OAK RIDGE NATIONAL LABORATORY

operated by

UNION CARBIDE CORPORATION

NUCLEAR DIVISION

for the

U.S. ATOMIC ENERGY COMMISSION **MASTER**

ORNL - TM - 2132

**FLYWHEEL-AUGMENTED FLOW COASTDOWN
FOR THE 3500 Mwt, UNCLAD,
METAL-FUELED FBR (UMBR).**

R. P. Wichner

NOTICE This document contains information of a preliminary nature and was prepared primarily for internal use at the Oak Ridge National Laboratory. It is subject to revision or correction and therefore does not represent a final report.

JUL 10 1968

DISCLAIMER

This report was prepared as an account of work sponsored by an agency of the United States Government. Neither the United States Government nor any agency Thereof, nor any of their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness, or usefulness of any information, apparatus, product, or process disclosed, or represents that its use would not infringe privately owned rights. Reference herein to any specific commercial product, process, or service by trade name, trademark, manufacturer, or otherwise does not necessarily constitute or imply its endorsement, recommendation, or favoring by the United States Government or any agency thereof. The views and opinions of authors expressed herein do not necessarily state or reflect those of the United States Government or any agency thereof.

DISCLAIMER

Portions of this document may be illegible in electronic image products. Images are produced from the best available original document.

LEGAL NOTICE

This report was prepared as an account of Government sponsored work. Neither the United States, nor the Commission, nor any person acting on behalf of the Commission:

- A. Makes any warranty or representation, expressed or implied, with respect to the accuracy, completeness, or usefulness of the information contained in this report, or that the use of any information, apparatus, method, or process disclosed in this report may not infringe privately owned rights; or
- B. Assumes any liabilities with respect to the use of, or for damages resulting from the use of any information, apparatus, method, or process disclosed in this report.

As used in the above, "person acting on behalf of the Commission" includes any employee or contractor of the Commission, or employee of such contractor, to the extent that such employee or contractor of the Commission, or employee of such contractor prepares, disseminates, or provides access to, any information pursuant to his employment or contract with the Commission, or his employment with such contractor.

Contract No. W-7405-eng-26

NUCLEAR DESALINATION PROGRAM

FLYWHEEL-AUGMENTED FLOW COASTDOWN
FOR THE 3500 Mwt, UNCLAD,
METAL-FUELED FBR (UMBR).

R. P. Wichner

JULY 1968

OAK RIDGE NATIONAL LABORATORY
Oak Ridge, Tennessee
operated by
UNION CARBIDE CORPORATION
for the
U. S. ATOMIC ENERGY COMMISSION

LEGAL NOTICE

This report was prepared as an account of Government sponsored work. Neither the United States, nor the Commission, nor any person acting on behalf of the Commission
A. Makes any warranty or representation, expressed or implied, with respect to the accuracy, completeness, or usefulness of the information contained in this report, or that the use of any information, apparatus, method, or process disclosed in this report may not infringe privately owned rights, or
B. Assumes any liabilities with respect to the use of, or for damages resulting from the use of any information, apparatus, method, or process disclosed in this report.
As used in the above, "person acting on behalf of the Commission" includes any employee or contractor of the Commission, or employee of such contractor, to the extent that such employee or contractor of the Commission, or employee of such contractor prepares, disseminates, or provides access to, any information pursuant to his employment or contract with the Commission, or his employment with such contractor.

,

,

,

,

TABLE OF CONTENTS

	<u>Page</u>
ABSTRACT.....	1
INTRODUCTION.....	1
ROLE OF COASTDOWN IN SODIUM-COOLED REACTOR SAFETY.....	3
PAST WORK ON FLOW COASTDOWN.....	5
ASSUMPTIONS AND DEVELOPMENT.....	7
RESULTS FOR THE UMBR.....	15
CONCLUSIONS.....	23
BIBLIOGRAPHY.....	25
APPENDICES	
A. UMBR Primary Loop Characteristics.....	27
B. Flow Coastdown with Impeller Removed.....	29
C. Upper Limit on Flywheel Size.....	31
D. Program KINGS8D for the Calculation of Square, 8-Shoe Kingsbury Bearing Characteristics and Coastdown Flowrates.....	33

FLYWHEEL-AUGMENTED FLOW COASTDOWN
FOR THE 3500 Mwt, UNCLAD,
METAL-FUELED FBR (UMBR).

Abstract

Equations are presented for the coastdown flowrate in a loop following pump power failure which include the effects of diminishing pump efficiency during coastdown and the retarding torque of the thrust bearings on the pump rotation rate. The bearings are assumed to be of the Kingsbury type with eight, square shoes.

Results are given for the primary coolant loop of the 3500 Mwt, sodium-cooled, bare-metal, FBR, for cases utilizing light weight (15,000 kg), medium (60,000 kg), and heavy (240,000 kg) flywheels with 6 m diameter. It is shown that the useful coastdown duration lasts 15 seconds with no added flywheel, and about a factor of 30 increase is attained by adding the light flywheel to each of the primary pump motors. The medium weight flywheels could increase the useful coastdown duration to 2100 seconds, while the heavy flywheels resulted in coastdowns ranging from 6000 seconds to 8100 seconds, depending on the bearing design.

In all cases investigated the loss of pump efficiency during coastdown had a much larger effect than that of the thrust bearing retarding torque. Thrust bearing design was found to appreciably affect the useful coastdown duration only for the "heavy" flywheel cases where a 25% range was obtained between the lightest and heaviest bearing loadings.

1. INTRODUCTION

This report considers the possibility of augmenting the coastdown flowrate in a loop following pump power failure by attaching flywheels to the motor shafts. Equations are presented for the coastdown flow and pressure drop as a function of time which includes the effect of diminishing pump efficiency during coastdown and the retarding torque

due to the friction in the thrust bearings. Previous closed form solutions neglected these effects, which are shown to be significant for coastdowns of long duration, whereas other more detailed works are completely numerical.

The main assumption which enables mathematical simplification while yet allowing inclusion of the efficiency loss and thrust bearing effects is that the recovery of flow kinetic energy is negligible compared with the recovery of the rotational kinetic energy contained in the motor armature and flywheel. This is shown to be a valid assumption for the flow loop considered in this report, but its validity must be reexamined for application to other flow systems.

The case presently considered is the primary coolant loop of the unclad, metal-fueled fast breeder (hereafter termed the UMBR) described fully in reference (1). This is a 3500 Mwt, sodium-cooled concept containing three parallel circuits in the primary coolant system. Centrifugal pumps driven by 5,000 hp electric motors impel the coolant in each loop. While the presented results apply specifically to this case, the primary coolant system of the UMBR is not atypical in volume, flowrate, and pressure drop compared with published 1000 Mwe FBR designs. Hence, the general conclusions would apply to these cases also.

2. ROLE OF COASTDOWN IN SODIUM-COOLED REACTOR SAFETY

In the event of a site power failure, coastdown power could provide the primary coolant system motive force until an emergency powered coolant system becomes activated. This may involve a 10 to 20 second time lapse, which is sufficiently short so that no added flywheel on the motor shaft need be provided.

