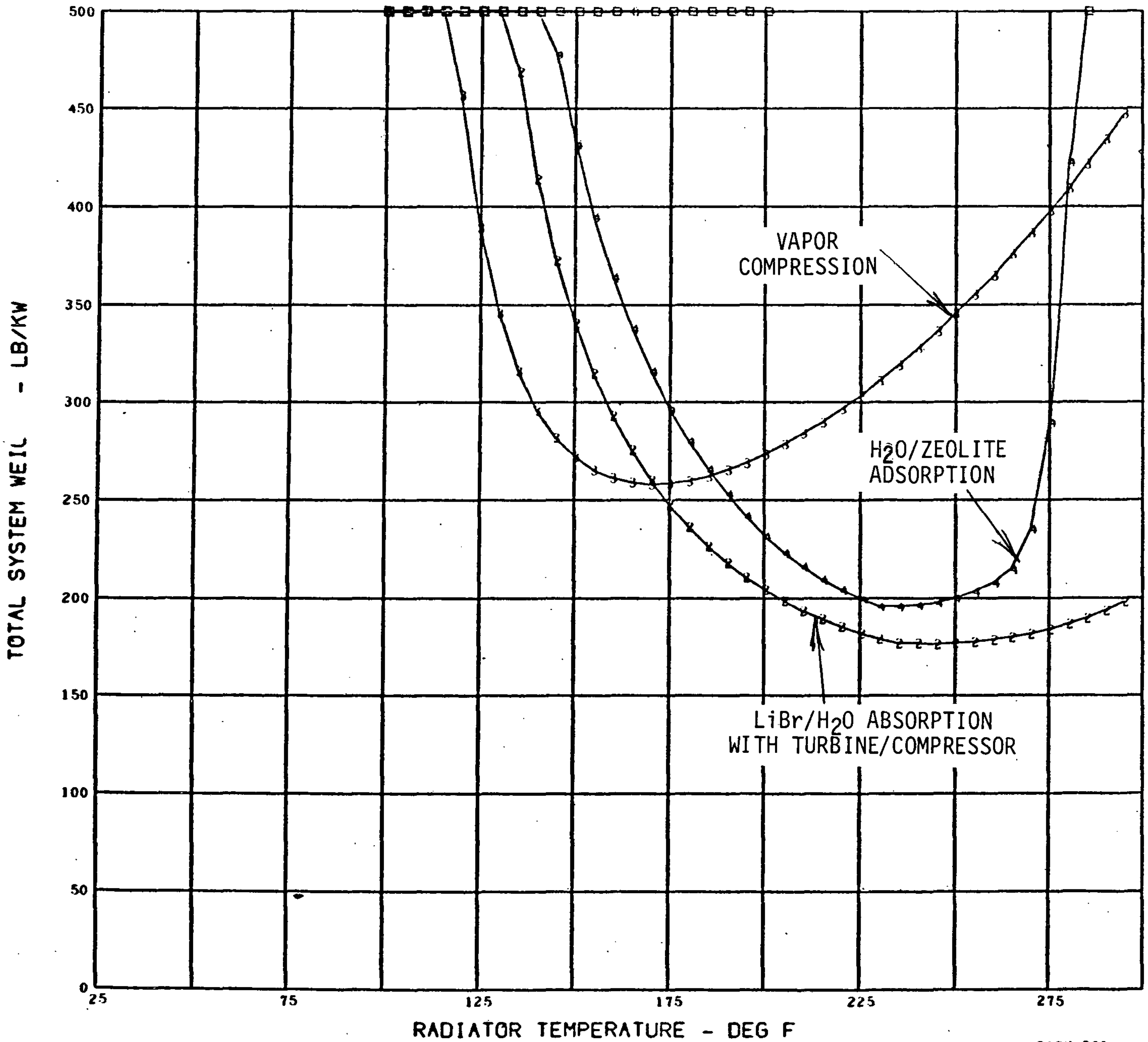
FRAP = 2LB/FT^2
FTHER = 0LB/KW$_T$



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FEPP = 500 LB/KW

FRAP = .1 LB/FT^2
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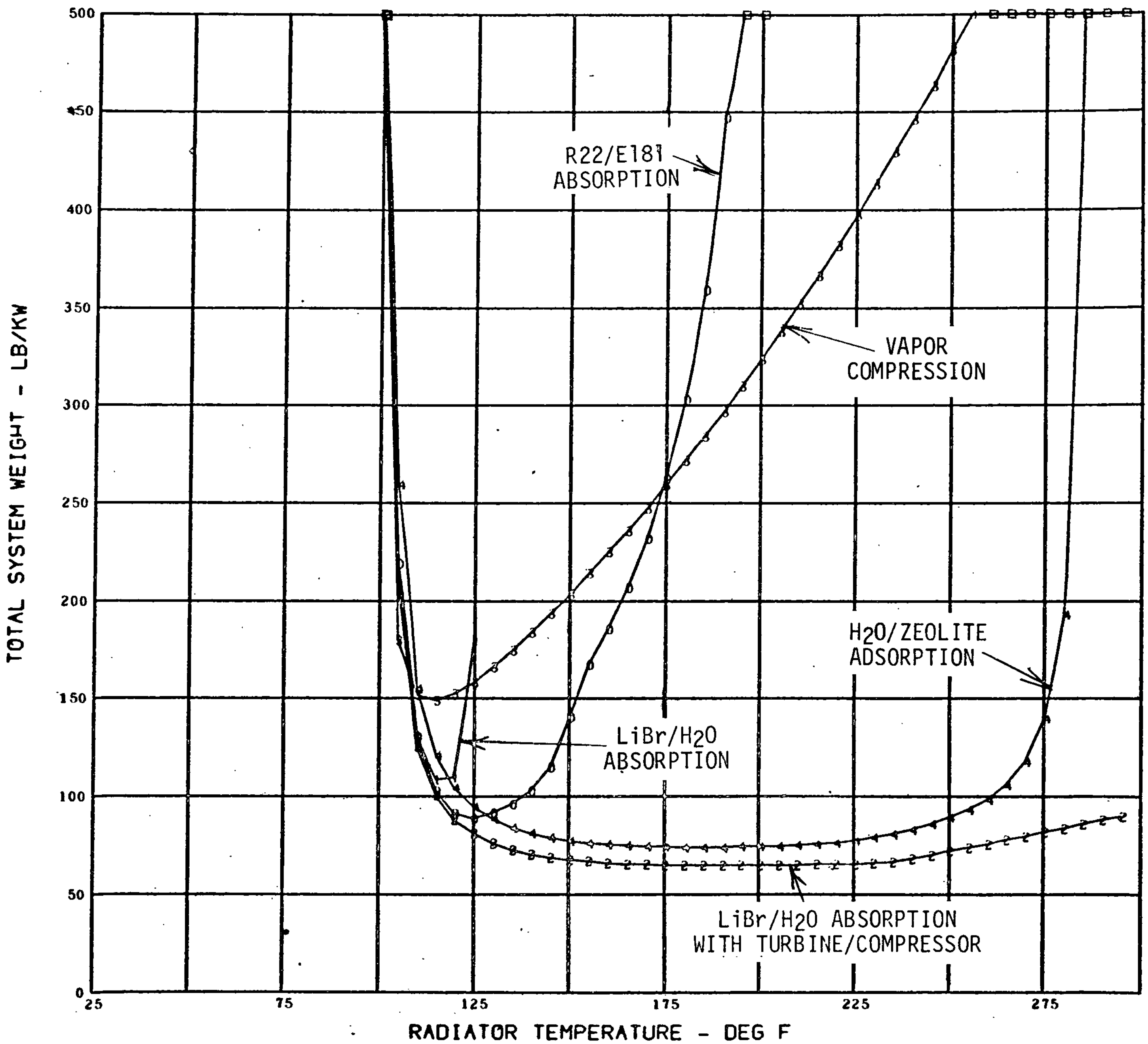

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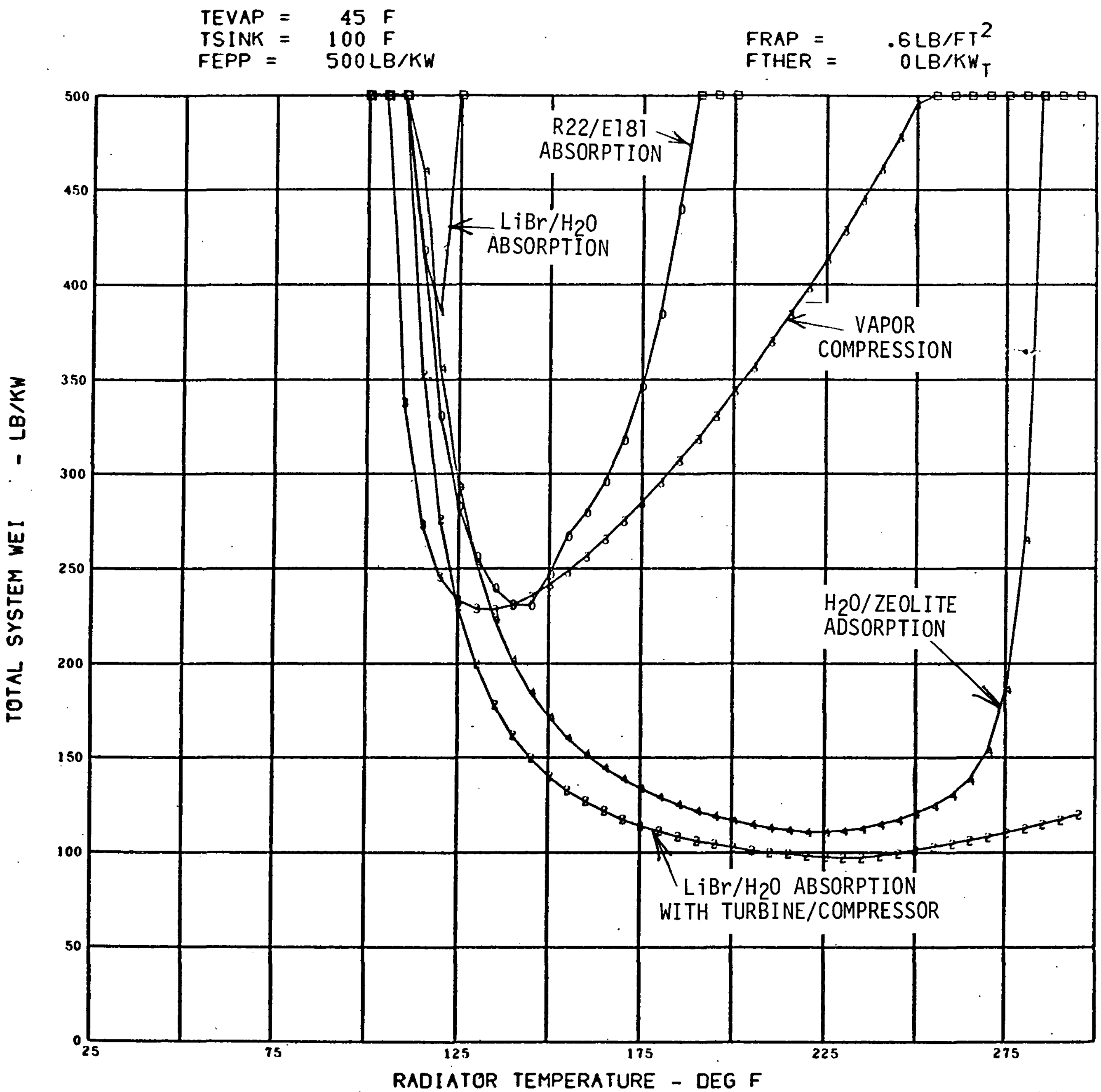