It is interesting to speculate on a possible larger role for pump coastdown involving added flywheels and time intervals of perhaps an hour or longer. For convenience in this preliminary discussion, a "useful coastdown period" is defined as that length of time following power cutoff in which the ratio of coastdown flow to rated flow exceeds the ratio of decay power to rated power. Such an interval exists because following the rapid decrease of fission power due to control rod action the decay power drops more slowly than the coastdown flow. During this period of time, it is reasonable to assume that the coastdown flowrate is sufficiently large to prevent formation of localized hot spots in the core. Whether or not this is actually the case must be verified by a more careful thermal-hydraulic analysis of the situation.

The chief advantage of a flywheel-motivated emergency cooling system would be increased reliability. No chain of events need be set in motion in an instant following an unexpected event. This is especially true if the coastdown flow is coupled with a natural convection heat sink - a combination which seems to offer the ultimate in reliability.

For example, a possible system might work as follows: Following a power failure or an emergency in which the primary pump power is lost, flywheel-augmented coastdown flow would prevent localized over heating in the core. Initially, the heat sink for the decay power would be the heat capacity of the primary loop, i.e., coolant, piping, reactor vessel, core and blanket fuel, heat exchanger, and reactor support structure. As the coolant temperature rises, the load on a natural convection loop rejecting heat to outside air (possibly through an intermediate exchanger) would tend to increase. Eventually, a balance will be attained when the driving force for natural convective

cooling rises to the point where the heat loss becomes equal to the decay power. The loop must be designed so that this occurs at a coolant temperature below which any structural failure or fuel damage occurs.

While such an emergency cooling system would be highly reliable, it would have the disadvantage of reducing the efficiency of the system. Both the added thrust bearing friction loss caused by the flywheel weight and the natural convection losses incurred during normal operation represent extra system losses. However, these are at least partially offset by savings in operating costs achieved by eliminating the maintenance and testing work required to keep a diesel system on immediate standby.

This type of emergency cooling system resembles that of EBR-II except that in EBR-II a standby EM pump plays the role taken by the coastdown-impelled flow. In EBR-II a NaK-filled natural convection loop discharges the decay heat to the atmosphere, the heat-transfer being aided by blowers. Power for the blowers and the EM pump is derived from diesel-driven generators.

In Fermi, DC pony-motors on the primary pump motors provide the mixing for the primary coolant in the event of a power failure. The decay heat is rejected to outside air via the reactor building ventilating system. Both the building fans and pony-motors are driven by diesel generated power.

The SEFOR emergency cooling system is essentially a complete, scaled-down version of the main coolant system, and hence differs substantially in concept from the above mentioned systems.

Whether or not flywheel-augmented coastdown is incorporated into a reactor safety system depends on an evaluation of the safety system as a whole with respect to the hazards criteria. This report merely points out that useful coastdown durations of about one hour seem reasonable for 3500 Mwt, sodium-cooled FBR's, and should be considered as a possible part of such a system.

3. PAST WORK ON FLOW COASTDOWN

The works of Burgreen³ and Arker and Lewis² are prominent in the study of coastdown flows.

Burgreen³ presents an analytical solution for the coastdown flowrate which, however, neglects both the retarding effect of the thrust bearing on the rotor speed, and the loss of pump efficiency during coastdown. Hence his equations are expected to be accurate only for times soon after power shutoff. It will be shown that at times farther along in the coastdown period, the flowrate is very sensitive to these effects. His results, in addition, may be unnecessarily complicated by expressing the coastdown flowrate as a function of a nondimensionalized time characterized as some multiple of a "loop half-life". This is defined as the time for the flow to decay to 1/2 the initial value upon removal of the pump impeller from the loop. Its value is a function of the kinetic energy contained in the circulating fluid. It will be shown that for the present design, the flow kinetic energy is a small fraction of the rotor kinetic energy, and the "loop half-life" is 0.39 seconds.

A comparison of the coastdown flowrate vs time predicted by Burgreen's equation with that calculated using the present method is given in the results section of this report. (See Figure 4, page 21).

Arker and Lewis² present a rather complete discussion of the theory of coastdown flowrates. Their method is entirely numerical, and calculated results compare well with loop data. However, the test loops were small; rotor polar moments were of the order 2.5 kg-m^2 compared with a smallest possible value for the present case of 5000 kg-m^2 . Also the results are presented in terms of "time units" which are not defined. The calculation presented by Arker and Lewis differs from the present analysis in the following respects:

1. Recovery of flow kinetic energy is taken into account in the Arker and Lewis study. For the present case it will be shown that flow kinetic energy is negligible compared to rotor kinetic energy, and hence can be neglected. An appreciable computational simplification results thereby.

2. The retarding torque due to windage was taken into account by Arker and Lewis while that due to thrust bearing friction was not. Windage torque diminishes as the square of the rotor speed; more rapidly than does the fluid retarding torque which is of much larger magnitude. Hence, it appears that the windage effect, which is initially small, would very quickly become negligible during coastdown and have no significance on the useful coastdown duration. On the other hand, the thrust bearing retarding torque diminishes more slowly during coastdown and may appreciably affect the useful coastdown duration.

3. Arker and Lewis took into account the retarding torque caused by the residual magnetic field in the motor, whereas this factor was neglected in the present analysis. The equations they used apply specifically to the canned-rotor, induction motor, and no indication is given of the magnitude of the effect on the coastdown flow. As discussed more fully later in the report, it appears that the residual field stopping torque is negligible for standard induction motors, and probably not negligible for synchronous motors.

4. It was pointed out by Arker and Lewis that the impeller efficiency varies with rotation speed. However, in the cases they carried through, its value remained constant, whereas in the present calculation its variation during coastdown was taken into account.

4. ASSUMPTIONS AND DEVELOPMENT

A. Energy Balance

The energy balance on the decelerating impeller - flywheel system may be written,

$$-\frac{d}{dt} \left(\frac{1}{2} I \omega^2 \right) = W_p + W_b + W_{em} + W_w \quad (1)$$

where

I = polar moment of rotating assembly,

W_p = pump work rate,

W_b = bearing friction work rate,

W_{em} = work rate required to overcome the residual magnetic field in the motor,

W_w = windage work rate.

The work rates are functions of the shaft rotation speed, ω . If these functions are known, Eq. (1) may be solved for the coastdown rotation rate, $\omega(t)$.

B. Coastdown Pump Power

The pump efficiency, η , is defined as the ratio of the useful pump output to the actual power input to the impeller. Hence we may write,

$$W_p = \frac{Q \Delta P}{\eta} \quad (2)$$

where the product of the volumetric flowrate and the pressure rise, $Q \Delta P$, represents the useful pump output. It will be assumed that the affinity law relating the developed head to the square of the rotation speed holds during the coastdown period. That is,

$$\Delta P(\omega) = K_1 \omega^2, \quad (3)$$

where the constant, K_1 , is evaluated from the loop design pressure drop, ΔP_o , and the rated shaft speed, ω_o .

The coolant flow developed by pressure difference, $\Delta P(t)$, is a function of the loop flow resistance. An approximation of this relationship may be written,

$$Q(\omega) = K_2' \sqrt{\Delta P(\omega)} \quad (4)$$

where again the constant K'_2 is evaluated from the known rated volume flow, Q_o , and the design pressure drop. Equation (4) is expected to be fairly accurate for a loop where the losses are primarily inertial (inlet, outlet, orifices, etc.) for which

$$\Delta P_{\text{losses}} \propto U^2, \quad (5)$$

where U is the flow velocity. For the present design¹ such losses account for 75% of the total pressure drop largely due to the core inlet orifices. The remaining 25% are viscous losses which vary with Reynolds number and surface roughness. However, Eq. (5) (and hence also (4)) applies for turbulent flows along rough surfaces; for other cases it is a conservative relationship, i.e., it predicts high losses.

Rewrite Eq. (4) more conveniently to obtain,

$$Q(\omega) = K_2 \omega. \quad (6)$$

The efficiency at rated conditions, η_o , of a large capacity centrifugal pump with a specific speed^a of 2400, as required for the primary coolant loop of UMBR, is estimated to be approximately 93%.⁴ As the shaft rotation speed diminishes during coastdown, the efficiency will drop at a rate that depends on the pump specific speed at rated conditions. Figure 1 shows this relationship as given by Stepanoff⁴ for a single suction pump^b with a specific speed of 2400. The indicated points were calculated from the parabolic approximation,

$$\eta(\omega) = \eta_o \left[1 - (1 - \omega/\omega_o)^2 \right], \quad (7)$$

which is seen to give accurate values for $\eta(\omega)$ throughout the entire range of relative flows. For specific speeds below 2400, Eq. (7) yields conservative (i.e., low) values for the efficiency. For higher specific speeds, the relationship tends toward a linear diminution of η with relative flow, and the following development could easily be revised for that case if desired.

^aSpecific speed = $\text{rpm} \sqrt{\text{gpm}} / (\text{ft head})^{\frac{3}{4}}$. See Appendix A for primary coolant loop data for the UMBR.

^bNote: Some of the efficiency data are given for double suction pumps. To compare with equivalent single suction pump, divide the specific speed by $\sqrt{2}$.

Fig. 1 Efficiency vs. Relative Flow for Single Suction Specific Speed of 2400. From Stepanoff⁴.

Substitution of Eqs. (3), (6) and (7) into Eq. (2) yields for the pump work during coastdown,

$$W_p(\omega) = \frac{K_1 K_2 \omega^3}{\eta_o [1 - (1 - \omega/\omega_o)^2]} \quad (8)$$

Some further conditions must be met for Eq. (8) to be valid. It has been implicitly assumed throughout this section that during coastdown the impeller pumps, not retards, the fluid. A situation is conceivable for a light rotor assembly with high friction where the momentum in the coolant would tend to maintain a flow larger than would be delivered by the coasting impeller. The results are not valid for such a condition where the impeller acts as a brake. In Appendix B, the coastdown rate of the UMBR primary coolant loop with the impeller removed is estimated, and a flow half-life in the order of 0.39 seconds may be expected from the recovery of the flow kinetic energy. Hence we would not expect the presented results to be realistic unless the calculated flow half-lives are substantially longer than 0.39 seconds, which they are for all cases considered.

Also, in writing Eq. (4), the recovery of the contained flow kinetic energy has been neglected. This approximation is valid when the kinetic energy stored in the rotating assembly - the recovery of which creates the pressure driving force for flow - greatly exceeds the kinetic energy recoverable from the circulating coolant. For the present case, a rotor assembly having a polar moment of 538,000 (kg-m²), which would be representative of a moderate weight flywheel^a afixed to a 5000 hp motor shaft driving a 2000 gpm impeller, contains 6×10^8 joules of rotational kinetic energy at the rated speed of 450 rpm. Thus the total contained rotational kinetic energy for the 3 pumps in the system is 18×10^8 joules as compared with an estimated flow kinetic energy of 2.6×10^6 joules^b at full design conditions. The flow kinetic energy is thus 0.14% of the rotational kinetic energy and hence may be neglected.

^aSee Tables 1 and 2.

^bSee Appendix A.

C. Windage Power Loss

The equation for the windage power loss as a function of shaft speed is given as,²

$$W_w = \text{const. } \omega^3 .$$

Comparison with Eq. (8) indicates that the windage power loss, which initially consumes about 1% of the total power, diminishes far more rapidly with ω than does the pumping power. Hence, we conclude that its effect on the duration of useful coastdown period is negligible.

D. Thrust Bearing Retarding Torque

The thrust bearing employed would most likely be of the Kingsbury type - a pivoted shoe, hydrodynamic bearing - which offers the combined advantages of long wear life and simplicity. Moderately loaded^a Kingsbury bearings require no oil pressure system, the lubricating film being maintained by the relative motion of the upper and lower shoes.

Hydrostatic thrust bearings on the other hand are far less common than Kingsbury bearings, but offer the possibility of lower frictional drag at rated conditions. In addition, it can be shown that the drag drops more rapidly with angular speed during coastdown, hence, useful coastdown durations could be increased by adopting hydrostatic thrust bearings. However, the disadvantage in their use lies in the fact that an oil pressurization system must function at all times, even during an emergency coastdown period. Kingsbury bearings were thus chosen for the present study because of their higher reliability.

Kent's Handbook⁵ gives the following semiempirical design formulae for square, 8-shoe, Kingsbury thrust bearings:

$$h_o = \sqrt{\frac{0.28\mu v d}{p_o}} , \quad (9)$$

$$W'_b = 1.494 \times 10^{-2} \mu \frac{A_s v^2}{h_o} \quad (10)$$

where h_o is the minimum oil film thickness in inches, p_o the loading in lb_f/in^2 , A_s the shoe area in in^2 , μ the oil viscosity in Reyns ($\text{lb}_f\text{-sec}/\text{in}^2$),

^aModerate loading is defined as bearing pressures less than approximately 500 psi. Heavily loaded Kingsbury bearings may require hydrostatic lift assistance at shutdown and startup to prevent shoe damage.⁷

v the mean relative velocity of the shoes in ft/sec, d the mean bearing diameter in inches. W_b' is the work rate per shoe in horsepower required to overcome the frictional resistance of the lubricating film.