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FRAP = .6LB/FT2
FTHER = 0LB/KWT
R22/E181 ABSORPTION
LiBr/H2O ABSORPTION
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TOTAL SYSTEM WEI - LB/KW
RADIATOR TEMPERATURE - DEG F
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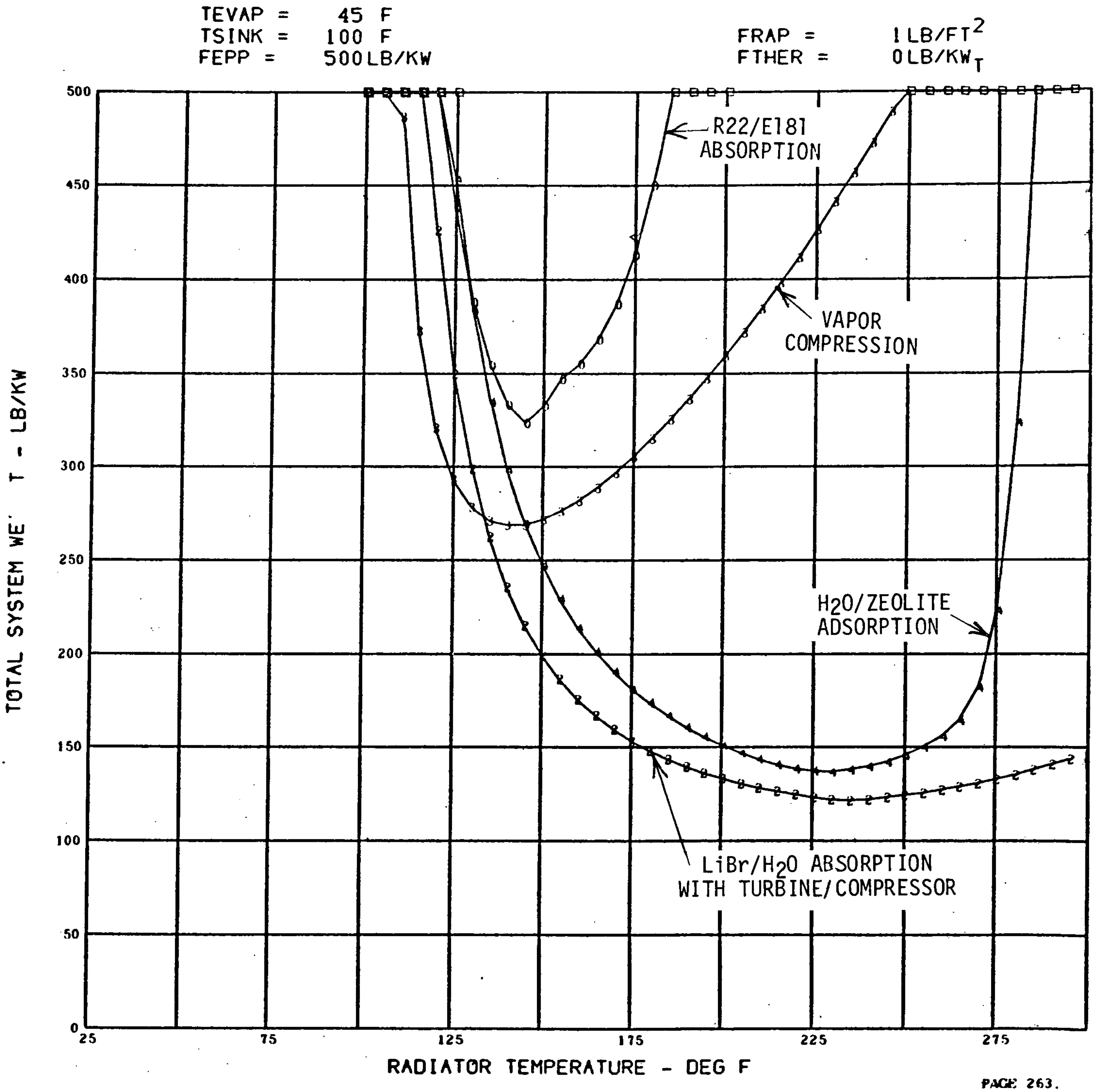