Substituting Eq. (9) into Eq. (10) and noting that

$$\frac{w \times d}{24} = v ,$$

yields the relationship between the bearing friction power velocity to be,

$$W_b = K_3 \omega^{3/2} . \quad (11)$$

To obtain W_b in MKS units, K_3 is evaluated from,

$$K_3 = \frac{1.3558 \times 550 \times 0.0149}{24^2} \times A_s d \sqrt{\frac{24 p_o \mu}{0.28}} \quad (12)$$

Note, from Eq. (9), that during coastdown the oil film thickness continually decreases from its initial design value as follows:

$$h_o \propto \sqrt{\omega} .$$

Experiments have shown that as the film thickness decreases, a critical point is reached where the nature of the lubricating film changes to one characterized by substantially increased resistance. While an accurate prediction of this critical film thickness can not be made, depending as it does on such things as bearing roughness and chemical nature of bearing and lubricant, Hersey⁸ states that 0.025 and 0.1 mils are reasonable limiting values. The lower value is given as the more uncertain one whereas it is stated with some positiveness that the thin-film regime is unlikely for lubricating films of 0.1 mil thickness and larger.

Hence, we will conservatively assume that the coastdown period is terminated when the film thins to 0.1 mils.

E. Effect of Motor Residual Field

Since the stator poles of an induction motor are specifically designed for rapid polarity change with minimum hysteresis loss, it is expected that the residual stator field will be quite small after power failure. However, it has not yet been established that "quite

small" is small enough to be negligible in effect on coastdown flows of long duration - although many people feel this is the case.

Both Arker and Lewis² and G. M. Boyd, et. al.,⁶ offer expressions for the EM retarding torque in canned-rotor type induction motors. These expressions are quite different, and in neither paper is there an indication of the magnitude of the term relative to other effects.

The residual field in a synchronous motor would be substantially larger than expected for an induction motor, and certainly would appreciably affect the coastdown flowrate. The armature of a synchronous motor is a DC electromagnet with a large permanent component which will interact with the ferromagnetic stator pole material long after power failure. If the stator circuit is closed, the motor will act like a generator powered by the rotational kinetic energy of the armature. In fact, short circuiting the stator windings is a common method of braking the rotation of synchronous motors. If braking action is not desired, circuit breakers are placed in the stator power supply which operate to open the circuit after power failure. These take from two to four cycles to become actuated, after which the stopping torque diminishes, but remains much larger than that of an induction motor.

F. Solution

Substituting Eq. (8) for the pumping power and Eq. (11) for the bearing power into Eq. (1), neglecting W_{em} and W_w , yields,

$$-\frac{d}{dt} \left(\frac{1}{2} I \omega^2 \right) = \frac{K_1 K_2 \omega^3}{\eta_o \left[1 - (1 - \omega/\omega_o)^2 \right]} + K_3 \omega^{3/2} \quad (13)$$

Letting $y = \omega/\omega_o$ and rearranging yields,

$$\frac{dy}{dt} = - \frac{Ay}{2-y} - B\sqrt{y} \quad (14)$$

where

$$A = \frac{K_1 K_2 \omega_o}{\eta_o I} \quad , \quad \text{and} \quad (15)$$

$$B = \frac{K_3}{I \sqrt{\omega_o}} \quad (16)$$

Changing variables,

$$z = \sqrt{y} \quad , \quad (17)$$

allows regrouping into forms appearing in a standard table of integrals,

e.g.,

$$dt = \frac{(2z^2 - 4)dz}{-Bz^2 + Az + 2B} \quad (18)$$

Integrating between the limits,

$$z = 1 \quad \text{at} \quad t = 0 \quad \text{to}$$

$$z = z' \quad \text{at} \quad t = t' \quad ,$$

yields the result (dropping the prime),

$$t = \frac{2(1-z)}{B} + \frac{C}{B} \ln \left(\frac{1+C}{2+Cz-z^2} \right) + \frac{C^2}{B\sqrt{8+C^2}} \ln \left[\frac{-2z+C-\sqrt{8+C^2}}{-2z+C+\sqrt{8+C^2}} \cdot \frac{-2+C+\sqrt{8+C^2}}{-2+C-\sqrt{8+C^2}} \right] \quad (19)$$

Where C is the ratio A/B.

5. RESULTS FOR THE UMBR

Pertinent characteristics of the primary coolant loop of the UMBR are presented in Table A.1 of Appendix A from which the following values of the coastdown constants K_1 and K_2 are determined:

$$K_1 = 300 \text{ kg/m}$$
$$K_2 = 0.12 \text{ m}^3 \text{ .}$$

In addition to the case where no flywheel is attached to the motor shaft, coastdown calculations were performed for representative light, medium and heavy added flywheels. The dimensions, masses and polar moments for these assumed flywheels are listed in Table 1, and the masses and polar moments of the armature - flywheel combinations are given in Table 2. It should be emphasized that these flywheel sizes are arbitrarily selected for illustrative purposes and careful attention was not given to flywheel design problems. It was assumed that 6 meters was a practical limit on flywheel diameter, which is well below the theoretical bursting diameter of 9.8 meters for carbon steel at 450 rpm (See Appendix C). Also, these assumed flywheel masses yielded very reasonable thrust bearing dimensions for various assumed moderate to low bearing pressures.

A Fortran-IV program named KINGS8D used for the coastdown calculations, is presented in Appendix D together with a typical output. The program first calculates the thrust bearing dimensions, initial oil-film thickness and bearing constant K_3 for a square, 8-shoe Kingsbury bearing. The times required to reach various imbedded values of the coastdown rotation rate are then calculated from Eqs. (11), (16), (17), and (19). An oil viscosity of 30 cp was assumed in all the calculations.

Results are presented in Figs. (2) and (3) for bearing pressures of 150 and 450 psi respectively, which are representative of light and moderate bearing loadings. The relative decay power, as given by the Way-Wigner equation is superimposed on the given relative coastdown flows. The "useful coastdown duration", the time at the intersection of the decay power and coastdown curves, represents the period of time after shutdown in which the relative flow exceeds the relative decay power.

Table 1. Assumed Flywheel Characteristics^a

	<u>Flywheel Characteristics</u>		
	<u>Light</u>	<u>Medium</u>	<u>Heavy</u>
Average diameter, m	6	6	6
Thickness, m	.1	.4	.8
Height, m	1	1	2
Mass, kg	15,000	60,000	240,000
Polar moment, kg-m ²	133,000	533,000	2,160,000

Table 2. Total Masses and Polar Moments of the Armature plus Flywheel Combinations in the Four Cases Studied

	<u>Mass (kg)</u>	<u>Polar Moment (kg-m²)</u>
Armature only ^b	10,000	5,000
Light Flywheel + Armature	25,000	138,000
Medium Flywheel + Armature	70,000	538,000
Heavy Flywheel + Armature	250,000	2,165,000

^aThe flywheels are assumed to be cylinders with thin annular cross-sections. The mass and polar moment associated with the structural attachments to the shaft are assumed to be negligible.

^bApproximate value for 5000 hp induction motor.

Fig. 2 Relative Coastdown Flow vs. Time After Shutdown for Various Flywheel Sizes. Bearing Pressure = 150 psi; Oil Viscosity = 30 cp. Curve a - No Flywheel, Curve b - Light Flywheel, Curve c - Medium Flywheel, Curve d - Heavy Flywheel. See Tables 1 and 2 for flywheel weights and dimensions.