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FEPP = 500 LB/KW
FRAP = 1 LB/FT2
FTHER = 0 LB/KWT
R22/E181 ABSORPTION
VAPOR COMPRESSION
H2O/ZEOLITE ADSORPTION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WE T - LB/KW
RADIATOR TEMPERATURE - DEG F
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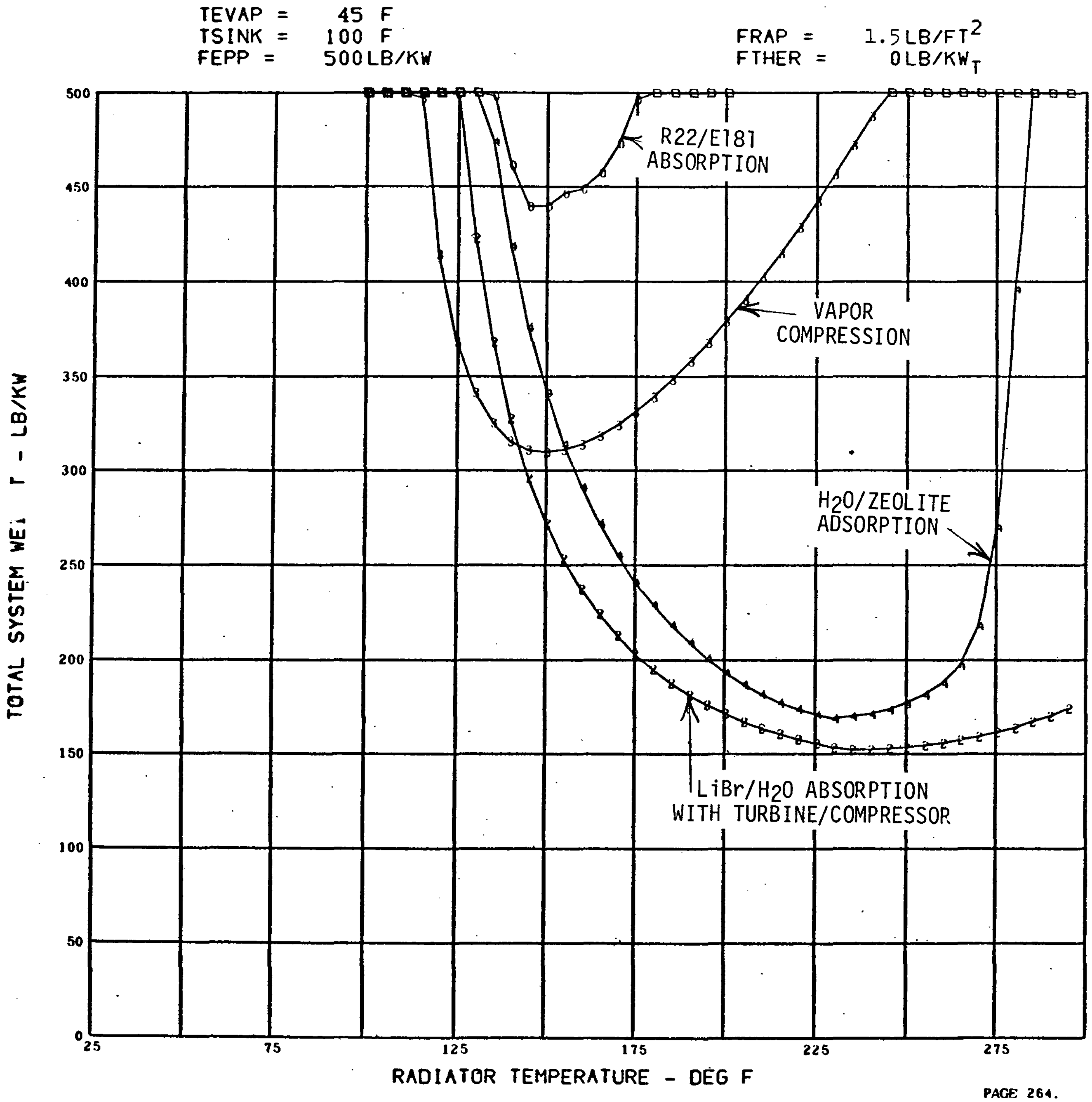
PAGE 263.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R22/E181 ABSORPTION
VAPOR COMPRESSION
H2O/ZEOLITE ADSORPTION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEI T - LB/KW
RADIATOR TEMPERATURE - DEG F