Fig. 3 Relative Coastdown Flow vs. Time after Shutdown for Various Flywheel Sizes. Bearing Pressure = 450 psi; Oil Viscosity = 30 cp. Curve a - No Flywheel, Curve b - Light Flywheel, Curve c - Medium Flywheel, Curve d - Heavy Flywheel. See Tables 1 and 2 for flywheel weights and dimensions.

The time at which the lubricating film thickness reaches its minimum value of 0.1 mil is indicated by the dashed line at lower right in Figs. (2) and (3). At this thickness, a substantial increase in bearing friction is possible, and it was conservatively assumed that the coastdown is terminated at this point. Note that thin-film termination plays no role in establishing the magnitude of the useful coastdown for the cases presented in Figs. (2) and (3).

Table 3 lists the useful coastdown durations for the cases shown in Figs. (2) and (3), as well as some cases at bearing pressures of 300 and 600 psi.

It may be seen that the bearing design appreciably affects the useful coastdown duration only for the heavy flywheel case where a 25% difference may be noted between the lightest and highest bearing loadings.

In no calculated case was the useful coastdown duration terminated by loss of adequate bearing film lubrication. For the medium and heavy flywheel cases, this would occur at bearing loadings above 600 psi, the highest bearing pressure case tested. With no flywheel, it appears that bearing loadings of 600 psi and above would result in termination of the useful coastdown period by loss of lubrication.

Other bearing data for each case are also presented in Table 3. The friction power loss in the bearing at rated conditions ranged between 0.025% and 9.38%. Mean bearing diameters ranged from 7.42 in to 64.3 in; initial lubricating film thicknesses varied from 0.54 mils to 8.12 mils. The longest useful coastdown duration was achieved with the heavy flywheel at 600 psi bearing loading - 8100 sec; this is to be compared with approximately 15 sec without flywheel.

Figure (4) illustrates the effect of varying the thrust bearing loading on the coastdown flow for cases utilizing the heavy flywheel. The solid lines represent coastdown flowrates calculated from Eq. (19) for the three indicated bearing pressures. These are compared with the dashed line obtained using Burgreen's equation³ which neglects the effect of both bearing friction and the loss of pump efficiency. The mark - ♦, which ends each relative flow curve, signifies coastdown termination due to loss of lubrication. As the bearing pressure increases

TABLE 3. CALCULATED USEFUL COASTDOWN DURATIONS AND KINGSBURY
THRUST BEARING CHARACTERISTICS FOR VARIOUS ASSUMED
ADDED FLYWHEELS.^a OIL VISCOSITY = 30 cp.

	<u>Without Flywheel</u>	<u>Light Flywheel</u>	<u>Medium Flywheel</u>	<u>Heavy Flywheel</u>
<u>Bearing Pressure = 150 psi</u>				
Useful Coastdown Duration (sec)	15	510	2000	6000
K_3 (newt-m-sec ^{$\frac{1}{2}$})	8.65	34.2	160	1080
Initial Friction Loss to Bearings (% Rated Power)	0.075	0.297	1.39	9.38
Initial Film Thickness (mils)	1.62	2.57	4.30	8.12
Mean Diameter of Bearing (in)	12.9	20.3	34.0	64.3
<u>Bearing Pressure = 300 psi</u>				
Useful Coastdown Duration (sec)	15	510	2050	7200
K_3 (newt-m-sec ^{$\frac{1}{2}$})	4.32	17.1	80.1	540
Initial Friction Loss (% Rated Power)	0.038	0.148	0.695	4.69
Initial Film Thickness (mils)	0.81	1.28	2.15	4.06
Mean Diameter of Bearing (in)	9.1	14.4	24.1	45.5
<u>Bearing Pressure = 450 psi</u>				
Useful Coastdown Duration (sec)	15	510	2100	7800
K_3 (newt-m-sec ^{$\frac{1}{2}$})	2.82	11.4	53.4	360
Initial Friction Loss (% Rated Power)	0.025	0.099	0.463	3.13
Initial Film Thickness (mils)	0.54	0.856	1.43	2.71
Mean Diameter of Bearing (in)	7.42	11.7	19.6	37.1
<u>Bearing Pressure = 600 psi</u>				
Useful Coastdown Duration (sec)			2140	8100
K_3 (newt-m-sec ^{$\frac{1}{2}$})			40.0	270
Initial Friction Loss (% Rated Power)			0.347	2.34
Initial Film Thickness (mils)			1.07	2.03
Mean Diameter of Bearing (in)			17.0	32.1

^aSee Tables 1 and 2 for flywheel weights and dimensions.

Fig. 4. Coastdown Flows and Useful Coastdown Durations for the Heavy Flywheel Case. Comparison of Calculated Values Assuming Constant Pump Efficiency and Zero Bearing Friction with Values Obtained Assuming Varying Efficiency and Bearing Pressures of 150, 300 and 600 psi.

(which means smaller bearings) the frictional drag in the bearing diminishes, but the lubricating film is lost sooner. The coastdown curve for the 600 psi loading more closely approximates that obtained assuming no friction in the bearing than the other two cases. It is seen that when the frictionless and non-varying efficiency assumptions are adopted a coastdown period of 2.1×10^5 sec is indicated as compared with 6000 sec for the 150 psi loading when these effects are taken into account.

6. CONCLUSIONS

- (1) An emergency cooling system involving flywheel-augmented pump coastdown and a natural convection heat sink appears to offer high reliability for sodium-cooled reactors.
- (2) Equations are derived for the coastdown flowrate in a loop following pump power failure which include the effect of loss of pump efficiency and the retarding torque of the thrust bearings on the pump rotation rate. Previous closed form solutions neglected these effects which appreciably shorten the useful coastdown duration. Other, more detailed works are completely numerical.
- (3) The main assumptions of the analysis are:
 - a. the pump developed head is proportional to the square of the shaft speed, and the flowrate is proportional to the shaft speed;
 - b. the pump loses efficiency with flowrate in a manner given by Stepanoff.⁴ The parabolic approximation of this relationship, (Eq. (7)) - which was used in the analysis, is quite accurate for the centrifugal pump with a specific speed of 2400 as required in the primary coolant loop of UMBR;
 - c. the kinetic energy of flow is small compared with the rotational energy of the armature plus flywheel;
 - d. the thrust bearing is of the Kingsbury type, and the design equations given in reference (5) apply;
 - e. the regime of "thin-film" bearing lubrication is entered when the oil film thickness decreases to 0.1 mil. At this point it is conservatively assumed that the coastdown ends.
- (4) The useful coastdown duration for the UMBR is about 15 sec with no flywheel attached to the motor shaft. This may be increased to 510 sec by adding a relatively light (15,000 kg) flywheel with a 6 meter diameter.
- (5) Medium weight flywheel (60,000 kg) with a 6 meter diameter could increase the useful coastdown duration in the UMBR primary loop to 2100 sec.