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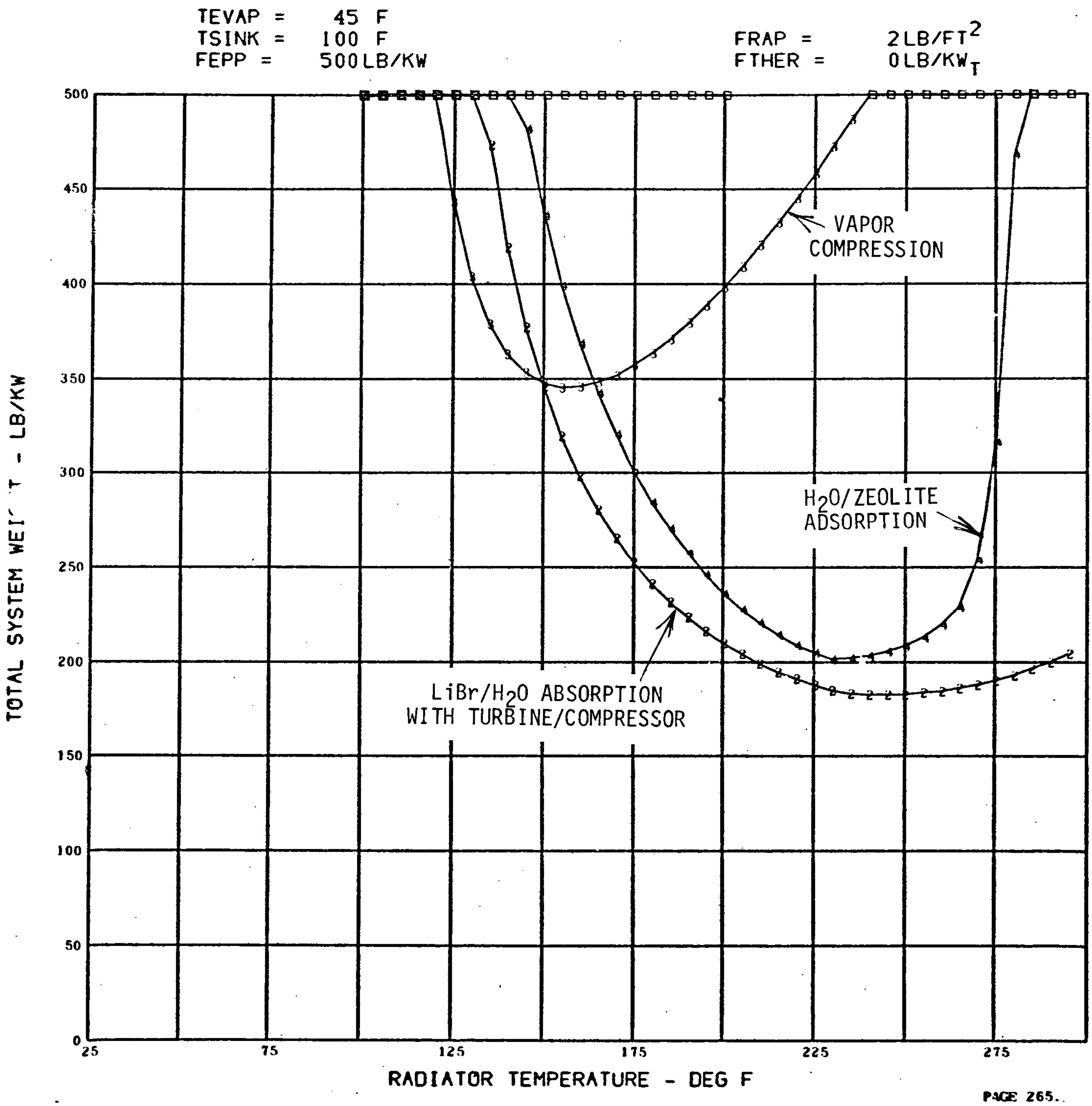
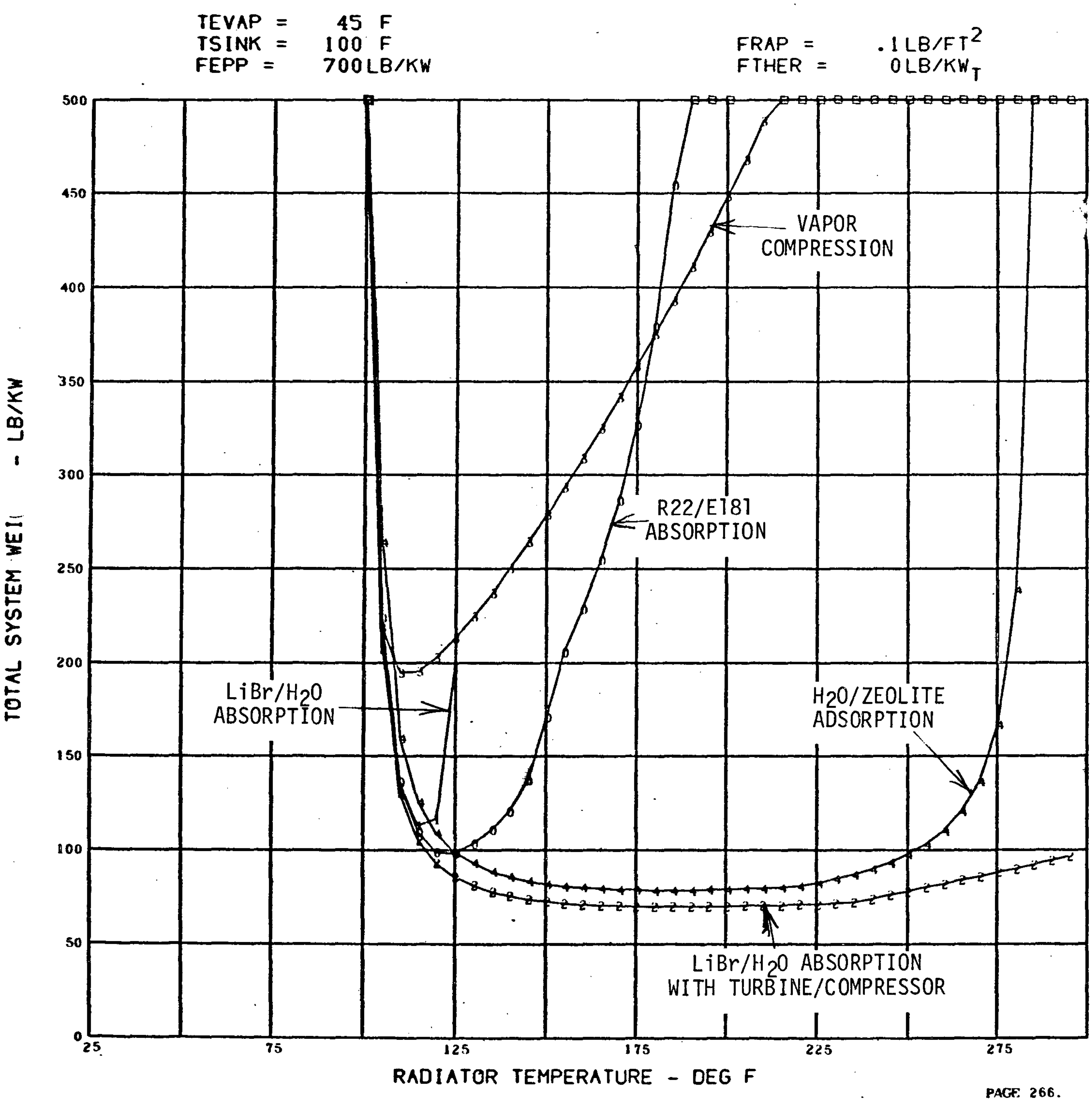
PAGE 264.


TEVAP = 45 F
TSINK = 100 F
FEPP = 500 LB/KW
FRAP = 2 LB/FT2
FTHER = 0 LB/KWT
VAPOR COMPRESSION
H2O/ZEOLITE ADSORPTION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEI' 'T - LB/KW
RADIATOR TEMPERATURE - DEG F
PAGE 265.