- (6) Utilization of heavy flywheels (240,000 kg) with 6 meter diameter could result in useful coastdown durations ranging between 6000 sec and 8100 sec depending on the thrust bearing design.
- (7) Higher thrust bearing design pressures mean smaller bearing and lower frictional drag. Hence, as the bearing pressure increases, the calculated coastdown curves approaches that attained when the retarding effect of bearing friction is neglected. However, increasing the thrust bearing loading also diminishes the lubricating film thickness, and entry into the "thin-film" regime, characterized by greatly increased drag, occurs at earlier in the coastdown period. However in none of the fourteen cases run, was the useful coastdown duration terminated by loss of adequate thrust bearing lubrication.
- (8) The bearing design appreciably affected the useful coastdown duration only for the heavy flywheel case where a 25% range was noted between the lightest and highest bearing loadings. In all cases, the loss of pump efficiency during coastdown was the dominating effect.

6. BIBLIOGRAPHY

- (1) R. P. Hammond, et al, "The Unclad Metal Breeder Reactor (UMBR) For Desalting or Power," ORNL-4202 (to be published).
- (2) A. J. Arker and D. G. Lewis, "Rapid Flow Transients in Closed Loops," TID-7529, Bk 1, 33-47 (1956).
- (3) D. Burgreen, "Flow Coastdown in a Loop After Pumping Power Cutoff," Nuc. Sci. and Eng. 6, 306-312 (1959).
- (4) A. J. Stepanoff, Centrifugal and Axial Flow Pumps, 2nd Edition John Wiley and Sons, New York (1957), pp 76, 163.
- (5) Kent's Mechanical Engineer's Handbook, C. Carmichael, Ed., 12th Edition, John Wiley and Sons, New York, pp. 12-34 to 12-35.
- (6) G. M. Boyd, et al, "Transient Flow Performance in a Multi-Loop Nuclear Reactor System," Trans. Am. Nuc. Soc. 2, 68(1959).
- (7) Personal Communication, A. Rosenquist, Assistant to the Chief Designer, Kingsbury Bearing Works, Philadelphia, Pa., verbal.
- (8) M. D. Hersey, Theory and Research in Lubrication, John Wiley and Sons, New York (1966), pp. 432-436.

ACKNOWLEDGEMENT

I would like to gratefully acknowledge the assistance given to me by M. L. Tobias, J. C. Moyers, D. L. Clark and R. D. Stulting of the Reactor Division of ORNL by their discussions and helpful comments. I am especially indebted to Lloyd V. Wilson, also of the Reactor Division, who called my attention to the manner in which the pump efficiency diminishes during coastdown and suggested the parabolic approximation (Eq. (7)) used in the analysis.

APPENDIX A. UMBR PRIMARY LOOP CHARACTERISTICS

The pertinent characteristics of the primary coolant loop are listed in Table A.1 from which the constants K_1 and K_2 in Eqs. (12) and (14) may be evaluated.

$$K_1 = \frac{\Delta P_o}{\omega_o^2} = \frac{6.6 \times 10^5 \text{ (newtons/m}^2\text{)}}{\left(\frac{450}{60}\right)^2 2\pi^2 \text{ (rad}^2\text{/sec}^2\text{)}}$$

$$\therefore K_1 = 300 \text{ (kg/m)}$$

$$K_2 = \frac{Q_o}{\omega_o} = \frac{5.88 \text{ (m}^3\text{/sec) per loop}}{\left(\frac{450}{60}\right) 2\pi \text{ (rad/sec)}}$$

$$\therefore K_2 = 0.12 \text{ (m}^3\text{)}$$

Note, the kinetic energy contained in the circulating coolant flow is 2.6×10^6 joules (assuming the kinetic energy in the primary heat exchanger is equal to that contained in the core and blanket) compared with a value of 18×10^8 joules for 3 rotor assemblies equipped with "medium" weight flywheels revolving at 450 rpm. Hence the flow kinetic energy is only 0.14% of the rotational kinetic energy, and thus may be neglected.

TABLE A.1. UMBR PRIMARY LOOP CHARACTERISTICS

Pressure drop, atm	6.6
Total Mass Flowrate, kg/s	15,000
Total Volume Flowrate, l/s	17,600
Number of Loops	3
<u>Loop Piping</u>	
Total length, m	137
Diameter, m	1.07
Flow area for 3 pipes, m ²	2.68
Velocity, m/s	6.6
Mass of contained sodium, kg	102,000
Kinetic energy contained, joules	2.22×10^6
<u>Core and Blanket</u>	
Flow area, m ²	2.45
Velocity, m/s	10
Mass of contained sodium, kg	1,800
Kinetic energy contained, joules	0.18×10^6
<u>Primary Heat Exchanger</u> (Flow conditions assumed similar to Core and Blanket)	
Pump rotation rate, rpm	450
Polar moment of motor armature, kg-m ² (5,000 hp induction motor)	~5,000
Mass of motor armature, kg	~10,000
Pump specific speed, rpm $\sqrt{\text{gpm}} / (\text{ft head})^{\frac{3}{4}}$	2,370

APPENDIX B. FLOW COASTDOWN WITH IMPELLER REMOVED

To assess the magnitude of flow momentum effect on coastdown assume the impeller is removed from the loop. The flow work rate, W_f , is then equal to the rate of recovery of flow kinetic energy, i.e.,

$$-\frac{d}{dt} \sum_i \frac{1}{2} m_i v_i^2 = W_f \quad , \quad (B.1)$$

where the subscript, i , refers to the various loop components - piping, core and blanket, heat exchanger, and m_i is the mass of sodium contained in each flowing with velocity v_i . Since,

$$W_f = \Delta P(t) Q(t) \quad (B.2)$$

and,

$$\Delta P(t) = K_4 Q^2 \quad , \quad (B.4)$$

which follows from Eq.(3), obtain

$$W_f = K_4 Q^3 \quad , \quad (B.5)$$

where K_4 is evaluated from the rated conditions as follows,

$$K_4 = \frac{\Delta P_o}{Q_o^3} \quad . \quad (B.6)$$

Also,

$$v_i = \frac{Q(t)}{A_i} \quad , \quad (B.7)$$

where A_i is the flow area in component i . Obtain by substitution of Eqs. (B.5) and (B.7) into (B.1),

$$-\frac{dQ}{dt} \sum_i \frac{m_i}{A_i^2} = K_4 Q^2 \quad . \quad (B.8)$$

Integrate from $t = 0$ when $Q = Q_0$ to time, t and obtain,

$$\frac{1}{Q(t)} - \frac{1}{Q_0} = \frac{K_4 t}{\sum \frac{m_i}{A_i^2}} \quad (\text{B.9})$$

Hence, for the loop flow half-life, $t_{1/2}$, when $Q = 1/2 Q_0$ obtain,

$$t_{1/2} = \frac{\sum \frac{m_i}{A_i^2}}{K_4 Q_0} \quad (\text{B.10})$$

Using values given in Appendix A yields the result,

$$t_{1/2} = \frac{\frac{102,000}{2.68^2} + \frac{2 \times 1800}{2.45^2}}{2130 \times 17.6}$$

$$t_{1/2} = 0.39 \text{ sec}$$

Hence, the flow stops quite abruptly with the impeller removed, and the assumption that the impeller provides the motive force during coastdown appears to be valid.

APPENDIX C. UPPER LIMIT ON FLYWHEEL SIZE

It can be shown that the tensile stress in a thin, annular flywheel generated by the centrifugal force is given by

$$\sigma = \frac{1}{2} \rho R^2 \omega^2, \quad (C.1)$$

where

ρ = flywheel material density,
 R = average radius,
 ω = angular speed.

At a particular ω , R_b is the bursting radius of the flywheel corresponding to the maximum allowable tensile stress, σ_m . Hence,

$$R_b = \sqrt{\frac{2\sigma_m}{\rho}} \omega. \quad (C.2)$$

Using typical values for steel,

$$\begin{aligned} \rho &= 7.85 \text{ g/cm}^3 = 7850 \text{ kg/m}^3 \\ \sigma_m &= 30,000 \text{ psi} = 2.1 \times 10^8 \text{ newtons/m}^2 \end{aligned}$$

obtain for $\omega = 47.1$ (450 rpm)

$$R_b = 4.9 \text{ m (16 ft)},$$

which is a flywheel size larger than considered.

Dimensions of this largest flywheel perhaps would be,

$$\begin{aligned} R &= 4.9 \text{ m (16 ft)} \\ \text{thickness} &= 0.15 \text{ m (0.5 ft)} \\ \text{height} &= 3 \text{ m (9 ft)} \end{aligned}$$

and would weigh,

$$M = 7850 \times 2\pi \times 4.9 \times 0.15 \times 3 = 108,000 \text{ kg}.$$

The polar moment would equal

$$I = MR^2 = 2.6 \times 10^6 \text{ kg-m}^2 .$$

This load appears to be well within present thrust bearing technology as may be seen by examining the table of Kingsbury bearing characteristics given in Kents Mechanical Engineers Handbook⁶. The largest eight-shoe bearing listed in the given table possesses a mean diameter of 63 inches and a load capability of 801 lb_f/in². This is equivalent to a total thrust load of 2.8×10^6 lb_f or the weight of a 1.3×10^6 kg rotor - substantially in excess of the hypothetical largest size flywheel cited above.

APPENDIX D. PROGRAM KINGS8D FOR THE CALCULATION OF SQUARE,
8-SHOE KINGSBURY BEARING CHARACTERISTICS AND
COASTDOWN FLOWRATES.

A Fortran IV program is given for the calculation of square, 8-shoe Kingsbury bearing characteristics including, oil film thickness, mean bearing diameter, shoe dimensions, friction loss rate and, retarding torque constant, K_3 , for Eq. (12). Times for attainment of various coastdown rotation rates are calculated together with the relative flow at that time (REL FLOW), the pump developed pressure in atmospheres (DEV PRES), the ratio of relative decay heat to relative flow (RAT(QREL/DEC.HT)), the oil film thickness in the thrust bearing (FLM THNS) in mils, and the friction loss in the bearing in horsepower (FRC LOSS). A sample output is given. An asterisk in column 1 of the output indicates the oil film has thinned to less than 0.1 mil.

The calculation routine may easily be adopted to other loops by changing a few imbedded values.

**FTN.L.E.G.

PROGRAM KNLS8D

IMPLICIT REAL*8(A-H,O-Z)

REAL*8 K1,K2,K3,MU,MU1,LS,MX,IX,MLB

DIMENSION RPM(20),TITLE(20)

C IMBED

1 PI= 3.1415900

K1= 300.00

KG/M

K2= .1700

M**3

MU= 30.00

CP

RPM0= 450.00

INIT RPM

HMIN=.100

MILMIN

CFFU=.9300

WU=2.00*PI*RPM0/60.00

RPM(1)=400.00

RPM(2)=350.00

RPM(3)=300.00

RPM(4)=250.00

RPM(5)=200.00

RPM(6)=150.00

RPM(7)=100.00

RPM(8)=75.00

RPM(9)=50.00

RPM(10)=40.00

RPM(11)=30.00

RPM(12)=20.00

RPM(13)=10.00

RPM(14)=8.00

RPM(15)=6.00

RPM(16)=4.00

RPM(17)=2.00

RPM(18)=1.00

RPM(19)=.500

RPM(20)=.100

2 READ(50,3)TITLE

3 FORMAT(20A4)

WRITE(51,4)TITLE

4 FORMAT(1H1,20A4)

5 READ(50,6)MX,IX,P0

6 FORMAT(3D10.0)

MLB=2.20500*MX

LB

AS=MLB/(8*P0)

INSD

LS=DSORT(AS)

IN SHOE

D=3*LS

IN BOIAM

MU1=.14500-6*MU

LRSE/INZ

H0=1.03*DSORT(.2800*MU1*D**2*WU/(P0*24))

FTHCKNIL

HSO=.0149400*E*AS*D*WU**1.500/(24**2)*DSORT(24*P0*MU1/.2800)

FRHP/RE

RFLR=100*HSO/5.03

PC RE FR

K3=1.355800*550*HSO/(W0**1.500)