TEVAP = 45 F
TSINK = 100 F
FEPP = 700 LB/KW
FRAP = .1 LB/FT2
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VAPOR COMPRESSION
R22/E181 ABSORPTION
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H2O/ZEOLITE ADSORPTION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEI - LB/KW
RADIATOR TEMPERATURE - DEG F
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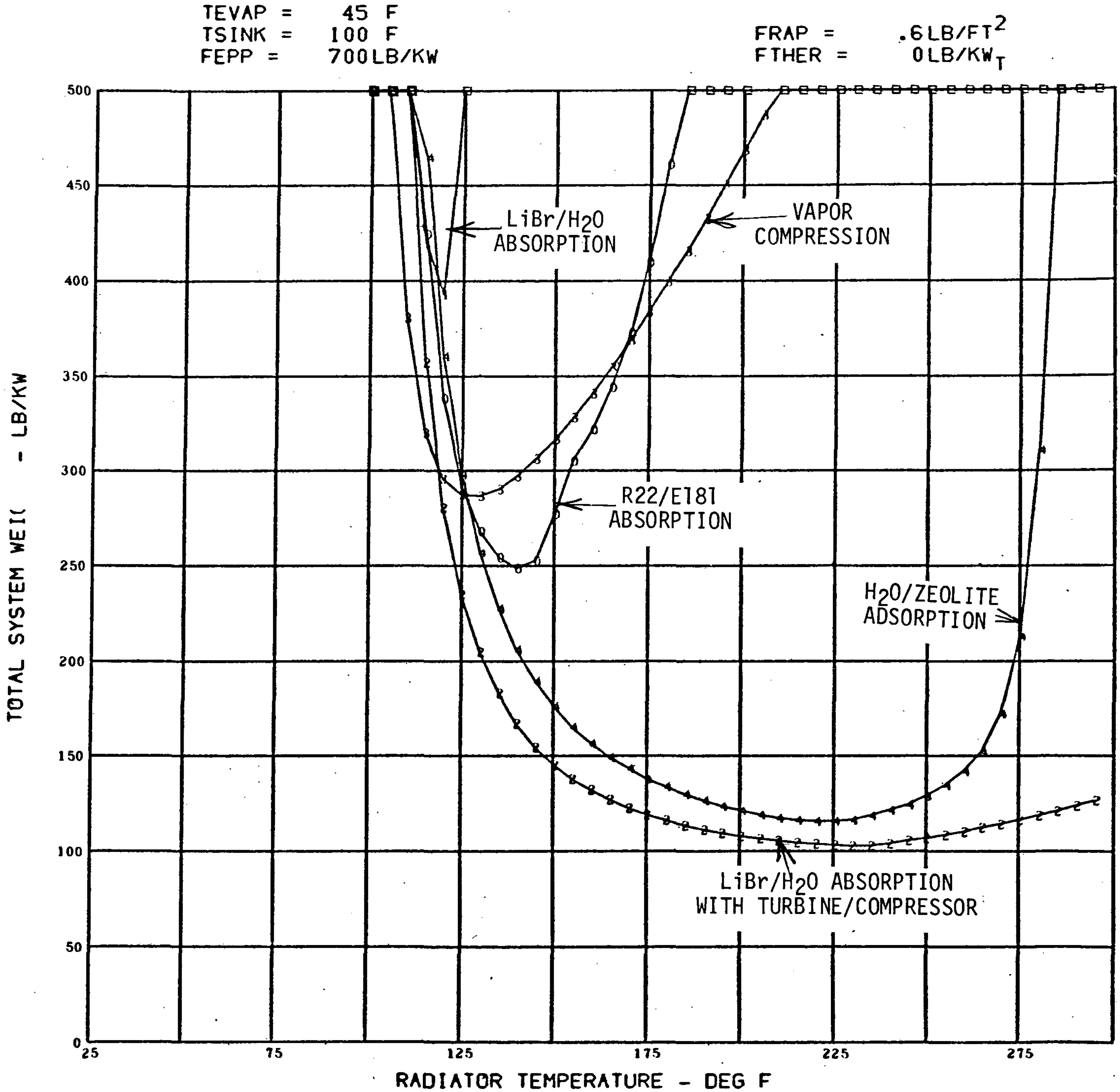

TEVAP = 45 F
TSINK = 100 F
FEPP = 700 LB/KW
FRAP = .6 LB/FT2
FTHER = 0 LB/KWT
LiBr/H2O ABSORPTION
VAPOR COMPRESSION
R22/E181 ABSORPTION
H2O/ZEOLITE ADSORPTION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEI(- LB/KW
RADIATOR TEMPERATURE - DEG F
0
50
100
150
200
250
300
350
400
450
500
25
75
125
175
225
275

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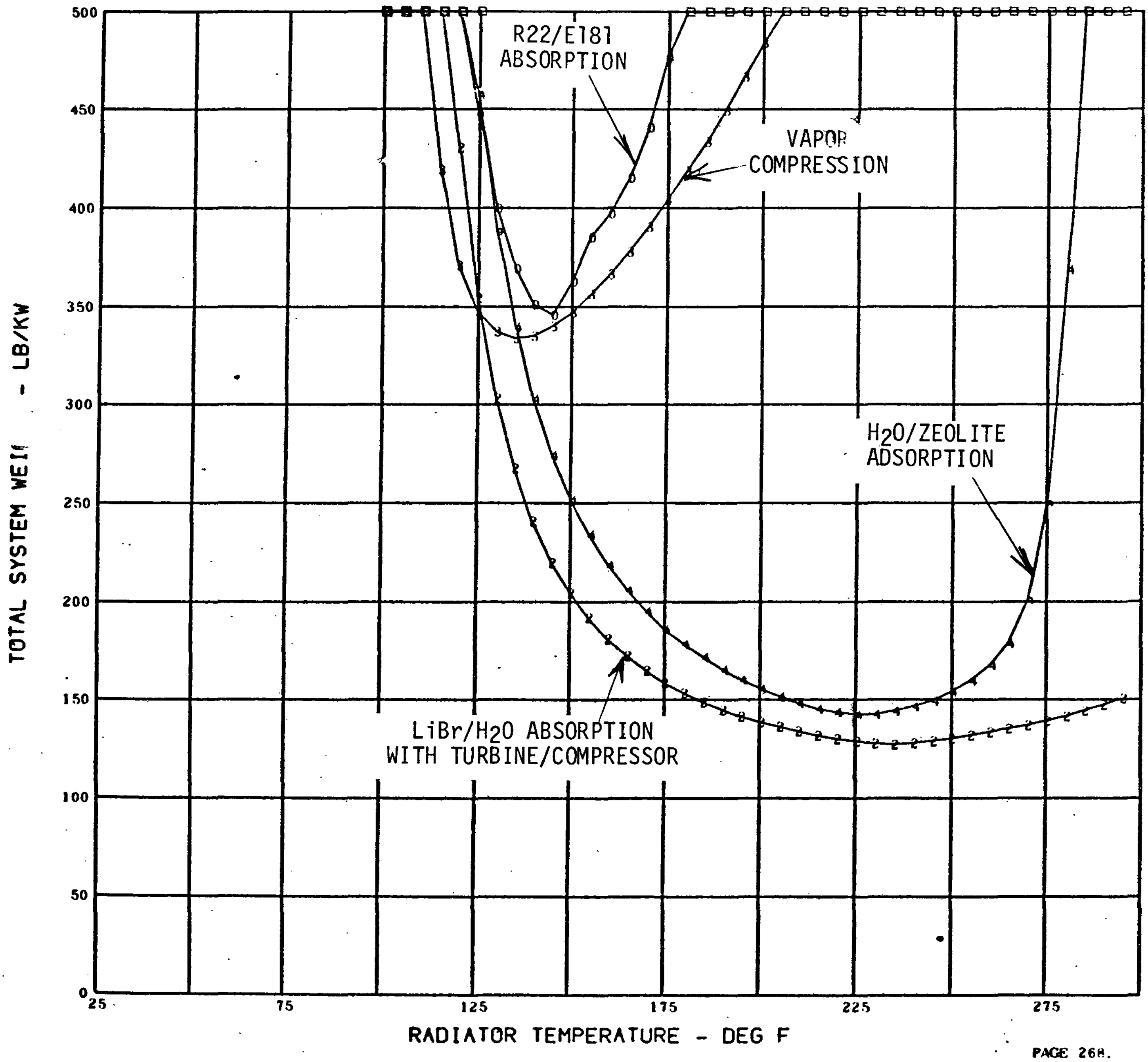
TEVAP = 45 F
TSINK = 100 F
FEPP = 700 LB/KW

FRAP = 1 LB/FT^2
FTHER = 0 LB/KW$_T$



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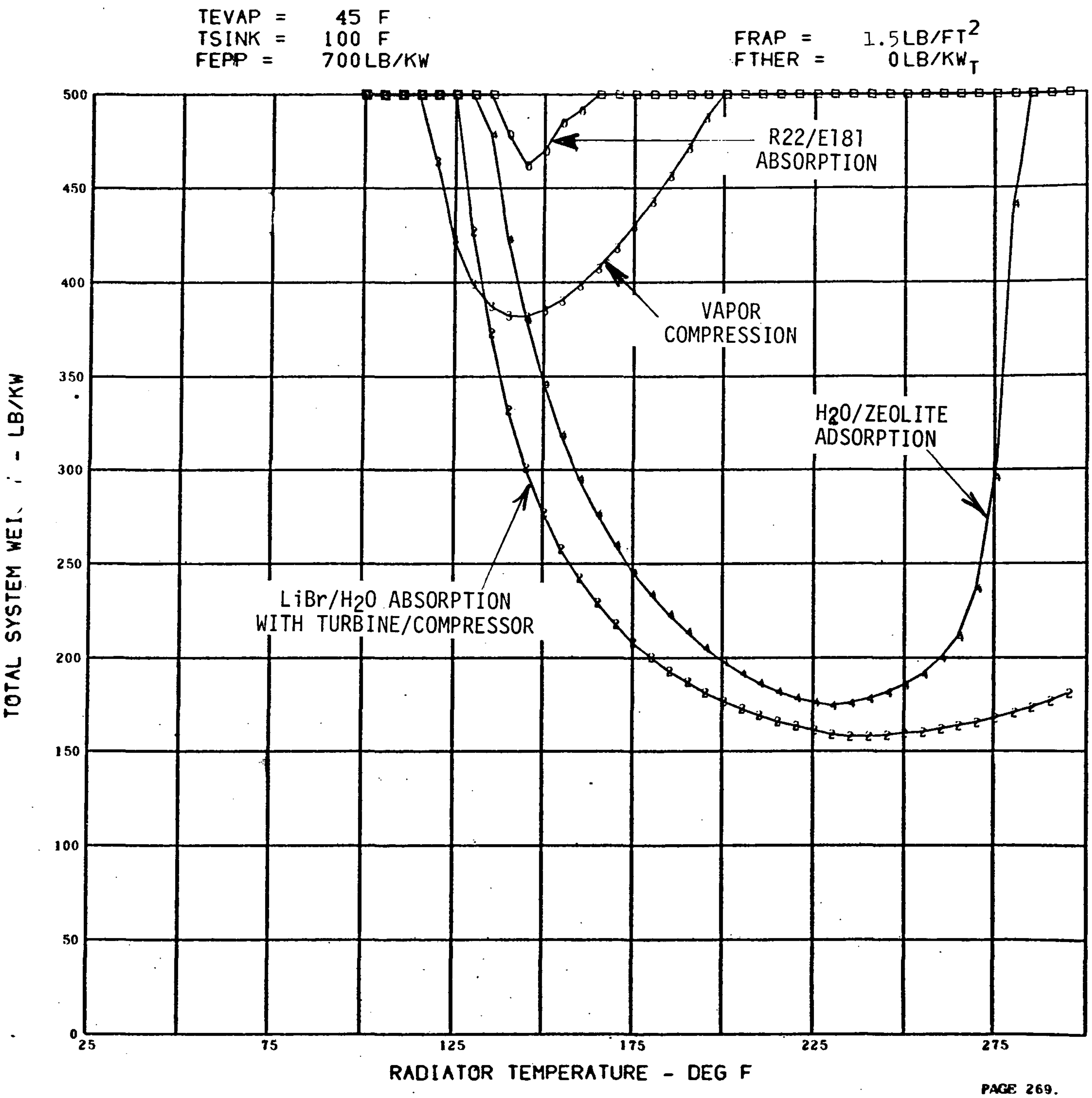
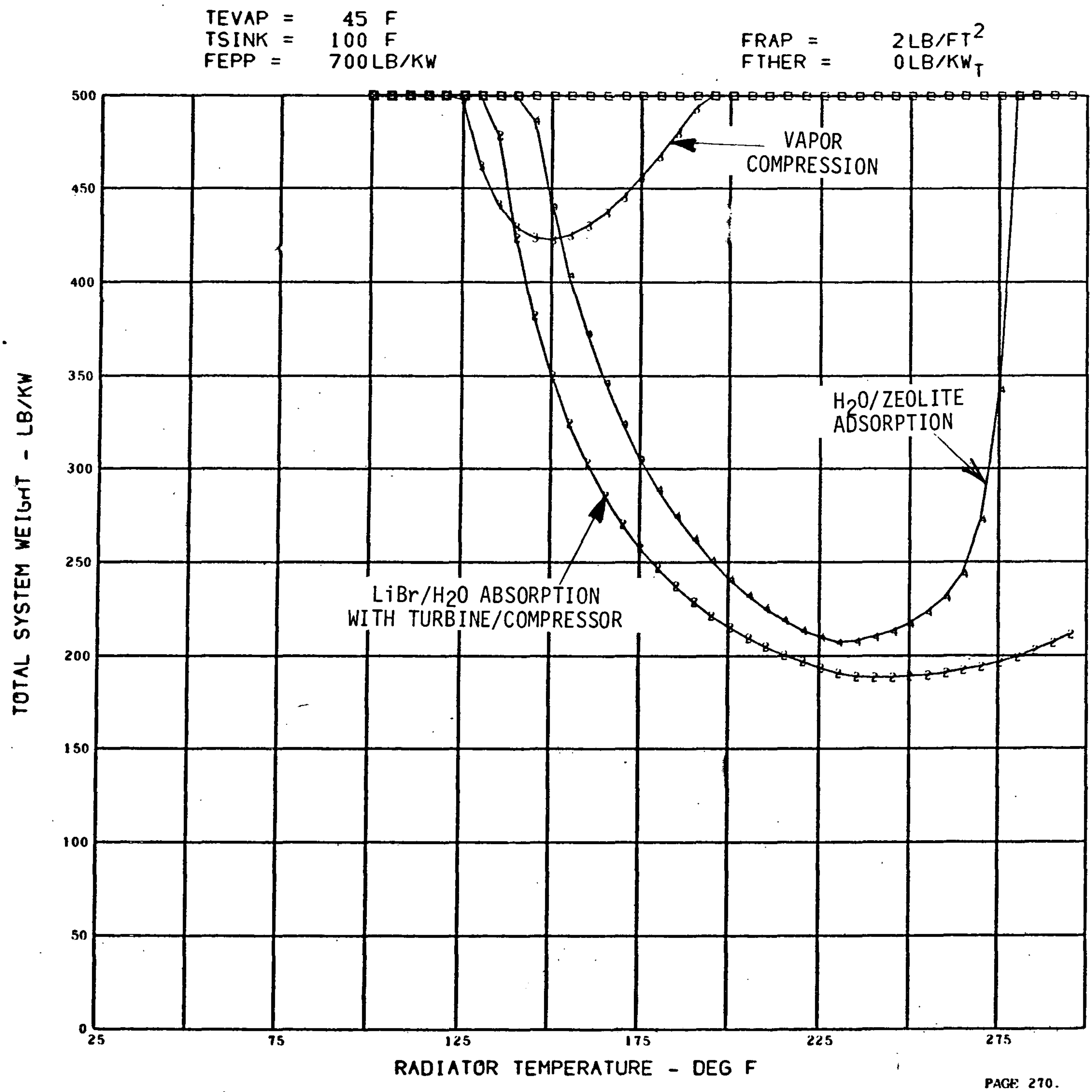


TEVAP = 45 F
TSINK = 100 F
FEPP = 700 LB/KW
FRAP = 1.5 LB/FT2
FTHER = 0 LB/KWT
R22/E181 ABSORPTION
VAPOR COMPRESSION
H2O/ZEOLITE ADSORPTION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEI - LB/KW
RADIATOR TEMPERATURE - DEG F
0
50
100
150
200
250
300
350
400
450
500
25
75
125
175
225
275
PAGE 269.



TEVAP = 45 F
TSINK = 100 F
FEPP = 700 LB/KW
FRAP = 2 LB/FT2
FTHER = 0 LB/KWT
VAPOR COMPRESSION
H2O/ZEOLITE ABSORPTION
LiBr/H2O ABSORPTION WITH TURBINE/COMPRESSOR
TOTAL SYSTEM WEIGHT - LB/KW
RADIATOR TEMPERATURE - DEG F
0
50
100
150
200
250
300
350
400
450
500
25
75
125
175
225
275
PAGE 270.