NW-M-SHF

7 WRITE(51,8)MX,IX,RPM0,K1,K2,P0,MU,HMIN,

INPUT

1 K3,D,LS,H0,HSO,RELFR

CALL

8 FORMAT(1H0,'ROTOR MASS (KG)= 'F7.0 /1H ,

MX

1'ROTOR POLAR MOMENT (KG-MSQ)= 'F8.0/1H ,

IX

2'INITIAL RPM= 'F5.0/1H ,

RPM0

3*K1 (KG/M)= 'F5.0/1H ,

K1

4*K2 (M**3)= 'F5.2/1H ,

K2

5'BEARING PRESSURE (PSI)= 'F5.0/1H ,

P0

6'LUKE VISCOSITY (CP)= 'F5.2/1H ,

MU

7'MIN FILM THICKNESS (MILS)= 'F5.2/1H0,

HMIN

8*K3 (NWT-M-SEC**5)= '1PE9.2/1H ,

K3

9'BLARING DIAMETER (IN)= '1PE9.2/1H ,

D

```

1*SHOE LENGTH (IN)= ' 1PE9.2/1H ,
2*INITIAL FILM THICKNESS (MILS)= '1PE9.2/1H ,
3*FRICTION LOSS TO BEARINGS,INITIAL (HP/BEARING)= '1PE9.2/1H ,
4*FRICTION LOSS TO BEARINGS,INITIAL (PLR-CENT)= ' 1PE9.2)
C OUTPUT TABLE *INDICATES FILM TOO THIN, H.LT.HMIN
WRITE(51,9)
9 FORMAT(1H0,
10 RPM TIME REL FLOW DEV PRES RATIO(REL/ FLN THNS FRC
2 LOSS*/1H .
30 SEC ATM DEC HT) MCLS HP
4/PUMP*/1H0)
10 DO20 I=1,20
W=2*PI*RPM(I)/60
H=H0*DSQRT(W/W0)
IF(H.LT.HMIN)11,12
11 TOUTHN=1H*
GOTO13
12 TOUTHN=IH
13 HS=HS0*(W/W0)**1.5
QREL= W/W0
DP=9.8690-6*K1*W**2
C TIME CALC
Z=DSQRT(QREL)
A=K1*K2*W0/(EFF0*IX)
H=K3/(IX*DSQRT(W0))
C1=A/R
X1=2*(1-7)
ARG1=(1+C1)/(2+C1*Z-Z**2)
X2= C1*ULOG(ARG1)
C2=DSQRT(8+C1**2)
XNUM= (-2*Z+C1-C2)*(-2+C1+C2)
XDEN= (-2*Z+C1+C2)*(-2+C1-C2)
ARG2=XNUM/XDEN
X3=(C1**2/C2)*ULOG(ARG2)
RT=X1+X2+X3
T=RT/8
TNOFR=(1/A-1/W0)*IX/(K1*K2)
C
QFQH=QREL/(.07/T**.2)
20 WRITE(51,21)TOUTHN,RPM(I),T,QREL,DP,QFQH,H,HS,TNOFR
21 FORMAT(1H ,A1,7(1PE9.2,1X),10X,1PE10.2)
GOTO 2
22 END

```

MEDIUM FLYWHEEL BEARING PRESSURE=600 PSI

ROTOR MASS (KG)= 70000.

ROTOR POLAR MOMENT (KG-MSQ)= 538000.

INITIAL RPM= 450.

K1 (KG/M)= 300.

K2 (M**3)= 0.12

BEARING PRESSURE (PSI)= 600.

LUBE VISCOSITY (CP)= 30.00

MIN FILM THICKNESS (MILS)= 0.10

K3 (N*1-M-SEC**5)= 4.000 01

BEARING DIAMETER (IN)= 1.700 01

SHOE LENGTH (IN)= 5.670 00

INITIAL FILM THICKNESS (MILS)= 1.070 00

FRICTION LOSS TO BEARINGS, INITIAL (HP/BEARING)= 1.740 01

FRICTION LOSS TO BEARINGS, INITIAL (PER-CENT)= 3.470-01

RPM	TIME SEC	REL FLOW	DEV PRES ATM	RAT (REL/ DEC RT)	FILM THNS MILS	FRC LOSS HP/7PUMP
4.000 02	3.660 01	8.890-01	5.190 00	2.610 01	1.010 00	1.460 01
3.500 02	8.240 01	7.780-01	3.980 00	2.680 01	9.470-01	1.140 01
3.000 02	1.400 02	6.670-01	2.920 00	2.560 01	8.770-01	9.450 00
2.500 02	2.150 02	5.560-01	2.030 00	2.320 01	8.010-01	7.190 00
2.000 02	3.130 02	4.440-01	1.300 00	2.000 01	7.160-01	5.150 00
1.500 02	4.490 02	3.330-01	7.310-01	1.610 01	6.200-01	3.340 00
1.000 02	6.530 02	2.220-01	3.250-01	1.160 01	5.060-01	1.820 00
7.500 01	8.040 02	1.670-01	1.830-01	9.070 00	4.380-01	1.180 00
5.000 01	1.020 03	1.110-01	8.120-02	6.350 00	3.580-01	6.430-01
4.000 01	1.150 03	8.890-02	5.190-02	5.200 00	3.200-01	4.600-01
3.000 01	1.310 03	6.670-02	2.920-02	4.000 00	2.770-01	2.990-01
2.000 01	1.530 03	4.440-02	1.300-02	2.750 00	2.260-01	1.630-01
1.000 01	1.920 03	2.220-02	3.250-03	1.440 00	1.600-01	5.750-02
8.000 00	2.050 03	1.780-02	2.080-03	1.170 00	1.430-01	4.120-02
6.000 00	2.210 03	1.330-02	1.170-03	8.880-01	1.240-01	2.670-02
4.000 00	2.430 03	8.890-03	5.190-04	6.040-01	1.010-01	1.460-02
* 2.000 00	2.810 03	4.440-03	1.300-04	3.110-01	7.160-02	5.150-03
* 1.000 00	3.170 03	2.220-03	3.250-05	1.590-01	5.060-02	1.820-03
* 5.000-01	3.520 03	1.110-03	8.120-06	8.130-02	3.580-02	6.430-04
* 1.000-01	4.260 03	2.220-04	3.250-07	1.690-02	1.600-02	5.750-05

INTERNAL DISTRIBUTION

1. G. M. Adamson	43. D. B. Lloyd
2. L. G. Alexander	44. R. N. Lyon
3. T. D. Anderson	45. M. I. Lundin
4. S. J. Ball	46. H. G. MacPherson
5. S. E. Beall	47. R. E. MacPherson
6. C. G. Bell	48. W. R. Martin
7. M. Bender	49. H. C. McCurdy
8. C. E. Bettis	50. A. J. Miller
9. R. E. Blanco	51. G. Morris
10. H. I. Bowers	52. J. C. Moyers
11. C. C. Burwell	53. M. L. Myers
12. D. D. Cannon	54. R. C. Olson
13. R. S. Carlsmith	55. H. R. Payne
14. R. H. Chapman	56. A. M. Perry
15. W. B. Cottrell	57. C. M. Podeweltz
16. F. L. Culler	58. J. T. Roberts
17. J. G. Delene	59. T. T. Robin
18. D. M. Eissenberg	60. M. W. Rosenthal
19. J. L. English	61. R. Salmon
20. W. K. Ergen	62. J. P. Sanders
21. D. E. Ferguson	63. A. W. Savolainen
22. A. P. Fraas	64. J. E. Savolainen
23. L. C. Fuller	65. W. F. Schaffer
24. C. H. Gabbard	66. M. J. Skinner
25. W. R. Gall	67. I. Spiewak
26. W. R. Grimes	68. B. Squires
27. A. G. Grindell	69. R. S. Stone
28. E. W. Hagen	70. D. A. Sundberg
29. R. P. Hammond	71. W. E. Thompson
30. P. N. Haubenreich	72. D. B. Trauger
31. E. C. Hise	73. C. S. Walker
32. H. W. Hoffman	74. A. M. Weinberg
33. J. E. Jones	75. J. R. Weir
34. J. K. Jones	76. G. D. Whitman
35. P. R. Kasten	77-81. R. P. Wichner
36. O. H. Klepper	82. L. V. Wilson
37. W. H. Kelley	83. M. M. Yarosh
38. J. O. Kolb	84. G. Young
39. R. B. Korsmeyer	85-86. Central Research Library (CRL)
40. K. A. Kraus	87-88. Y-12 Document Reference Section (DRS)
41. J. A. Lane	89-91. Laboratory Records Department (LRD)
42. C. G. Lawson	92. Laboratory Records Department -
Record Copy (LRD-RC)

EXTERNAL DISTRIBUTION

- 93-107. Division of Technical Information Extension (DTIE)
108. Laboratory and University Division, ORO